RECLAMATION
Managing Water in the West

Design Guide for Heating, Ventilating, and Air Conditioning Systems

U.S. Department of the Interior
Bureau of Reclamation
Denver, Colorado

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Mission Statements

The mission of the Department of the Interior is to protect and provide access to our Nation's natural and cultural heritage and honor our trust responsibilities to Indian Tribes and our commitments to island communities.

The mission of the Bureau of Reclamation is to manage, develop, and protect water and related resources in an environmentally and economically sound manner in the interest of the American public.

Design Guide for Heating, Ventilating, and Air Conditioning Systems

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Design Guide for Heating, Ventilating, and Air Conditioning Systems

A. SCOPE.—This guide assumes that prospective users will have some heating, ventilating, and air conditioning (HVAC) design experience but are not familiar with the specific requirements for the Bureau of Reclamation (Reclamation) facilities. The guide emphasizes design considerations and recommendations for applications specific to powerplants, pumping plants, and confined spaces such as tunnels, gate chambers, and valve vaults. The guide also identifies recognized codes and standards that should be referred to as necessary when designing HVAC systems for Reclamation facilities.

Most HVAC system designers should be familiar with the design requirements for office buildings. Except for recommended ambient conditions, office HVAC systems are not included in this guide.

Heating and cooling load calculations and procedures for Reclamation facilities are essentially similar to those for most structures familiar to the HVAC designer and are not addressed in this guide. The calculations and procedures are readily available in many of the references noted throughout the guide and listed in the appendix.

Throughout this guide, the term “Plant” applies equally to powerplants and pumping plants and the term “Units” refers equally to the main generating units, pump/generating units, or pump units unless otherwise stated.

B. **DESIGN CONSIDERATIONS**.—Plant air conditioning systems may include: heating, ventilating, cooling, humidifying, dehumidifying, filtering, and air distribution to maintain acceptable indoor plant environments. The primary objective of plant air conditioning systems are to ensure human comfort, health, and safety, and to ensure equipment protection. HVAC equipment must satisfy sanitary, hygienic, industrial, fire and emergency requirements and must comply with established standards for construction, installation, performance, energy conservation, and safety.

1. **Load Calculations and Weather Conditions.**—Heating and cooling loads should be calculated in accordance with the procedures outlined in the latest edition of *American Society of Heating, Refrigerating, and Air Conditioning Engineers (ASHRAE) Handbook of Fundamentals*. Design Heat Transmission Coefficients used for cooling and heating loads should be obtained from the Plant Structures Branch and should reflect the actual materials to be specified. Heat transmission coefficients from *ASHRAE* should be used only when actual data are not available such as preliminary calculations.

The outdoor air design temperature for heating calculations should be obtained from the project design data or the *ASHRAE Handbook* if insufficient design data are available. Because of the high potential for damage due to freezing water in pipes, the temperature selected for calculations should be the 99 percent value. During a normal winter, there would be approximately 22 hours where the outdoor temperature would be below the 99 percent value during the months of December through February.

The summer outdoor air-dry bulb and wet bulb design temperatures and daily temperature ranges should be determined as explained in the *ASHRAE Handbook of Fundamentals* for the area of construction. For office air conditioning (AC) systems where energy conservation should be a significant consideration, the wet and dry bulb temperatures selected for calculations should be the 2-1/2 percent value. At this value, the outdoor dry bulb temperature may be exceeded approximately 75 hours during the months of June through September. For plant cooling, continued operation under the most extreme conditions is the primary consideration. Therefore, the extreme design conditions should be used.

HVAC calculations rely on lengthy, iterative procedures that are very tedious if performed manually. Many software packages are now available to improve accuracy and reduce calculation time. These packages usually contain a complete set of *ASHRAE* weather data, and calculations designed for specific air system configurations i.e. constant volume (CV) and variable air volume (VAV), variable volume and temperature, etc. Reclamation projects are seldom constructed at one of the locations shown on the *ASHRAE* weather data, however, the data can be adjusted to approximate local conditions. Designers should be aware of the intended applications and underlying assumptions used by the software designer.

2. **Plant Design and Construction**.—Small plants are generally designed to house most of the heat producing equipment such as motors, generators, switchgear, and busses in one large open bay. The HVAC system for this type of plant is the easiest to design due to the simplicity of the air distribution system and controls required. Larger plants usually contain a service bay area and several equipment rooms, shops, offices, control rooms, lunchrooms etc. Most equipment can operate over a wide range of temperatures; therefore, the rooms have similar ambient design requirements and may be conditioned by a

single HVAC unit. Some rooms such as uninterrupted power supply (UPS), communication, office, and control rooms generally have different ambient requirements and may require conditioning by separate HVAC units.

Plants are normally divided into unit bays where each bay is equal to the portion of the structure required for the unit. Unit bays may be cooled simultaneously by a central air conditioning system, individually by dedicated HVAC equipment, or by a combination of centrally located and unit specific HVAC equipment depending on the plant layout, heating and cooling loads, and the type of temperature controls desired.

Central systems will generally maintain the most uniform temperature conditions throughout the plant and can be designed for general smoke evacuation. Central systems may become complicated when the plant is compartmentalized requiring extensive use of exposed and embedded ducts, fire dampers, smoke dampers, and control dampers. Furthermore, unless the system includes redundancy, failure of the central air conditioning system may cause plant temperatures to rise excessively and may cause shutdown of some units. However, plant shutdowns due to HVAC system failure are rare and are most likely occur when the units are cooled with 100 percent unconditioned outdoor air. HVAC equipment commonly used for this type of system may include air-handling units supplying unconditioned or conditioned air. When conditioned air is required, the following options are generally considered in the order given: air handling unit with cold water coils using site water (generally limited to water temperatures below 65 degrees Fahrenheit [ºF]); direct evaporative coolers; indirect/direct evaporative coolers; chilled water, and direct expansion (DX) coils. Well water is also an option if a sufficient quantity is available. It should be noted that many local jurisdictions forbid the use of treated (potable) water for cooling HVAC heat rejection equipment such as condensers.

Reclamation practice has generally been to provide indoor mechanical equipment rooms for centrally located HVAC equipment. Indoor equipment rooms are most often provided to protect equipment from vandalism. When a centralized location is not possible, dedicated HVAC equipment may be located near the unit(s) to be cooled. The *ASHRAE Handbook – HVAC Systems and Equipment* provides the following recommendations for sizing mechanical equipment rooms: The floor space requirements for mechanical equipment room's varies from 4 to 9 percent of the gross building area, however, most buildings require an equipment room ranging from 6 to 9 percent of the gross floor area. Ceiling height should be 12 to 20 feet depending on the amount of overhead ductwork to be used. Interior shafts for ductwork should have an aspect ratio ranging from 2:1 to 4:1 to facilitate installation of ducts between the mechanical room and shaft. Shafts should be sized 10 to 15 percent larger than initial requirements to allow space for future modifications. When possible, separate supply and exhaust shafts should be provided to enable use of the exhaust chase as a plenum.

Depending on the type of control desired, the dedicated HVAC equipment may be interlocked to operate with the unit to be cooled or may be controlled by a remote thermostat located near the unit. If the interlocking control scheme is used, any heating equipment provided must be independently controlled to provide freeze protection when the units are shutdown during winter months. HVAC equipment commonly used for this type of system may consist of: basic air handling units supplying unconditioned air, direct evaporative cooling units with positive exhaust fans, water coil units (use site water or well water if cool enough and available in sufficient quantity), or chilled water cooling units. Packaged vertical DX type AC can also be used for computer rooms or office areas; however, they are rarely used for plant cooling applications.

Where all units are identical in size and layout, the air conditioning systems for each bay should also be identical, to the maximum extent possible, to simplify the HVAC system design and minimize the number of spare parts which must be stored.

Air conditioning systems for bays containing service areas, office spaces, control rooms, computer rooms, and visitor facilities should be designed so that multiple pieces of equipment identical to the equipment used in unit bays can be furnished where feasible. Dedicated air conditioning system with independent controls should be provided for offices and control rooms since the ambient temperatures in these areas must often satisfy human comfort and frequently house temperature sensitive equipment.

3. **Special Environmental Conditions.**—Some plant rooms such as control, computer and UPS rooms, lunch rooms, and office spaces require tighter environmental control to avoid wide temperature fluctuations, to ensure proper operation of equipment, and to provide comfort conditions for sedentary personnel. Furthermore, since these rooms may be occupied when the plant HVAC system is shutdown,

the HVAC systems and controls should be completely independent of the Plant HVAC system. In most instances, these spaces require similar environmental conditions and a single centrally located HVAC system may be used.

4. **Redundant Systems or Equipment.**—When inadvertent temperature shutdown of the main units cannot be tolerated, redundancy must be considered. Redundancy may be complete or partial. Complete redundancy generally consists of two equally sized air conditioning systems, with each system capable of providing the total heating and cooling requirements. This degree of redundancy is the most fail-safe, however, it is also the most expensive, difficult to justify, and seldom used.

   Frequently, redundant equipment is sized for 50 to 75 percent of the total capacity required to maintain the design conditions. This may or may not be acceptable depending on whether it results in under sizing or over sizing of equipment. A more reasonable approach when sizing for partial redundancy, is to select each piece of equipment to maintain the minimum or maximum requirements that will prevent damage to equipment or jeopardize the health of occupants. For example, Reclamation plants are typically designed to maintain temperatures of 45 ºF in winter and 85 to 95 ºF in summer. The extremes to be avoided are freezing of standing water and ambient temperatures of 104 ºF that is the maximum recommended ambient temperature required to prolong insulation life. Therefore, the full load (combined) capability of the HVAC system would be sized to maintain the minimum temperature at 45 ºF and the maximum temperature at 85 to 95 ºF. However, each piece of equipment would also be capable of maintaining the low limit at 35 ºF and the high limit at 102 ºF.

   The controls for redundant systems or equipment should include automatic and manual switching to equalize wear on all components. The automatic switching controls should also energize redundant or backup equipment when the primary component fails.

   Although redundant systems can be very expensive, their use may be justified for a particular project. Designers should contact project personnel to discuss the criticality of maintaining the plant on-line under extreme temperature conditions and to determine the degree of redundancy necessary or required. These discussions should take place early in the design phase before proceeding with complete air conditioning system designs and equipment selections.

5. Basic HVAC Scheme for Plants

a. **Location of Air Intakes and Exhaust Openings.—**When possible, cooling should begin with 100 percent outdoor air. Outdoor air louvers should be located opposite major heat sources such as electrical control equipment and transformers. The bottom of the intake louvers should be located a minimum of 2-feet from ground level. Locate louvers away from air polluting sources, such as engine generators, and building exhaust air openings. Avoid locations near light fixtures that tend to attract insects at night. The *Uniform Building Code* (UBC) provides guidelines for relative location between intake and exhaust air openings. When specific information is not available, allow 50 to 75 feet of space between air intake and exhaust openings.

In areas where noise generated by the plant is limited by locally established noise criteria, the location and design of intake openings should be given special consideration. Sound insulation or attenuating equipment may be necessary to prevent unacceptable plant or air conditioning system noise from reflecting through the air intakes into the yard and surrounding area.

b. **Airflow.—**The substructure levels of Reclamation plants are always cooler than the main level. In general, air distribution systems should be designed to supply air to the lower levels of the plant and return air from the upper levels. This will allow the warm air to pre-cool as it flows over exposed cold surfaces, such as pipes, discharge lines and penstocks. Air is then circulated and transferred up through areas containing major electrical equipment where many of the heat gains are concentrated. Air supplied directly to generator or motor rooms is usually too hot to transfer to other areas; therefore, it should be exhausted or returned to the air conditioning unit as return air.

Air transfer fans are commonly used to move air around the plants. These fans are especially advantageous when space constraints prevent or limit installation of ductwork. Air transfer is normally across the plant and upwards. Air should flow the length of the bay before transferring to the next room or level. The air temperature entering and leaving each space should be calculated to ensure that adequate cooling is still possible. If necessary, spot coolers can be located to provide supplemental cooling.

c. **Substructure Heating, Cooling, and Ventilation**.—The substructures of plants are heated and cooled to maintain minimum and maximum temperatures for equipment protection, to prevent freezing, and to ensure personnel comfort. The minimum ambient design temperature is usually 45 ºF for equipment freeze protection. Maximum ambient design temperatures may range from 85 to 95 ºF when refrigeration type conditioning equipment is used, to 104 ºF when minimally conditioned or unconditioned air is used. Areas, such as control rooms and offices that are occupied by sedentary personnel are maintained at 68 to 75 ºF.

Air transfer from unit bays to oil, paint, battery, toilet, sumps, janitor, welding, or chemical storage rooms is the preferred method for ventilating these spaces. However, air from these rooms is considered contaminated and must be directly exhausted outdoors away from air intake louvers.

d. **Superstructure Heating, Cooling, and Ventilation.**—In general, superstructures should not be cooled or heated except in those cases where: temperatures must be maintained to protect equipment from freezing or excessively high temperatures; and Condensation of water on metalwork will cause corrosion or other damage to the metalwork, or where water droplets will cause damage to equipment below.

Superstructure ventilation system should be independent of the main HVAC system. The space below the high superstructure ceiling tends to get very hot due to external solar heat gain and rising warm air from internal heat sources. If undisturbed, the air will stratify with the hottest air immediately below the ceiling and progressively cooler air moving towards the conditioned space below. Allowing air to stratify can reduce the cooling load in the conditioned space. Except for the radiant effects (approximately 50 percent for fluorescent and 65 percent for incandescent) the heat gain from lights can be stratified. Furthermore, only the radiant effect from the roof and upper walls (approximately 33 percent) actually reaches the floor level. Because of stratification, only the lower part of the superstructure, i.e. the occupied area needs to be cooled with conditioned air. The following equations (1) and (2) (developed from *Jennings*) may be used to estimate ceiling and average temperatures (in ºF) in superstructures due to stratification, i.e., no fans opposing heat rise:

(1) Ceiling temperature (T_c) for room heights (H) less than 20 ft, is given by:

$$T_c = [1.00 + 0.02h]\ T_b$$

Where: h = the room height in feet above the breathing line not to exceed 15-feet,

T_b = the breathing line temperature. The breathing line is measured 5 feet above floor level in °F.

(2) Average room temperature (T_{avg}) for room height (H) less than 20 feet is given by:

$$T_{avg} = [1.00 + 0.02(H/2 - 5)]\ T_b$$

Note: $T_{avg} = (T_c + T_f)/2,$

Where: T_f = *Floor temperature at* $h = -5$

When the roof fans are operating, stratification is not possible, and the temperature difference between the floor and ceiling may be small. The following equations (3) and (4) (developed from *Jennings*) may be used as a conservative estimate for the average temperatures (in °F) in superstructures with fans opposing heat rise.

(3) Average temperature (T_{avg}) for a ceiling height (H) less than or equal to 15 feet is given by:

$$T_{avg} = [1.00 + 0.01(H - 5)]\ T_b$$

(4) Average temperature (T_{avg}) for a ceiling height (H) greater than 15 feet is given by:

$$T_{avg} = (1.1)\ T_b + 0.1(H\text{-}15)$$

During the winter, stratification is not desirable because the heat supplied to the conditioned space rises towards the ceiling where it is lost due to heat transfer through the ceiling and upper wall areas. Multi-purpose power roof fans capable of supplying, exhausting, and circulating air provide a means of coping with stratification and the cooling and heating requirements of superstructures. These fans consist of a propeller fan, housing, hood, filters and control dampers. The fans can be purchased as preassembled package units with associated controls or may be assembled from components as required. The powered roof exhausters are controlled by a thermostat to operate continuously in the exhaust mode when 100 percent outdoor air is cooling the generator or motor room floor. When the HVAC system switches to conditioned air, the roof exhausters should de-energize and the air allowed to stratify to reduce the cooling load. During winter, the fan controls should switch to divert hot air from the ceiling towards the floor to reduce heating load.

6. **Ductwork**.—Ductwork should be designed in accordance with the provisions of the *ASHRAE Handbook of Fundamentals*, or the *Sheet Metal and Air Condition Contractor's National Association (SMACNA) HVAC System Duct Design Manual*. Installation of ductwork should comply with *SMACNA HVAC Duct Construction Standards*.

a. Design Considerations

(1) ***Minimize the number of fittings.***—Fittings are expensive and cause significant increases in pressure loss compared to straight duct.

(2) ***Use semi-extended plenums***.—Plenums reduce the number of transition fittings and facilitate balancing. Figures 1, 2, and 3, illustrate how use of the extended plenum concept can reduce the number of fittings and simplify a duct design.

(3) ***Seal ductwork.***—Standard duct sealing requirements are shown on table 1.

Table 2 shows air leakage from the longitudinal seams of unsealed ductwork. The Longitudinal seam leakage for metal duct is approximately 10 to 15 percent of the total duct leakage.

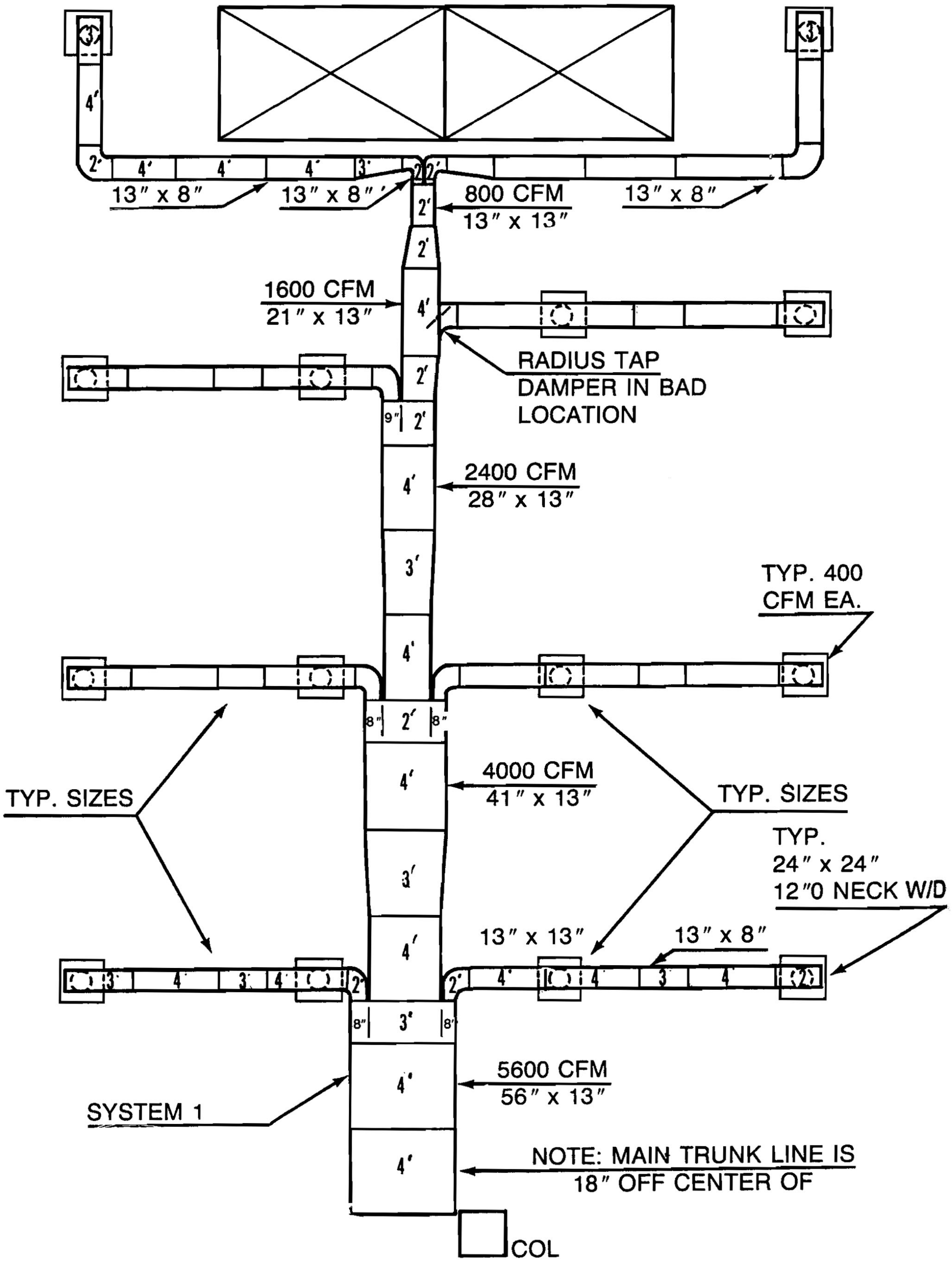


Figure 1.—Duct layout – original design.

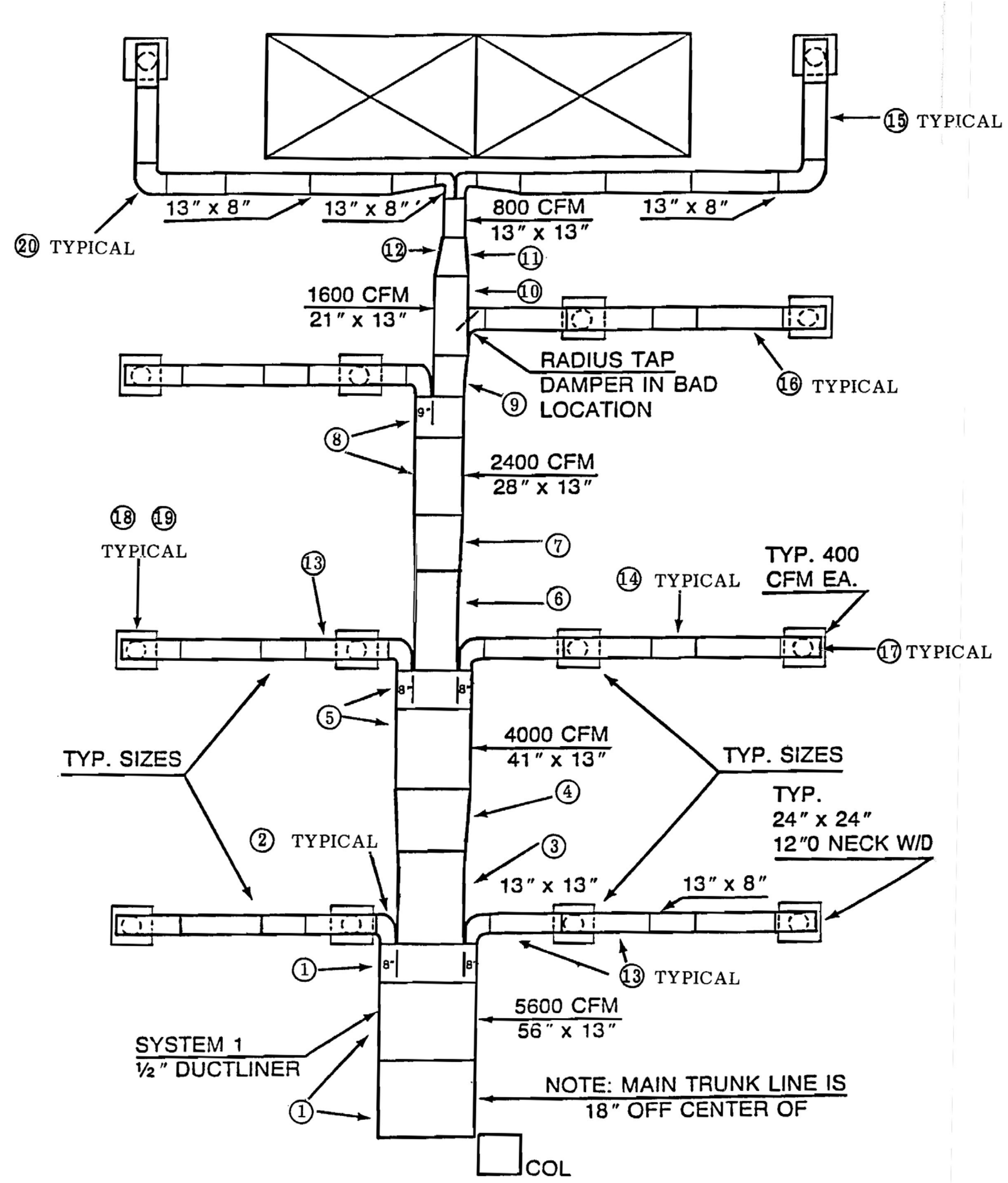


Figure 2.—Duct layout – redesign.

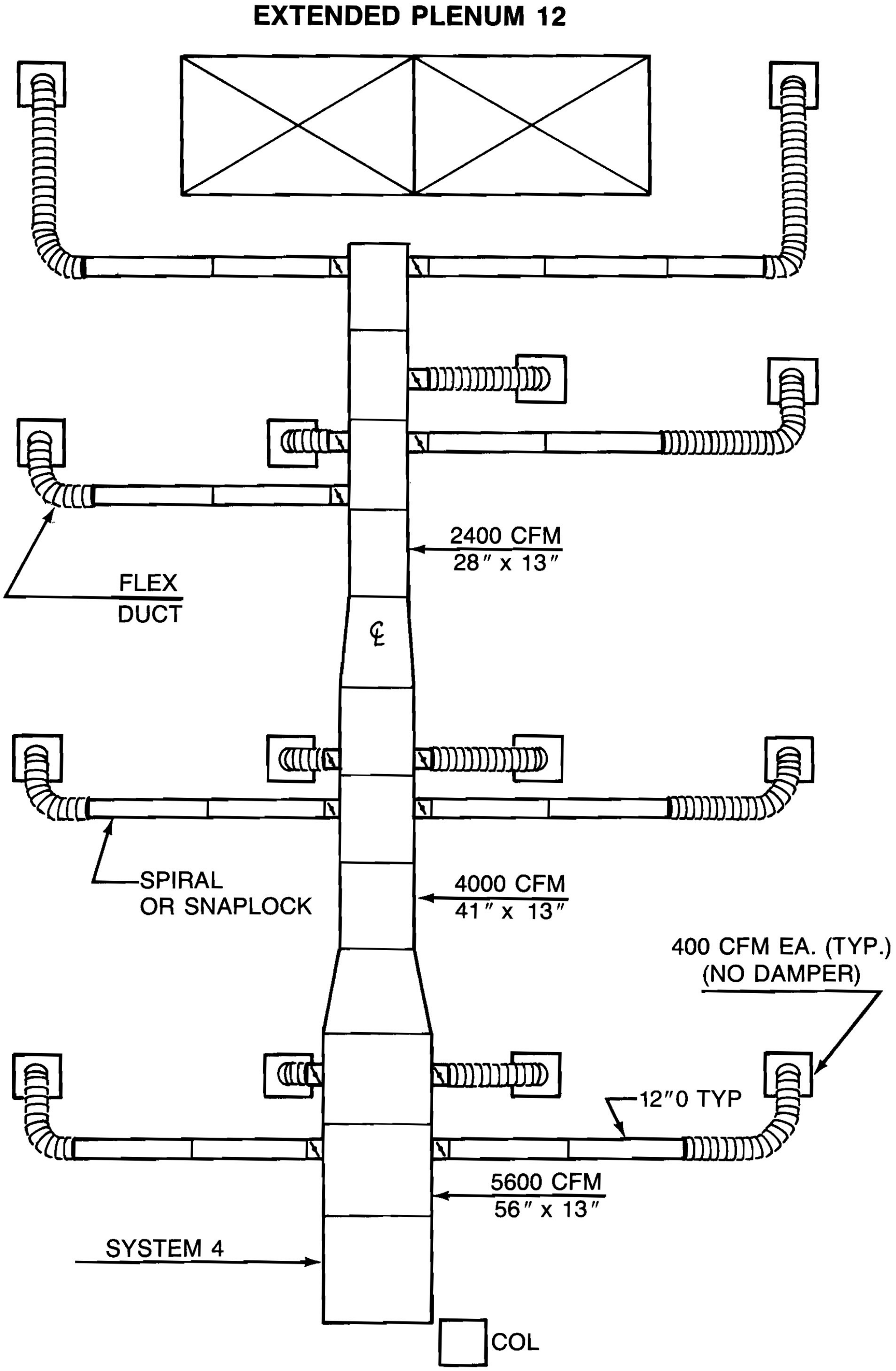


Figure 3.—Duct layout – extended plenum design.

Table 1.—Standard duct sealing requirements

Seal class	Sealing requirements	Pressure class	
		inches w.g.	Pa
A	Class A: All Transverse joints, longitudinal seams, and duct wall penetrations	4- and up	1,000
B	Class B: All Transverse joints and longitudinal seams only	3	750
C	Class C: All Transverse joints only	2	500

In addition to the above, any variable volume system duct of 1-in. w.g. (250 Pa) and ½-in. w.g. (125 Pa) construction class that is upstream of the VAV boxes shall meet Seal Class C. SMACNA HCAC Duct Construction Standards, 1995.

Table 2.—Unsealed longitudinal seam leakage for metal ducts

Type of duct/seam	Leakage[1]			
	cfm/ft (seam length)		l/s per meter (seam length)	
	Range	Average	Range	Average
Rectangular				
Pittsburg Lock	0.01 to 0.56	0.16	0.015 to 0.87	0.25
Button Punch Snap Lock	0.01 to 0.16	0.08	0.015 to 0.25	0.10
Round				
Snap Lock	0.04 to 0.14	0.11	0.06 to 0.22	0.17
Grooved	0.11 to 0.18	0.12	0.17 to 0.28	0.19

[1] Leakage rate is at 1-inch w.g. (250 Pa) static pressure. SMACNA HVAC Duct System Design, 1990.

Table 3 shows applicable leakage classes and can be used in conjunction with the following equation to estimate the amount of air leakage for a given duct system:

$$F = C_L P^N$$

Where: F = leakage rate (cfm/100 ft^2 duct surface)
C_L = Leakage Class Constant (table 3)
P = Static pressure, inch wind generated (w.g.)
N = 0.65, exponent related to turbulence

Example: Estimate the air leakage from a round metal distribution system with surface area A = 800 ft^2 and static pressure P = 2.5 inch w.g.

Table 3 does not show a 2.5 inch w.g., therefore, use 3 inch w.g. to obtain a leakage class constant $C_{L\,=}\,6$.
The estimated leakage rate is:

$$F = (6)\,(2.5^{\,0.65}) = \underline{10.9\ \text{cfm}/100\ \text{ft}^2}$$

and the estimated total leakage Q_L is:

$$Q_L = AF = (800)\,(10.9/100) = \underline{87.2\ \text{cfm}}$$

(4) ***Use round duct.***—For a given perimeter, round duct has lower friction loss than rectangular duct.

(5) ***Reduce aspect ratio.***—The surface area and cost of ductwork increases with increased aspect ratio (H/W). On rectangular ducts maintain an aspect ratio as close to 1 as possible. Try not to exceed a ratio of 1:4

b. **Galvanized Duct.**—Galvanized duct is commonly used and is the preferred material for Reclamation applications.

(1) ***Pressure class***.—Specify and use the lowest pressure class. If no class is given, *SMACNA* standards allow a default value of 1-inch for systems other than VAV, or 2-inches for VAV systems. The default value may or may not be adequate for the intended application. The pressure class has significant effects on the metal gauge thickness and reinforcement requirements for the duct. Pressure classes are shown in table 4.

Table 3.—Applicable leakage classes[1]

Duct class (see table 4)	½, 1, 2, inch w.g.		3 inch w.g.	4, 6, 10 inch w.g.
Seal class	None	C	B	C
Applicable sealing	N/A	Transverse Joints Only	Transverse Joints and Seams	All Joints, Seams, and Wall Penetrations
Leakage class constant (C_L) cfm/100 ft^2 at 1 inch w.g.				
Rectangular metal	48	24	12	6
Round and oval metal	30	12	6	3
Rectangular fibrous glass	N/A	6	N/A	N/A
Round fibrous glass	N/A	3	N/A	N/A

[1] The leakage classes listed in this table are average values based on tests conducted by SMACNA/ASHRAE/TIMA. Leakage classes listed are not necessarily recommendations on allowable leakage. The designer should determine allowable leakage and specify acceptable duct leakage classifications. SMACNA HVAC System Duct Design, 1990.

Table 4.—HVAC duct pressure – velocity classification[1]

Static pressure class inch w.g.	Operating pressure in. w.g.	Type of pressure	Seal class	Maximum velocity ft/min
½	up to ½	Pos/Neg	C	2,000
1	over ½ to 1	Pos/Neg	C	2,500
2	over 1 to 2	Pos/Neg	C	2,500
3	over 2 to 3	Pos/Neg	B	4,000
4	over 3 to 4	Pos	A	4,000
6	over 4 to 6	Pos	A	As specified
10	over 6 to 10	Pos	A	As specified

[1] Modified version of Table 4-1 shown in SMACNA HVAC System Duct Design, 1990.

(2) ***Galvanized coating.***—Sheet steel should be G-60 (for dry or low humidity) or G-90 (for damp or wet areas) coated galvanized steel of lock forming grade conforming to *American Society for Testing and Materials (ASTM) A653* and *A924* standards.

(3) ***Reinforcement.***—Depending on the pressure class, duct dimensions, and metal gauge, duct may or may not require reinforcement. *SMACNA Duct Construction Standards – Metal and Flexible*, dictates when and what type of reinforcement is required. Therefore, designers may select the pressure class, metal gauge thickness and reinforcement requirements or choose to specify the pressure class requirements and leave the details to the contractor. The latter has been the more common approach on Reclamation jobs.

(4) ***Leakage class.***—All duct constructed for ½ and 1-inch pressure classes should be sealed to Class C requirements. Standard sealants are not intended for extreme applications such as weatherproof, waterproof, or ultra-violet resistant; temperatures above 120 ºF; submerged below water; or totally leak free. Where any of these conditions are to be encountered, special sealants should be specified.

(5) ***Leakage testing.***—Unless the application is critical, leakage testing of systems below 3-inch pressure class is not recommended due to the expense. Government inspectors should ensure that the contractors work reflects good quality control. Ductwork rated above 3-inch pressure class should be leak tested only if the expense can be justified.

c. **Polyvinyl Chloride (PVC) Duct.—**PVC duct is not widely used in Reclamation projects. The most common applications for PVC duct are systems exhausting corrosive fumes. PVC duct can be used for 2-inch, 6-inch, and 10-inch negative and positive pressure applications in round and rectangular shapes. Because PVC material is seldom used, no further discussion is included in this manual. Designers should refer to *SMACNA Thermoplastic Duct PVC Construction Manual* for PVC duct construction standards, guide specifications, and material specifications.

d. **Fiberglass Reinforced Plastic (FRP) Duct.**—FRP duct is only used in tunnel ventilation systems where high moisture or wet conditions prevail. Refer to the section on confined spaces in this manual for additional information concerning use of FRP duct.

7. **Heating Equipment.**—Most plants are constructed at remote locations where natural gas is not available. Propane is not used for heating applications due to storage, the below grade location of mechanical rooms, and the logistics of frequent refilling at remote locations. Primary heating is usually provided by electric duct heaters or unit heaters. Electric heaters should be avoided in spaces containing explosive gases or atmospheres. Heaters for these spaces must comply with the *National Fire Protection Association (NFPA) 70* – otherwise known as the *National Electric Code (NEC)*, explosive-proof construction requirements. Many Reclamation plants operate throughout the year; therefore, these plants are capable of generating recoverable heat. When possible, the heat generated by electric motors or generators should be recovered and used to temper cooler air.

a. **Duct Heaters**.—Duct heaters are available in two configurations, open element and finned-tube. The open element tends to run cooler, have very low pressure drops, are much lighter, have greater clearance between the elements, and are lower cost. Finned tube coils are safer because the element is encased, less prone to damage form airborne contaminants, can be used in hazardous locations, easier to service individual elements, rugged construction provides greater mechanical stability and protection from physical damage, more uniform airflow reduces hot spots, enable more precise control due to higher thermal inertia. Finned-tube electric resistance coils are preferred over the open element (blast) coils due to the more rugged construction and safety.

(1) ***Minimum velocity requirements.***—Duct heaters require minimum airflow velocities to ensure safe operation. Design velocities should be obtained from manufacturer's data. Designers should verify that duct sizes and airflows are selected to provide the minimum velocities required. This is especially important in multi-speed fan applications where airflows may vary significantly between the minimum and maximum requirements. In multi-speed fan applications, single stage heaters should be avoided and multi-stage or

modulated heater control should be provided. Regardless of which fan speed or heating strategy is used, the air velocity into the coil must always exceed the minimum velocity required for the heater output at any given time.

(2) ***Installation considerations.***—Heaters are also available for flanged or slip-in installation. The location of electric duct heaters within a duct is very important to ensure uniform flow across the entire face of the heater. Heaters must be installed in accordance with the terms of their *Underwriters Laboratories (UL)* listing. Duct heaters should be installed a minimum of 4 ft downstream and upstream from obstructions that may cause non-uniform airflow such as fans, elbows, contractions, expansions, and filters.

Occasionally the duct dimensions are larger than standard heater dimensions. Custom heaters are not necessary for these conditions. One major heater manufacturer indicates that standard slip-in heaters can be used with oversize ducts, and no screens or plates, provided the heater area (HxW including frame) occupies at least 80 percent of the duct cross sectional area. This type of installation should be verified by the heater manufacturer for any given installation. Typically, perforated plates or screens are installed in the open space between the standard heater and the duct to equalize the airflow over the cross-sectional area of the duct and heater. The plate or screen should have at least 50 percent free area.

For general heating, the coils are usually installed in the main air-handling units. However, heaters are frequently installed in the supply air ducts to provide supplemental heating when necessary.

(3) ***Safety devices.***—Various safety devices are used to prevent failures due to overheating. Differential pressure switches are normally installed across the duct heater to de-energize the heater when sufficient airflow is not provided. High limit thermostats are normally installed downstream of the duct heater to de-energize the heater if the air temperature rises to predetermined set points usually 135 to 165 °F.

b. **Unit Heaters**.—When centralized air systems are not used, or when spot heating is necessary, wall mounted unit heaters containing a heating coil, fan, and integral thermostat are recommended. Unit heaters are generally mounted along the exterior plant walls. The installed height varies with heater size. For best results, unit heaters should be oriented to discharge air parallel to the walls to create a circular airflow pattern. One or two 10 to 15 kW heaters should be provided near large overhead service bay doors to provide supplemental heat when the doors are opened. Safety devices are included with the heaters. Thermostats may be integral or remote. Integral thermostats are satisfactory when equipment protection is the primary objective. When heaters are located high above the occupied zone, and personnel comfort is the main objective, remote thermostats installed 5-feet above floor level (commonly referred to as the 5-foot breathing line) provide the best ambient control.

c. **Convection Heaters**.—Floor or wall mounted convection heaters are commonly used for spot heating in offices or toilets. Radiant panel heaters are usually restricted to toilets. The panels may be ceiling mounted (8-foot ceilings only) over commodes or on adjacent walls.

d. **Portable Heaters.—**Portable electric heaters are primarily used to provide supplemental heat in work areas. Sizing and acquisition of portable heaters are not normally included in Reclamation designs.

8. **Cooling Systems.—**In plants having units rated over 1,500 kilowatts per unit, the generators or motors should be furnished with integral water-cooled systems. Water-cooled units generally need 0.8 gal/min per kilowatt of motor or generator loss. The maximum ambient (discharge) temperature for water-cooled units is 40 ºC (104 ºF). The plant air conditioning units should handle all other loads within the plant.

When the main motor or generator units are air cooled, cooling can be provided by ventilation, evaporative cooling units, water coil cooling units, or refrigeration equipment. Frequently, cooling water is available directly from reservoirs, penstocks, or discharge lines. When the water temperature is too high (above 60 ºF) refrigerating cooling equipment will probably be required. In these cases, water can be used for heat rejection from refrigeration equipment compressors or condensers.

Supply air should be directed at the cold surfaces of discharge lines and penstocks to remove excess moisture from the air and to cool the air before using it to cool areas with high heat sources.

a. Ventilation Cooling.—Ventilation cooling (100 percent untreated outdoor air) can be provided in locations where summer design temperatures are at least 10 ºF cooler than the required indoor ambient temperature. Air cooling with temperature differential less than 10 ºF is usually not practical because of the very high air flows required.

(1) ***When used.***—Ventilation cooling systems are most common in small pumping plants such as those which employ vertical turbine or floor mounted centrifugal pumps where motors do not exceed 450 horsepower (Hp). The pump motors at Glendive Pumping Plant (Buffalo Rapids Project, Montana) are rated at 1,500 Hp and are air cooled. This plant and others, with much smaller units, have a history of unit shutdowns due to overheating caused by inadequate cooling.

(2) ***Airflow requirements.***

(a) *Air cooled motors*.—Air cooled motors or generators usually require cooling air flows of 100 to 125 ft^3/min per kilowatt of motor or generator loss. The air is usually supplied to the units at a maximum temperature of 104 ºF and is warmed 50 to 68 ºF in passing through the motors before it is rejected into the same space.

(b) *Indoor dry-type transformers.*—Indoor transformers require airflows approximating 100 ft^3/min per kilowatt of transformer loss. A maximum ambient temperature of 104 ºF is normally recommended for transformer rooms at sea level. This temperature is an average for a 24 hour period. The maximum ambient temperature at any time should not exceed 18 ºF above the maximum average temperature or 122 ºF. Generally the exhaust air should not exceed 27 ºF over the inlet air temperature. The recommended ambient temperature varies with the transformer class and altitude. A transformer specialist should be consulted to determine the transformer class proposed. At 3300 feet above sea leave, the maximum ambient temperature should not exceed 86 ºF. Above 3300 feet the ambient temperature for Class AA 60 and 80 ºC transformers should be lowered approximately 3 and 4 ºF respectively for every 3000 foot increase in elevation. For Class AA/FA and AFA 60 and 80 ºC transformers the ambient temperature should be lowered approximately 6 and 8 ºF respectively for every 3000 foot increase in elevation. When natural ventilation is used to cool transformer rooms, inlet and outlet openings should be sized to provide a minimum free area of 1 ft^2 for transformers totaling less than 50 kVA plus 1 ft^2 for every 100 kVA transformer (provided not less than 3-$inch^2$ free area per kVA in service). Intake openings should be located near the floor and exhaust openings near the ceiling.

(3) ***Design considerations.***—Most air cooled motors draw cool air from the surrounding space near the floor and exhaust hot air into the same space through openings near the top of the motor. To avoid excessive drafts, when possible, air should be ducted directly to the unit air intakes and exhaust. However, ducting should not be installed as a retrofit without consultation with the motor/generator manufacturer. Addition of ductwork to a unit designed for free air induction may actually hinder airflow and cause the motor to overheat. This condition was a contributing factor in the overheating problems experienced at the Glendive Pumping Plant. Outdoor air is normally supplied directly to the pump or generator room.

On single story plant, a typical ventilating system arrangement includes several powered roof exhausters with outdoor air intake louvers along the walls. Another common arrangement employs propeller fans mounted low along one long wall and exhaust louvers along the opposite wall near the ceiling.

On multi-level plants, an air handling unit may supply air to the cooler lower level of the plant where the air is pre-cooled by cold surfaces such as water pipes and below grade concrete walls. The air is allowed to flow upward through floor openings directly below the unit air intakes.

Plants cooled by ventilation air only tend to be very noisy because of the high airflows required. Noise sources should be evaluated and preventive measures implemented to maintain acceptable levels. Refer to the *ASHRAE Noise and Vibration Control* for design, specifications and construction considerations.

b. Cold Water Cooling.—Air handling units with water coils have been used in many Reclamation applications. Water must be cold enough to provide adequate cooling. Water drawn from the lower levels of reservoirs is colder and better suited for AC applications. Water is usually taken from a tap connected to the discharge/penstock lines that may provide the required pressure and eliminate the need for pumps. Alternatively, water may also be taken from the suction line or draft tube. Normal airflow velocity is 500 ft/min. Normal water flow velocity is approximately 3 ft/sec. Generally, the difference between the entering water temperature and the leaving air temperature is approximately 11 to 12 degrees.

(1) ***Plant cooling.***—For plant cooling where equipment protection is the primary consideration, the water temperature should not exceed 65 ºF. This will provide a supply air temperature of approximately 76 to 77 ºF, and maintain a room temperature of 85 to 90 ºF. With the narrow temperature differential between the room air and supply air, supply airflows may be high if the cooling loads are also high. However, since plants are basically unoccupied, drafts can be tolerated.

(2) ***Occupied spaces.***—Typical supply air temperatures for office spaces is 56 ºF with a 20 to 25 ºF differential between the room and supply temperatures. These conditions provide good air movement while preventing excessive drafts. For occupied spaces where human comfort is essential, the water temperature should not exceed 50 ºF. However, this will result in above normal supply air temperature of approximately 61 ºF and require above normal airflows. Furthermore, excessive airflows cause uncomfortable drafts and increase noise which and are not acceptable for comfort applications. Although these conditions can be mitigated by increasing the number of supply grilles or diffusers, additional equipment and space will be required. When these are not feasible, a DX or chilled water system should be considered. These systems are described elsewhere in this manual.

c. Evaporative Cooling.—In addition to the main units, plants require significant amounts of air to cool other loads including: transformers, switchgear, busses, panels, lights, auxiliary pump motors, and skin loads due to solar energy. When the outdoor air temperature, cooling loads, or both are too high for ventilation cooling, evaporative cooling should be considered before resorting to refrigeration cooling systems. Modern commercial evaporative coolers are significantly more effective (efficient) than those previously available. The new coolers are available in single stage (direct) or multistage (indirect/direct) configurations.

(1) Design considerations.

(a) *Media and saturation effectiveness.*—The performance capability of an evaporative cooler depends on the difference between the outdoor dry bulb temperature and the wet bulb temperature. This difference is commonly referred to as the wet bulb depression. The greater the wet bulb depression, the greater the cooling effect possible. The supply air temperature from an evaporative cooler is based on the "saturation effectiveness" of the evaporative cooling media that is usually published in manufacturer's data. The effectiveness is a measure of how close the leaving air dry bulb temperature approaches the wet bulb temperature of the air entering the media, or alternatively, how close the leaving air comes to the saturation temperature.

Although many types of evaporative cooler media are available, the most common type consists of a rigid, cross-fluted fiberglass or cellulose commonly referred to by the trade names Glas-dek and Cel-dek respectively. As shown in table 5, the saturation effectiveness of the media increases with the thickness and decreases with higher airflow velocities.

Table 5.—Evaporative cooler media saturation efficiency

Face velocity (fpm/sq. ft.)	Saturation efficiency — percent				
	Evaporative media thickness				
	6 inches	8 inches	12 inches	18 inches	24 inches
300	74.0%	84.0%	91.0%	98.5%	99.7%
350	72.0%	82.0%	90.0%	98.2%	99.6%
400	71.0%	81.0%	89.5%	98.0%	99.5%
450	69.0%	80.0%	89.0%	97.8%	99.3%
500	68.0%	79.0%	**88.8%**	97.5%	99.0%
550	67.0%	78.0%	88.5%	97.0%	98.9%
600	66.0%	77.0%	88.2%	96.8%	98.8%
650	65.0%	76.0%	88.0%	96.3%	98.7%
	Pressure drop — inches w.c.				
300	0.04	0.06	0.08	0.10	0.14
350	0.05	0.07	0.11	0.14	0.19
400	0.06	0.09	0.14	0.19	0.26
450	0.08	0.12	0.17	0.24	0.32
500	0.10	0.14	**0.21**	0.29	0.39
550	0.12	0.17	0.26	0.36	0.47
600	0.14	0.20	0.30	0.42	0.56
650	0.16	0.24	0.35	0.49	0.64

Note:
- Data based on CELdek® or GLASdek® evaporative media.
- 500 FPM is standard.

Media are available from 4 to 24-inches thick. The standard airflow velocity is 500 ft/min. The most common media thickness used on Reclamation plants is 12 inches; however, 8, 12, and 18 inch with saturation effectiveness of approximately 79, 88.5, and 97.8 per cent, respectively, at the 500 ft/min, have been used. Some have been designed with velocities ranging from 400 to 550 ft/min.

Once the saturation effectiveness, outdoor dry and wet bulb temperatures, and indoor design temperature are known, the required supply air temperature can be calculated or determined from the manufacturers published ratings. The following equation can be used to calculate the leaving air temperature:

$$T_{lvg} = T_{db} - e_s(T_{db} - T_{wb})$$

Where: T_{lvg} = air temperature leaving the media
T_{db} = outdoor dry bulb temperature entering the media
e_s = saturation effectiveness (from table 5)
T_{wb} = outdoor wet bulb temperature

Example: Assume outdoor design conditions of 109 ºF dry bulb, coincident wet bulb of 71 ºF, 500 ft/min airflow velocity, the leaving air temperature would be:

109 – (0.79)(109 – 71) = <u>79 ºF for 8-inch media</u>;

109 – (0.89)(109 – 71) = <u>75.4 ºF for 12-inch media</u>;

109 – (0.98)(109 – 71) = <u>71.8 ºF for 18-inch media</u>;

Since the fan and motor are normally installed in the airstream, the actual supply air temperature to the conditioned space would be slightly higher (2 to 3 ºF) depending on the supply fan and motor losses.

(b) *Freeze protection.*—Freeze protection is another major concern with evaporative cooling systems. This is especially true for unmanned plants in remote locations where freezing is possible. For these locations and conditions, the water circuit for the evaporative coolers should be designed for automatic draining. Automatic draining usually requires thermostatically controlled solenoid valves in the supply and drain piping. The valves open when a preset temperature, usually 35 to 40 ºF, is sensed. All piping and the cooling unit sumps and other auxiliary equipment such as strainers, pumps, etc. should be drained. Heat tape should be applied to all piping with low points that will not drain but are subject to freezing. Refer to section *F.10.f (2)* of this document for heat tape sizing procedures. Once fall outdoor temperatures drop to near freezing, a further need for evaporative cooling is highly unlikely for the remaining season. The system controls should lockout the water supply valve to preclude filling the water circuits and sumps.

(c) *Water quality and filtration.*—Water quality is very important to successful application of evaporative cooling equipment. Correct bleed-off rates and adequate makeup water must be available to prevent fouling due to increased mineral concentrations and deposits. In some cases chemical treatment may be required. To prevent plugging of nozzles and valves, water should pass through a strainer capable of filtering particles larger than 1/32-inch. Motor operated self-cleaning strainers are preferable to the manual backwash type.

(d) *Humidity*.—High relative humidity will adversely impact the effectiveness of these units. Most Reclamation plants are located in arid areas with low relative humidity. However, plants in Arizona can be adversely affected during the annual monsoon season when relative humidity is high. Under these conditions, evaporative cooling is not very effective and refrigerated cooling may be required. Plant humidity levels should be maintained between 30 and 60 percent to control growth of bacteria, viruses, fungi, and condensation on metal surfaces that are subject to corrosion. Room humidistat should be provided to de-energize the cooling unit water pump when the relative humidity approaches 60 percent.

(e) *Exhaust*.—Direct evaporative cooling systems should include good exhaust capability to prevent high humidity. HVAC systems are commonly designed to pressurize the conditioned spaces to stop outdoor air infiltration that increases the cooling load and room humidity. While this practice is acceptable in refrigerated systems, it has not always worked well in Reclamation plants using evaporative cooling systems. Perhaps the exhaust openings have been undersized or the humidity control was not operating properly. Exhaust air openings and equipment should be sized to prevent over pressurization. Use of exhaust fans instead of non-powered roof exhaust ventilators or exhaust louvers has been found to be more effective.

(2) ***Direct evaporative coolers.***—Direct evaporative coolers consist of an air handler section and direct evaporative cooling section. Direct cooling units have been used extensively in the Reclamation's Southern Nevada Pumping Plants. These units are operated in the fan only mode when the outdoor air temperature is low enough to meet the cooling load requirements. When ventilation cooling is inadequate, the water pump circuit is energized to provide evaporative cooling. The effectiveness of direct coolers may range between 80 and 99 percent.

(3) ***Indirect evaporative coolers.***—Indirect evaporative coolers cool air without adding moisture therefore they do not increase humidity. Indirect coolers include an air-to-air heat exchanger, evaporative media (such as the media described above), and a secondary (outdoor) air fan and outdoor air louvers. The secondary air is cooled as it flows through the evaporative media. The cooled air flows through a heat exchanger that indirectly cools the primary (supply) air to the space.

Another type of indirect cooler used on Reclamation plants incorporates cooling tower, which includes similar components except that the air-to-air heat exchanger is replaced with a water cooling coil. The cooling tower provides cooled water to a water cooling coil. Primary air flows through the cooling coil and is cooled without addition of moisture. The primary benefit of indirect coolers is their capability to cool without adding moisture, i.e. increasing the humidity of the primary air.

The effectiveness of indirect coolers may range between 50 and 75 percent.

(4) ***Indirect/direct coolers.***—When increased cooling without refrigeration is desired, direct and indirect coolers can be combined into multistage units referred to as indirect/direct cooling units. These units are typically staged to operate as follows: first stage – ventilation only; second stage – indirect cooling; and third stage – direct evaporative cooling.

Equipment manufacturers have published overall saturation effectiveness for these units ranging between 115 and 125 percent. Indirect/direct evaporative coolers can also be provided with refrigeration coils to provide cooling when the outdoor air humidity is too high for evaporative cooling.

d. **Refrigeration Cooling.**—Refrigeration cooling is used in plant cooling systems when: cooling loads are very high, greater humidity control is desired, or other cooling methods are not practical or acceptable. Mechanical equipment rooms must be provided to isolate the refrigeration equipment from other areas of the plant. These rooms have specific size, ventilation and fire code requirements that should be observed. Mechanical equipment rooms should comply with the requirements of the *Uniform Mechanical Code* (UMC) and (UBC).

(1) ***Plant chilled water cooling***.—The loads in large plant equipment are usually too high for DX equipment, therefore, chilled water systems with water cooled condensers are fairly common. Water chillers are usually the reciprocating type. A water temperature at least 5 ºF below the apparatus dew point temperature is required for an economical design. Plant chilled water systems are typically selected to supply 45 ºF water to air handler units (AHUs) and fan coil units (FCUs). A 10 ºF water temperature rise is normal for Reclamation work. The following may be used for very rough estimates in the preliminary stages of a design:

Airflow = 1 cfm per ft^2 of motor/generator room floor
Air supply temperature = 56 ºF
Chiller size = 60 BTUH per ft^2 of motor/generator room floor
Chiller Hp = 1.4 x No. of tons cooling
Condenser water = 3.2 gals/ton (assuming 7.5 ºF rise)
Chilled water = 2.4 gals/ton (assuming 10 ºF rise)

(2) ***Occupied spaces.***—Refrigeration cooling is normally reserved for occupied spaces such as control rooms, shops, and offices where human comfort is essential. Refrigeration air conditioning systems are also used for cooling communications rooms, electronic rooms, and computer rooms, where close temperature and humidity tolerances are necessary for proper equipment operation. The air conditioning systems for these spaces should be completely independent of the plant cooling systems where equipment can tolerate higher temperatures.

(a) *Minimum ventilation requirements.*—Minimum outdoor air ventilation requirements, as dictated by *ASHRAE Ventilation Standards*, must be provided at all times to ensure odor control. Use of economizers should be considered to conserve energy by providing free cooling when outdoor air conditions are favorable. In dry climates dry bulb controllers provide adequate control of the economizer. Where humidity is a concern, enthalpy controllers provide better control of the economizer and should be considered.

(b) *Air systems.*—The type of air system and refrigeration equipment used for air conditioning applications varies considerably. Direct expansion CV systems, such as packaged vertical or horizontal units, are common for small plants and spaces. Where spaces contain windows and loads vary throughout the day, due to solar affect, VAV should be considered. Humidification capability is seldom installed in these systems. The cooling system condensers may be air or water cooled.

(c) *Sizing criteria.*—The following criteria may be used for rough preliminary sizing of AC systems for small offices, lunch rooms, UPS, electric shops, and similar spaces:

Cooling = 40 BTUH/ft^2;
Heating = 38 BTUH/ft^2;
Air supply = 1.3 cfm/ft^2;
Supply air temperature = 56 ºF

(3) ***Computer rooms.***—Computer rooms require very close temperature and humidity controls usually achieved through dedicated cooling equipment specifically designed for computer room applications. True computer rooms are seldom occupied and require close temperature and humidity tolerances. Computer room AC systems should be designed to re-circulate air continuously. When these spaces are occupied, minimal outdoor air is required and is usually provided by tapping into another air conditioning unit providing air to general office spaces.

(a) *General system configuration.*—Computer room air conditioners are usually packaged vertical, CV, water cooled, DX units. These units are equipped with a disposable water canister to provide humidification or a waterline if clean water is available.

(b) *Sizing criteria.*—The following criteria may be used for preliminary sizing of computer rooms AC systems:

Cooling = 140 BTUH/ft^2;
Heating = 20 BTUH/ft^2;
Airflow = 4.7 cfm/ft^2;
Supply air temperature = 56 ºF.

9. **Ventilating Fans**.—Many types of ventilating fans are available, each well suited to a specific application. Figure 4 identifies the different types of fans available and provides a brief description of their performance characteristics and applications. All types of fans shown in Figure 4 have been used in Reclamation applications.

a. **Fan Types and Applications.—**Following is a list of the types of fans available and where they are used in Reclamation facilities:

(1) ***Centrifugal.***—Centrifugal fans are preferred for high volume, high pressure, ducted systems.

(a) *Airfoil and backwards inclined.*—Normally found in large (above 20,000 cfm) air handling units providing the main air supply to the plant.

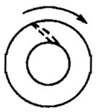
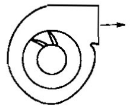
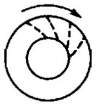
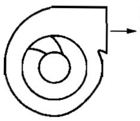
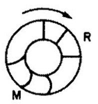
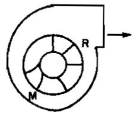
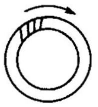
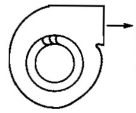
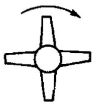
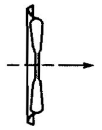
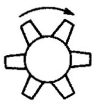
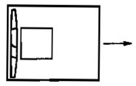
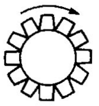
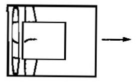
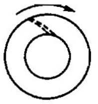
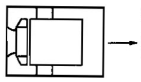
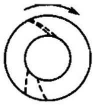
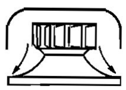
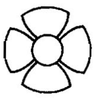
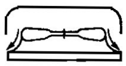
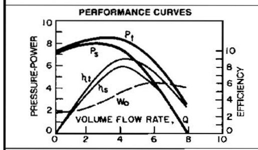
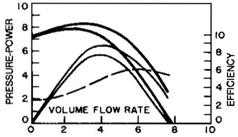
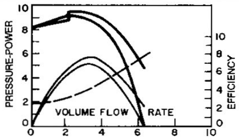
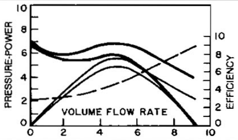
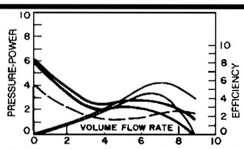
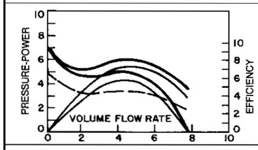
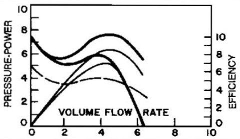
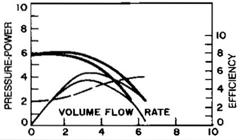
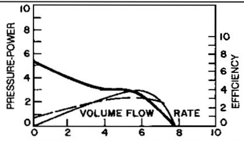
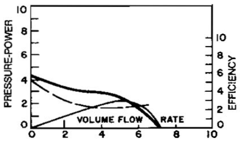
	TYPE		IMPELLER DESIGN		HOUSING DESIGN	
CENTRIFUGAL FANS	AIRFOIL			Highest efficiency of all centrifugal fan designs. Ten to 16 blades of airfoil contour curved away from direction of rotation. Deep blades allow efficient expansion within blade passages. Air leaves impeller at velocity less than tip speed. For given duty, has highest speed of centrifugal fan designs.		Scroll design for efficient conversion of velocity pressure to static pressure. Maximum efficiency requires close clearance and alignment between wheel and inlet.
	BACKWARD-INCLINED BACKWARD-CURVED			Efficiency only slightly less than airfoil fan. Ten to 16 single-thickness blades curved or inclined away from direction of rotation. Efficient for same reasons as airfoil fan.		Uses same housing configuration as airfoil design.
	RADIAL			Higher pressure characteristics than airfoil, backward-curved, and backward-inclined fans. Curve may have a break to left of peak pressure and fan should not be operated in this area. Power rises continually to free delivery.		Scroll. Usually narrowest of all centrifugal designs. Because wheel design is less efficient, housing dimensions are not as critical as for airfoil and backward-inclined fans.
	FORWARD-CURVED			Flatter pressure curve and lower efficiency than the airfoil, backward-curved, and backward-inclined. Do not rate fan in the pressure curve dip to the left of peak pressure. Power rises continually toward free delivery. Motor selection must take this into account.		Scroll similar to and often identical to other centrifugal fan designs. Fit between wheel and inlet not as critical as for airfoil and backward-inclined fans.
AXIAL FANS	PROPELLER			Low efficiency. Limited to low-pressure applications. Usually low-cost impellers have two or more blades of single thickness attached to relatively small hub. Primary energy transfer by velocity pressure.		Simple circular ring, orifice plate, or venturi. Optimum design is close to blade tips and forms smooth airfoil into wheel.
	TUBEAXIAL			Somewhat more efficient and capable of developing more useful static pressure than propeller fan. Usually has 4 to 8 blades with airfoil or single-thickness cross section. Hub is usually less than half the fan tip diameter.		Cylindrical tube with close clearance to blade tips.
	VANEAXIAL			Good blade design gives medium- to high-pressure capability at good efficiency. Most efficient have airfoil blades. Blades may have fixed, adjustable, or controllable pitch. Hub is usually greater than half fan tip diameter.		Cylindrical tube with close clearance to blade tips. Guide vanes upstream or downstream from impeller increase pressure capability and efficiency.
SPECIAL DESIGNS	TUBULAR CENTRIFUGAL			Performance similar to backward-curved fan except capacity and pressure are lower. Lower efficiency than backward-curved fan. Performance curve may have a dip to the left of peak pressure.		Cylindrical tube similar to vaneaxial fan, except clearance to wheel is not as close. Air discharges radially from wheel and turns 90° to flow through guide vanes.
	POWER ROOF VENTILATORS	CENTRIFUGAL		Low-pressure exhaust systems such as general factory, kitchen, warehouse, and some commercial installations. Provides positive exhaust ventilation, which is an advantage over gravity-type exhaust units. Centrifugal units are slightly quieter than axial units.		Normal housing not used, because air discharges from impeller in full circle. Usually does not include configuration to recover velocity pressure component.
		AXIAL		Low-pressure exhaust systems such as general factory, kitchen, warehouse, and some commercial installations. Provides positive exhaust ventilation, which is an advantage over gravity-type exhaust units.		Essentially a propeller fan mounted in a supporting structure. Hood protects fan from weather and acts as safety guard. Air discharges from annular space at bottom of weather hood.

Figure 4.—Types of fans.

	PERFORMANCE CURVES*	PERFORMANCE CHARACTERISTICS	APPLICATIONS
CENTRIFUGAL FANS		Highest efficiencies occur at 50 to 60% of wide-open volume. This volume also has good pressure characteristics. Power reaches maximum near peak efficiency and becomes lower, or self-limiting, toward free delivery.	General heating, ventilating, and air-conditioning applications. Usually only applied to large systems, which may be low-, medium-, or high-pressure applications. Applied to large, clean-air industrial operations for significant energy savings.
		Similar to airfoil fan, except peak efficiency slightly lower.	Same heating, ventilating, and air-conditioning applications as airfoil fan. Used in some industrial applications where environment may corrode or erode airfoil blade.
		Higher pressure characteristics than airfoil and backward-curved fans. Pressure may drop suddenly at left of peak pressure, but this usually causes no problems. Power rises continually to free delivery.	Primarily for materials handling in industrial plants. Also for some high-pressure industrial requirements. Rugged wheel is simple to repair in the field. Wheel sometimes coated with special material. Not common for HVAC applications.
		Pressure curve less steep than that of backward-curved fans. Curve dips to left of peak pressure. Highest efficiency to right of peak pressure at 40 to 50% of wide-open volume. Rate fan to right of peak pressure. Account for power curve, which rises continually toward free delivery, when selecting motor.	Primarily for low-pressure HVAC applications, such as residential furnaces, central station units, and packaged air conditioners.
AXIAL FANS		High flow rate, but very low-pressure capabilities. Maximum efficiency reached near free delivery. Discharge pattern circular and airstream swirls.	For low-pressure, high-volume air moving applications, such as air circulation in a space or ventilation through a wall without ductwork. Used for makeup air applications.
		High flow rate, medium-pressure capabilities. Performance curve dips to left of peak pressure. Avoid operating fan in this region. Discharge pattern circular and airstream rotates or swirls.	Low- and medium-pressure ducted HVAC applications where air distribution downstream is not critical. Used in some industrial applications, such as drying ovens, paint spray booths, and fume exhausts.
		High-pressure characteristics with medium-volume flow capabilities. Performance curve dips to left of peak pressure because of aerodynamic stall. Avoid operating fan in this region. Guide vanes correct circular motion imparted by wheel and improve pressure characteristics and efficiency of fan.	General HVAC systems in low-, medium-, and high-pressure applications where straight-through flow and compact installation are required. Has good downstream air distribution. Used in industrial applications in place of tubeaxial fans. More compact than centrifugal fans for same duty.
SPECIAL DESIGNS		Performance similar to backward-curved fan, except capacity and pressure is lower. Lower efficiency than backward-curved fan because air turns 90°. Performance curve of some designs is similar to axial flow fan and dips to left of peak pressure.	Primarily for low-pressure, return air systems in HVAC applications. Has straight-through flow.
		Usually operated without ductwork; therefore, operates at very low pressure and high volume. Only static pressure and static efficiency are shown for this fan.	Low-pressure exhaust systems, such as general factory, kitchen, warehouse, and some commercial installations. Low first cost and low operating cost give an advantage over gravity flow exhaust systems. Centrifugal units are somewhat quieter than axial flow units.
		Usually operated without ductwork; therefore, operates at very low pressure and high volume. Only static pressure and static efficiency are shown for this fan.	Low-pressure exhaust systems, such as general factory, kitchen, warehouse, and some commercial installations. Low first cost and low operating cost give an advantage over gravity-flow exhaust systems.

*These performance curves reflect general characteristics of various fans as commonly applied. They are not intended to provide complete selection criteria, because other parameters, such as diameter and speed, are not defined.

Figure 4a.—Types of fans.

(b) *Forward curve*.—The most commonly used fan for general ventilation and exhaust applications. Generally used in most types of packaged air handling units and equipment.

(c) *Radial*.—Primarily used in particulates exhaust applications to prevent clogging. The most common Reclamation applications include: wood shaving, saw dust, and sand.

(d) *In-line centrifugal*.—Primarily used in applications where sufficient floor space is not available and the fan must fit directly in the duct system. The fans are available in a square or round configuration. These fans are not commonly used in Reclamation applications.

(2) ***Axial***.—Axial fans are preferred for high volume, low pressure, non-ducted systems.

(a) *Propeller*.—Propeller fans are intended for high flow, very low pressure applications where ductwork is not required. Maximum efficiency reached near free delivery, i.e. with no suction and discharge ductwork. These fans are very common in Reclamation plants. Primarily used to improve air circulation within rooms or galleries, to transfer air between rooms, and to exhaust air directly outdoors. Roof exhaust assemblies consisting of a propeller fan, exhaust hoods, control damper, and thermostat are commonly used to provide passive and powered roof exhaust in plants.

(b) *Tubeaxial*.—High flow medium pressure capability. Pressure curve dips to the left of peak pressure. Avoid operation in this area. Fan may be used in low to medium pressure applications with ductwork. Common in Reclamation work and primarily used for air transfer applications between floors and where floor space is limited and the fan must be installed in-line with the ductwork. Operating range is 65 to 90 percent of wide open cfm.

(c) *Vaneaxial*.—Medium flow, high pressure capability. Pressure curve dips to the left of peak pressure. Avoid operation in this area. Blades may have fixed, adjustable or controllable pitch. Guide vanes are provided to straighten airflow and improve efficiency. Fan may be used in low, medium and high pressure applications with ductwork where straight-through flow and compact design are required. Common in Reclamation work and primarily used for air transfer applications between floors and where floor space is limited and the fan must be installed in-line with the ductwork. Maximum static efficiency is higher than the tubeaxial fan and is approximately 65 percent.

(3) ***Powered roof ventilators***.—Fans are available in centrifugal or axial configurations. The centrifugal design is slightly quieter. These fans are available in supply and exhaust and down blast and up blast models. Commonly used for general low pressure, non-ducted applications where positive exhaust is necessary or desirable. The down blast models are used for most exhaust application where a packaged low silhouette roof mounting is desirable. Wall exhaust applications are also common. The up blast model direct exhaust air away from the roof surface and are used for exhausting toxic chemicals or fumes.

b. **Fan Sizing**.—The steps followed when sizing a fan are commonly available in HVAC handbooks and texts and will not be discussed. The reader is referred to the *ASHRAE Handbook of Fundamentals*. The *ASHRAE Handbook* tends to be heavily weighted towards the theoretical and less towards the practical; however, it is revised every four years, so the information more closely reflects new research and industry trends. The *SMACNA HVAC System Duct Design* tends to be more practical and user friendly. The following discussion pertains to common mistakes made by inexperienced personnel when designing duct systems. These mistakes have significant effects on the performance of the fan and ventilating or exhaust system.

(1) ***Density (altitude and temperature) correction.***—The data presented in fan manufacturers performance tables or curve are based on sea level altitude and 70 ºF. Before using the fan manufacturers tables or curves for selecting a fan, the criteria at the required altitude and temperature must be corrected to sea level conditions. Motor cooling is also affected by air density and decreases with increasing altitude. The following recommendations are offered:

(a) *Fan corrections.*—For altitudes changes between 0 and 2000 ft above sea level, and temperatures between 32 and 120 ºF, no corrections are necessary for fans used in normal Reclamation applications. See table 6 for applicable altitude and temperature corrections.

(b) *Motor corrections*.—For altitudes below 3,000 ft, motor operation should not be adversely affected. Above 3000 ft motor cooling may be affected and the motor manufacturer should be consulted. In many cases the motor induction rating may have to be increased.

(2) ***System effect***.—Manufacturers performance tables or curves for a fan are based on testing conducted with ductwork to produce uniform entrance and exit conditions at the fan. The tables represent the best performance possible under controlled laboratory conditions. However, field installations may vary significantly from laboratory conditions. Frequently, either through inexperience or space limitations, the ductwork connections to a fan are less than ideal and contribute to poor fan performance. The most common occurrence adversely affecting fan performance is installing elbows too close to the fan suction and discharge openings. Either of these conditions will produce non-uniform flow causing pressure losses exceeding those attributed to the elbows alone. This excess pressure loss is commonly referred to as a “system effect loss.” System effect cannot be measured in the field. The designer must be aware of the conditions causing the system effect losses and eliminate the condition when possible. If the condition cannot be eliminated, calculate the system effect losses and add them to the losses attributed to other ductwork and system components. Figures 5 through 14 illustrate various conditions causing system effects and how to calculate the resulting pressure losses.

Table 6a.—Altitude correction factors

Altitude (feet)	Barometer		Specific volume (ft^3/lb)	Relative density, static pressure, or Hp correction factor	Air Density (lbs/ft^3)	CFM transmission factor	CFM correction factor
	Inches mercury	lbs/in^2 atmospheric					
0	29.92	14.70	13.340	1.000	0.0750	1.080	1.000
100	29.81	14.64	13.381	0.996	0.0747	1.076	1.003
200	29.71	14.59	13.430	0.993	0.0745	1.072	1.007
300	29.60	14.54	13.478	0.989	0.0742	1.068	1.010
400	29.49	14.48	13.527	0.986	0.0739	1.065	1.014
500	29.38	14.43	13.576	0.982	0.0737	1.061	1.018
600	29.28	14.38	13.626	0.979	0.0734	1.057	1.021
700	29.17	14.33	13.675	0.975	0.0731	1.053	1.025
800	29.07	14.28	13.725	0.971	0.0729	1.049	1.029
900	28.96	14.22	13.775	0.968	0.0726	1.045	1.033
1000	28.86	14.17	13.825	0.964	0.0723	1.042	1.036
1100	28.75	14.12	13.876	0.961	0.0721	1.038	1.040
1200	28.65	14.07	13.926	0.957	0.0718	1.034	1.044
1300	28.54	14.02	13.977	0.954	0.0715	1.030	1.048
1400	28.44	13.97	14.028	0.950	0.0713	1.027	1.052
1500	28.33	13.92	14.079	0.947	0.0710	1.023	1.055
1600	28.23	13.87	14.131	0.944	0.0708	1.019	1.059
1700	28.13	13.82	14.183	0.940	0.0705	1.015	1.063
1800	28.03	13.76	14.235	0.937	0.0703	1.012	1.067
1900	27.92	13.71	14.287	0.933	0.0700	1.008	1.071
2000	27.82	13.66	14.339	0.930	0.0697	1.004	1.075
2100	27.72	13.61	14.392	0.926	0.0695	1.001	1.079
2200	27.62	13.56	14.445	0.923	0.0692	0.997	1.083
2300	27.52	13.51	14.498	0.920	0.0690	0.993	1.087
2400	27.42	13.47	14.551	0.916	0.0687	0.990	1.091
2500	27.31	13.42	14.605	0.913	0.0685	0.986	1.095
2600	27.21	13.37	14.659	0.910	0.0682	0.982	1.099
2700	27.11	13.32	14.713	0.906	0.0680	0.979	1.103
2800	27.02	13.27	14.767	0.903	0.0677	0.975	1.107
2900	26.92	13.22	14.822	0.900	0.0675	0.972	1.111

Table 6a.—Altitude correction factors—continued

Altitude (feet)	Barometer		Specific volume (ft^3/lb)	Relative density, static pressure, or Hp correction factor	Air density (lbs/ft^3)	CFM transmission factor	CFM correction factor
	Inches mercury	lbs/in^2 atmospheric					
3000	26.82	13.17	14.876	0.896	0.0672	0.968	1.115
3100	26.72	13.12	14.931	0.893	0.0670	0.964	1.119
3200	26.62	13.07	14.987	0.890	0.0667	0.961	1.123
3300	26.52	13.03	15.042	0.886	0.0665	0.957	1.128
3400	26.42	12.98	15.098	0.883	0.0662	0.954	1.132
3500	26.33	12.93	15.154	0.880	0.0660	0.950	1.136
3600	26.23	12.88	15.210	0.877	0.0657	0.947	1.140
3700	26.13	12.83	15.267	0.873	0.0655	0.943	1.144
3800	26.03	12.79	15.323	0.870	0.0653	0.940	1.149
3900	25.94	12.74	15.380	0.867	0.0650	0.936	1.153
4000	25.84	12.69	15.438	0.864	0.0648	0.933	1.157
4100	25.75	12.65	15.495	0.860	0.0645	0.929	1.162
4200	25.65	12.60	15.553	0.857	0.0643	0.926	1.166
4300	25.56	12.55	15.611	0.854	0.0641	0.922	1.170
4400	25.46	12.50	15.669	0.851	0.0638	0.919	1.175
4500	25.37	12.46	15.728	0.848	0.0636	0.916	1.179
4600	25.27	12.41	15.786	0.845	0.0633	0.912	1.183
4700	25.18	12.37	15.845	0.841	0.0631	0.909	1.188
4800	25.08	12.32	15.905	0.838	0.0629	0.905	1.192
4900	24.99	12.27	15.964	0.835	0.0626	0.902	1.197
5000	24.90	12.23	16.024	0.832	0.0624	0.899	1.201
5100	24.80	12.18	16.084	0.829	0.0622	0.895	1.206
5200	24.71	12.14	16.145	0.826	0.0619	0.892	1.210
5300	24.62	12.09	16.205	0.823	0.0617	0.889	1.215
5400	24.53	12.05	16.266	0.820	0.0615	0.885	1.219
5500	24.43	12.00	16.327	0.817	0.0612	0.882	1.224
5600	24.34	11.96	16.389	0.814	0.0610	0.879	1.229
5700	24.25	11.91	16.450	0.811	0.0608	0.875	1.233
5800	24.16	11.87	16.512	0.807	0.0606	0.872	1.238

Table 6a.—Altitude correction factors—continued

Altitude (feet)	Barometer		Specific volume (ft^3/lb)	Relative density, static pressure, or Hp correction factor	Air density (lbs/ft^3)	CFM transmission factor	CFM correction factor
	Inches mercury	lbs/in^2 atmospheric					
5900	24.07	11.82	16.575	0.804	0.0603	0.869	1.242
6000	23.98	11.78	16.637	0.801	0.0601	0.866	1.247
6100	23.89	11.73	16.700	0.798	0.0599	0.862	1.252
6200	23.80	11.69	16.763	0.795	0.0597	0.859	1.257
6300	23.71	11.64	16.827	0.792	0.0594	0.856	1.261
6400	23.62	11.60	16.890	0.789	0.0592	0.853	1.266
6500	23.53	11.56	16.954	0.786	0.0590	0.849	1.271
6600	23.44	11.51	17.019	0.783	0.0588	0.846	1.276
6700	23.35	11.47	17.083	0.780	0.0585	0.843	1.281
6800	23.26	11.43	17.148	0.778	0.0583	0.840	1.285
6900	23.18	11.38	17.213	0.775	0.0581	0.837	1.290
7000	23.09	11.34	17.279	0.772	0.0579	0.833	1.295
7100	23.00	11.30	17.344	0.769	0.0577	0.830	1.300
7200	22.91	11.25	17.410	0.766	0.0574	0.827	1.305
7300	22.83	11.21	17.477	0.763	0.0572	0.824	1.310
7400	22.74	11.17	17.543	0.760	0.0570	0.821	1.315
7500	22.65	11.13	17.610	0.757	0.0568	0.818	1.320
7600	22.57	11.08	17.678	0.754	0.0566	0.815	1.325
7700	22.48	11.04	17.745	0.751	0.0564	0.811	1.330
7800	22.40	11.00	17.813	0.749	0.0561	0.808	1.335
7900	22.31	10.96	17.881	0.746	0.0559	0.805	1.340
8000	22.23	10.92	17.950	0.743	0.0557	0.802	1.346
8100	22.14	10.87	18.019	0.740	0.0555	0.799	1.351
8200	22.06	10.83	18.088	0.737	0.0553	0.796	1.356
8300	21.97	10.79	18.157	0.734	0.0551	0.793	1.361
8400	21.89	10.75	18.227	0.732	0.0549	0.790	1.366
8500	21.80	10.71	18.297	0.729	0.0547	0.787	1.372
8600	21.72	10.67	18.367	0.726	0.0544	0.784	1.377
8700	21.64	10.63	18.438	0.723	0.0542	0.781	1.382

Table 6a.—Altitude correction factors—continued

Altitude (feet)	Barometer		Specific volume (ft^3/lb)	Relative density, static pressure, or Hp correction factor	Air density (lbs/ft^3)	CFM transmission factor	CFM correction factor
	Inches mercury	lbs/in^2 atmospheric					
8800	21.55	10.59	18.509	0.720	0.0540	0.778	1.387
8900	21.47	10.55	18.580	0.718	0.0538	0.775	1.393
9000	21.39	10.50	18.652	0.715	0.0536	0.772	1.398
9100	21.31	10.46	18.724	0.712	0.0534	0.769	1.404
9200	21.22	10.42	18.796	0.709	0.0532	0.766	1.409
9300	21.14	10.38	18.869	0.707	0.0530	0.763	1.414
9400	21.06	10.34	18.942	0.704	0.0528	0.760	1.420
9500	20.98	10.30	19.015	0.701	0.0526	0.757	1.425
9600	20.90	10.26	19.089	0.698	0.0524	0.754	1.431
9700	20.82	10.22	19.163	0.696	0.0522	0.751	1.437
9800	20.74	10.19	19.237	0.693	0.0520	0.749	1.442
9900	20.66	10.15	19.312	0.690	0.0518	0.746	1.448
10,000	20.58	10.11	19.387	0.688	0.0516	0.743	1.453
10,100	20.50	10.07	19.463	0.685	0.0514	0.740	1.459
10,200	20.42	10.03	19.538	0.682	0.0512	0.737	1.465
10,300	20.34	9.99	19.615	0.680	0.0510	0.734	1.470
10,400	20.26	9.95	19.691	0.677	0.0508	0.731	1.476
10,500	20.18	9.91	19.768	0.674	0.0506	0.728	1.482
10,600	20.10	9.87	19.845	0.672	0.0504	0.726	1.488
10,700	20.02	9.84	19.922	0.669	0.0502	0.723	1.493
10,800	19.95	9.80	20.000	0.667	0.0500	0.720	1.499
10,900	19.87	9.76	20.079	0.664	0.0498	0.717	1.505
11,000	19.79	9.72	20.157	0.661	0.0496	0.714	1.511
11,100	19.71	9.68	20.236	0.659	0.0494	0.712	1.517
11,200	19.64	9.64	20.316	0.656	0.0492	0.709	1.523
11,300	19.56	9.61	20.395	0.654	0.0490	0.706	1.529
11,400	19.48	9.57	20.475	0.651	0.0488	0.703	1.535
11,500	19.41	9.53	20.556	0.649	0.0486	0.701	1.541
11,600	19.33	9.49	20.637	0.646	0.0485	0.698	1.547

Table 6a.—Altitude correction factors—continued

Altitude (feet)	Barometer		Specific volume (ft^3/lb)	Relative density, static pressure, or Hp correction factor	Air density (lbs/ft^3)	CFM transmission factor	CFM correction factor
	Inches mercury	lbs/in^2 atmospheric					
11,700	19.26	9.46	20.718	0.644	0.0483	0.695	1.553
11,800	19.18	9.42	20.800	0.641	0.0481	0.692	1.559
11,900	19.10	9.38	20.882	0.639	0.0479	0.690	1.565
12,000	19.03	9.35	20.964	0.636	0.0477	0.687	1.572
12,100	18.95	9.31	21.047	0.634	0.0475	0.684	1.578
12,200	18.88	9.27	21.130	0.631	0.0473	0.682	1.584
12,300	18.81	9.24	21.213	0.629	0.0471	0.679	1.590
12,400	18.73	9.20	21.297	0.626	0.0470	0.676	1.596
12,500	18.66	9.16	21.382	0.624	0.0468	0.673	1.603
12,600	18.58	9.13	21.466	0.621	0.0466	0.671	1.609
12,700	18.51	9.09	21.551	0.619	0.0464	0.668	1.616
12,800	18.44	9.06	21.637	0.616	0.0462	0.666	1.622
12,900	18.36	9.02	21.723	0.614	0.0460	0.663	1.628
13,000	18.29	8.98	21.809	0.611	0.0459	0.660	1.635
13,100	18.22	8.95	21.896	0.609	0.0457	0.658	1.641
13,200	18.15	8.91	21.983	0.607	0.0455	0.655	1.648
13,300	18.08	8.88	22.071	0.604	0.0453	0.652	1.654
13,400	18.00	8.84	22.159	0.602	0.0451	0.650	1.661
13,500	17.93	8.81	22.247	0.599	0.0449	0.647	1.668
13,600	17.86	8.77	22.336	0.597	0.0448	0.645	1.674
13,700	17.79	8.74	22.425	0.595	0.0446	0.642	1.681
13,800	17.72	8.70	22.515	0.592	0.0444	0.640	1.688
13,900	17.65	8.67	22.605	0.590	0.0442	0.637	1.695
14,000	17.58	8.63	22.696	0.587	0.0441	0.634	1.701
14,100	17.51	8.60	22.787	0.585	0.0439	0.632	1.708
14,200	17.44	8.56	22.878	0.583	0.0437	0.629	1.715
14,300	17.37	8.53	22.970	0.580	0.0435	0.627	1.722
14,400	17.30	8.50	23.062	0.578	0.0434	0.624	1.729
14,500	17.23	8.46	23.155	0.576	0.0432	0.622	1.736

Table 6a.—Altitude correction factors—continued

Altitude (feet)	Barometer		Specific volume (ft^3/lb)	Relative density, static pressure, or Hp correction factor	Air density (lbs/ft^3)	CFM transmission factor	CFM correction factor
	Inches mercury	lbs/in^2 atmospheric					
14,600	17.16	8.43	23.248	0.574	0.0430	0.619	1.743
14,700	17.09	8.39	23.341	0.571	0.0428	0.617	1.750
14,800	17.02	8.36	23.436	0.569	0.0427	0.614	1.757
14,900	16.95	8.33	23.530	0.567	0.0425	0.612	1.764
15,000	16.89	8.29	23.625	0.564	0.0423	0.610	1.771

Notes:

Altitude	Feet above sea level (s.l.)
Barometer	29.92 in. Hg or 14.7 lbs/in^2 = 1 atmosphere at s.l. @ 70ºF i.e. : 1 in. Hg = 0.491 lbs/in^2 Formula: in. Hg @ given alt. = $29.921(1-(0.00000688)(\text{alt.}))^{5.2559}$
Specific Volume	Standard Air @ s.l. @ 70 ºF has Specific Volume = 13.34 ft^3/lb. Formula for how Specific Volume varies with change in alt.: Specific Volume @ given alt. = 13.34 x (Relative Density @ s.l. ÷ Relative Density @ given alt.)
Relative Density	Ratio of the Density of Air @ given alt. @ 70 ºF to the Density of Air @ s.l. @ 70 ºF (= 0.075 lbs/ft^3)
Static Pressure Correction Factor	Static Pressure varies as the density (or specific volume) changes For constant cfm and rpm: $P_2/P_1 = V_1/V_2$, where: P_1 = Static Pressure @ given alt. P_2 = Static Pressure @ s.l. V_1 = Specific Volume @ given alt. V_2 = Specific Volume @ s.l. Example: Determine Static Pressure used for sizing fan (at s.l.) if calculated Static Pressure requirement is 0.82 at 1800 ft P_2 = 0.82 in. w.g. x 14.235/13.34 = 0.875 in. w.g. @ s.l.
Horsepower Correction Factor	Brake Horsepower varies as Relative Density @ constant cfm and rpm $Hp_2/Hp_1 = V_1/V_2$, where: Hp_1 = Brake Hp @ given alt. Hp_2 = Brake Hp @ s.l. V_1 = Specific Volume @ given Alt. V_2 = Specific Volume @ s.l. Similarly, $Hp_1 = Hp_2$ x Relative Density @ given alt.
Air Density	Reciprocal of Specific Volume Formula: Air Density @ given alt. = Relative Density @ given alt. x Air Density @ s.l. (0.075)
CFM Transmission Factor	Heat capacity of air. Formula: CFM Trans. Factor = Air Density @ given alt. @ 70 ºF x 60 minutes x 0.24 (specific heat of air)
CFM Correction Factor	Obtained from $Q_2/Q_1 = V_1/V_2$, where: Q_2 = cfm @ s.l. and 70 ºF Q_1 = cfm @ given alt. and temperature V_2 = Specific Volume @ s.l. and 70 ºF V_1 = Specific Volume @ given alt. and temperature This factor is used in the formula Q1 = Q2 (f1 x f2), where: Q_2 = cfm @ s.l. and 70 ºF Q_1 = cfm @ given alt. and temperature f_1 = CFM correction factor f_2 = Temperature correction factor (see Table 6b)
Fan or Blower RPM	Change in RPM is inversely proportional to the square root of the Density of Air

Table 6b.—Temperature/Density Correction Factors
Relative Density Correction Factor = (529.67)/(T+459.67)

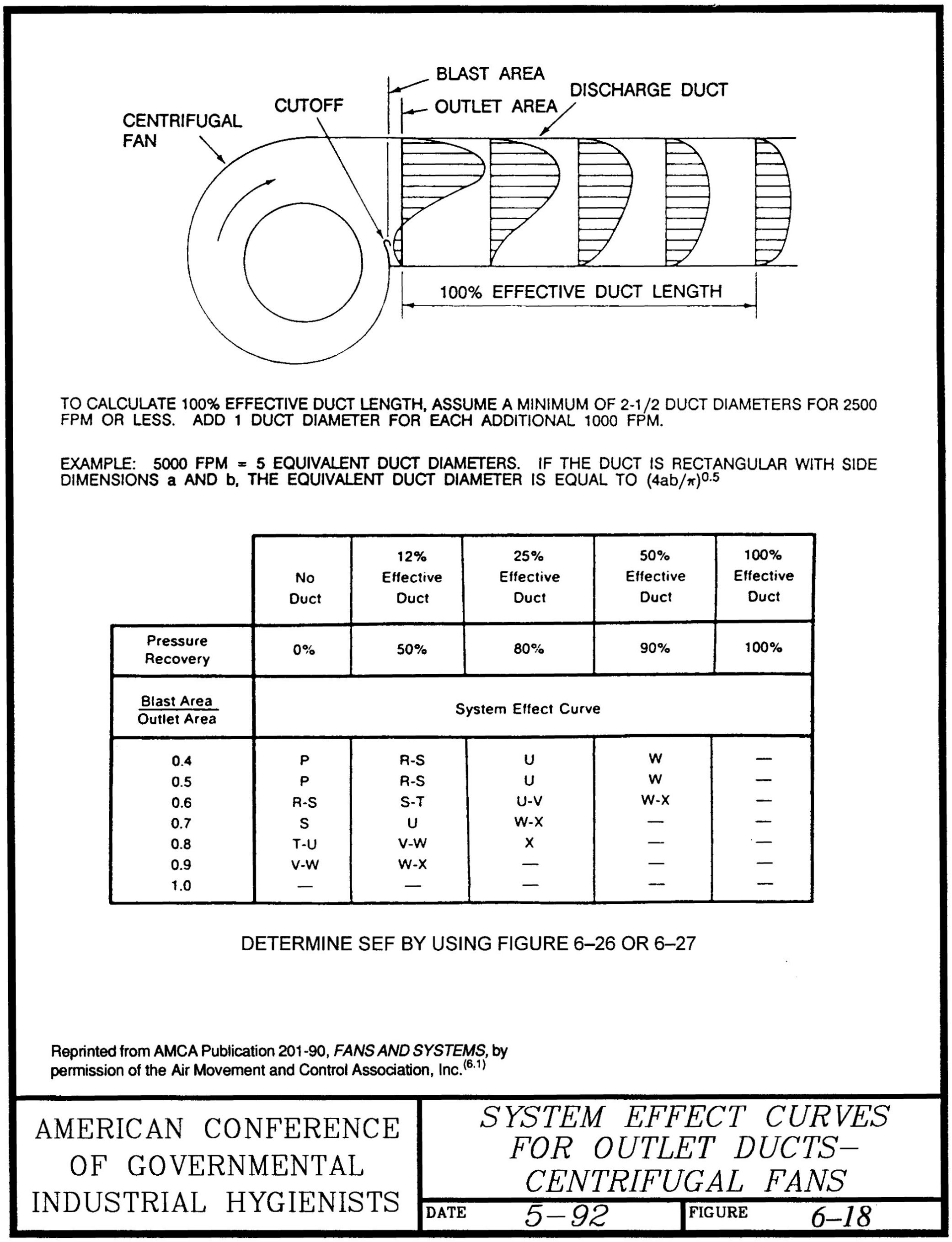
Air Temp °F	Relative Density Correction Factor	Air Temp °F	Relative Density Correction Factor	Air Temp °F	Relative Density Correction Factor	Air Temp °F	Relative Density Correction Factor
-40	1.262	48	1.043	136	0.889	224	0.775
-38	1.256	50	1.039	138	0.886	226	0.772
-36	1.250	52	1.035	140	0.883	228	0.770
-34	1.244	54	1.031	142	0.880	230	0.768
-32	1.239	56	1.027	144	0.877	232	0.766
-30	1.233	58	1.023	146	0.875	234	0.764
-28	1.227	60	1.019	148	0.872	236	0.761
-26	1.221	62	1.015	150	0.869	238	0.759
-24	1.216	64	1.011	152	0.866	240	0.757
-22	1.210	66	1.008	154	0.863	242	0.755
-20	1.205	68	1.004	156	0.860	244	0.753
-18	1.199	70	1.000	158	0.858	246	0.751
-16	1.194	72	0.996	160	0.855	270	0.726
-14	1.188	74	0.993	162	0.852	272	0.724
-12	1.183	76	0.989	164	0.849	274	0.722
-10	1.178	78	0.985	166	0.847	276	0.720
-8	1.173	80	0.981	168	0.844	278	0.718
-6	1.168	82	0.978	170	0.841	280	0.716
-4	1.162	84	0.974	172	0.839	282	0.714
-2	1.157	86	0.971	174	0.836	284	0.712
0	1.152	88	0.967	176	0.833	286	0.710
2	1.147	90	0.964	178	0.831	288	0.708
4	1.142	92	0.960	180	0.828	290	0.707
6	1.137	94	0.957	182	0.825	292	0.705
8	1.133	96	0.953	184	0.823	294	0.703
10	1.128	98	0.950	186	0.820	296	0.701
12	1.123	100	0.946	188	0.818	298	0.699
14	1.118	102	0.943	190	0.815	300	0.697
16	1.114	104	0.940	192	0.813	302	0.695
18	1.109	106	0.936	194	0.810	304	0.694
20	1.104	108	0.933	196	0.808	306	0.692
22	1.100	110	0.930	198	0.805	308	0.690
24	1.095	112	0.927	200	0.803	310	0.688
26	1.091	114	0.923	202	0.801	312	0.686
28	1.086	116	0.920	204	0.798	314	0.685
30	1.082	118	0.917	206	0.796	316	0.683
32	1.077	120	0.914	208	0.793	318	0.681
34	1.073	122	0.911	210	0.791	320	0.679
36	1.069	124	0.907	212	0.789	322	0.678
38	1.064	126	0.904	214	0.786	324	0.676
40	1.060	128	0.901	216	0.784	326	0.674
42	1.056	130	0.898	218	0.782	328	0.672
44	1.052	132	0.895	220	0.779	330	0.671
46	1.047	134	0.892	222	0.777	332	0.669

Table 6b.—Temperature/Density Correction Factors—continued
Relative Density Correction Factor = (529.67)/(T+459.67)

Air Temp °F	Relative Density Correction Factor	Air Temp °F	Relative Density Correction Factor	Air Temp °F	Relative Density Correction Factor	Air Temp °F	Relative Density Correction Factor
334	0.667	428	0.597	522	0.540	616	0.492
336	0.666	430	0.595	524	0.538	618	0.491
338	0.664	432	0.594	526	0.537	620	0.491
340	0.662	434	0.593	528	0.536	622	0.490
342	0.661	436	0.591	530	0.535	624	0.489
344	0.659	438	0.590	532	0.534	626	0.488
346	0.657	440	0.589	534	0.533	628	0.487
348	0.656	442	0.587	536	0.532	630	0.486
350	0.654	444	0.586	538	0.531	632	0.485
352	0.653	446	0.585	540	0.530	634	0.484
354	0.651	448	0.584	542	0.529	636	0.483
356	0.649	450	0.582	544	0.528	638	0.483
358	0.648	452	0.581	546	0.527	640	0.482
360	0.646	454	0.580	548	0.526	642	0.481
362	0.645	456	0.578	550	0.525	644	0.480
364	0.643	458	0.577	552	0.524	646	0.479
366	0.642	460	0.576	554	0.523	648	0.478
368	0.640	462	0.575	556	0.521	650	0.477
370	0.638	464	0.573	558	0.520	652	0.476
372	0.637	466	0.572	560	0.519	654	0.476
374	0.635	468	0.571	562	0.518	656	0.475
376	0.634	470	0.570	564	0.517	658	0.474
378	0.632	472	0.569	566	0.516	660	0.473
380	0.631	474	0.567	568	0.515	662	0.472
382	0.629	476	0.566	570	0.514	664	0.471
384	0.628	478	0.565	572	0.513	666	0.471
386	0.626	480	0.564	574	0.512	668	0.470
388	0.625	482	0.562	576	0.511	670	0.469
390	0.623	484	0.561	578	0.510	672	0.468
392	0.622	486	0.560	580	0.509	674	0.467
394	0.620	488	0.559	582	0.508	676	0.466
396	0.619	490	0.558	584	0.508	678	0.466
398	0.618	492	0.557	586	0.507	680	0.465
400	0.616	494	0.555	588	0.506	682	0.464
402	0.615	496	0.554	590	0.505	684	0.463
404	0.613	498	0.553	592	0.504	686	0.462
406	0.612	500	0.552	594	0.503	688	0.462
408	0.610	502	0.551	596	0.502	690	0.461
410	0.609	504	0.550	598	0.501	692	0.460
412	0.608	506	0.549	600	0.500	694	0.459
414	0.606	508	0.547	602	0.499	696	0.458
416	0.605	510	0.546	604	0.498	698	0.458
418	0.603	512	0.545	606	0.497	700	0.457
420	0.602	514	0.544	608	0.496	702	0.456
422	0.601	516	0.543	610	0.495	704	0.455
424	0.599	518	0.542	612	0.494	706	0.454
426	0.598	520	0.541	614	0.493	708	0.454

Table 6b.—Temperature/Density Correction Factors—continued
Relative Density Correction Factor = (529.67)/(T+459.67)

Air Temp °F	Relative Density Correction Factor	Air Temp °F	Relative Density Correction Factor	Air Temp °F	Relative Density Correction Factor	Air Temp °F	Relative Density Correction Factor
710	0.453	804	0.419	898	0.390	992	0.365
712	0.452	806	0.418	900	0.390	994	0.364
714	0.451	808	0.418	902	0.389	996	0.364
716	0.451	810	0.417	904	0.388	998	0.363
718	0.450	812	0.417	906	0.388	1000	0.363
720	0.449	814	0.416	908	0.387	–	–
722	0.448	816	0.415	910	0.387	–	–
724	0.447	818	0.415	912	0.386	–	–
726	0.447	820	0.414	914	0.386	–	–
728	0.446	822	0.413	916	0.385	–	–
730	0.445	824	0.413	918	0.384	–	–
732	0.444	826	0.412	920	0.384	–	–
734	0.444	828	0.411	922	0.383	–	–
736	0.443	830	0.411	924	0.383	–	–
738	0.442	832	0.410	926	0.382	–	–
740	0.442	834	0.409	928	0.382	–	–
742	0.441	836	0.409	930	0.381	–	–
744	0.440	838	0.408	932	0.381	–	–
746	0.439	840	0.408	934	0.380	–	–
748	0.439	842	0.407	936	0.380	–	–
750	0.438	844	0.406	938	0.379	–	–
752	0.437	846	0.406	940	0.378	–	–
754	0.436	848	0.405	942	0.378	–	–
756	0.436	850	0.404	944	0.377	–	–
758	0.435	852	0.404	946	0.377	–	–
760	0.434	854	0.403	948	0.376	–	–
762	0.434	856	0.403	950	0.376	–	–
764	0.433	858	0.402	952	0.375	–	–
766	0.432	860	0.401	954	0.375	–	–
768	0.431	862	0.401	956	0.374	–	–
770	0.431	864	0.400	958	0.374	–	–
772	0.430	866	0.400	960	0.373	–	–
774	0.429	868	0.399	962	0.373	–	–
776	0.429	870	0.398	964	0.372	–	–
778	0.428	872	0.398	966	0.372	–	–
780	0.427	874	0.397	968	0.371	–	–
782	0.427	876	0.397	970	0.370	–	–
784	0.426	878	0.396	972	0.370	–	–
786	0.425	880	0.395	974	0.369	–	–
788	0.425	882	0.395	976	0.369	–	–
790	0.424	884	0.394	978	0.368	–	–
792	0.423	886	0.394	980	0.368	–	–
794	0.422	888	0.393	982	0.367	–	–
796	0.422	890	0.392	984	0.367	–	–
798	0.421	892	0.392	986	0.366	–	–
800	0.420	894	0.391	988	0.366	–	–
802	0.420	896	0.391	990	0.365	–	–



TO CALCULATE 100% EFFECTIVE DUCT LENGTH, ASSUME A MINIMUM OF 2-1/2 DUCT DIAMETERS FOR 2500 FPM OR LESS. ADD 1 DUCT DIAMETER FOR EACH ADDITIONAL 1000 FPM.

EXAMPLE: 5000 FPM = 5 EQUIVALENT DUCT DIAMETERS. IF THE DUCT IS RECTANGULAR WITH SIDE DIMENSIONS a AND b, THE EQUIVALENT DUCT DIAMETER IS EQUAL TO $(4ab/\pi)^{0.5}$

	No Duct	12% Effective Duct	25% Effective Duct	50% Effective Duct	100% Effective Duct
Pressure Recovery	0%	50%	80%	90%	100%
Blast Area / Outlet Area	System Effect Curve				
0.4	P	R-S	U	W	—
0.5	P	R-S	U	W	—
0.6	R-S	S-T	U-V	W-X	—
0.7	S	U	W-X	—	—
0.8	T-U	V-W	X	—	—
0.9	V-W	W-X	—	—	—
1.0	—	—	—	—	—

DETERMINE SEF BY USING FIGURE 6–26 OR 6–27

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AMERICAN CONFERENCE
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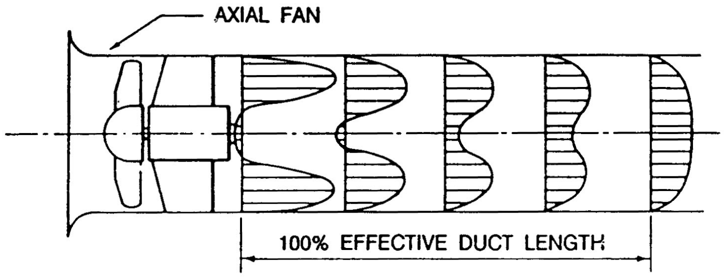
INDUSTRIAL HYGIENISTS

SYSTEM EFFECT CURVES
FOR OUTLET DUCTS–
CENTRIFUGAL FANS

DATE *5–92* FIGURE *6–18*

Figure 5.—System effect curves for outlet ducts—centrifugal fans.



TO CALCULATE 100% EFFECTIVE DUCT LENGTH, ASSUME A MINIMUM OF 2-1/2 DUCT DIAMETERS FOR 2500 FPM OR LESS. ADD 1 DUCT DIAMETER FOR EACH ADDITIONAL 1000 FPM.

EXAMPLE: 5000 FPM = 5 EQUIVALENT DUCT DIAMETERS

	No Duct	12% Effective Duct	25% Effective Duct	50% Effective Duct	100% Effective Duct
Tubeaxial Fan	—	—	—	—	—
Vaneaxial Fan	U	V	W	—	—

DETERMINE SEF BY USING FIGURE 6–26 OR 6–27

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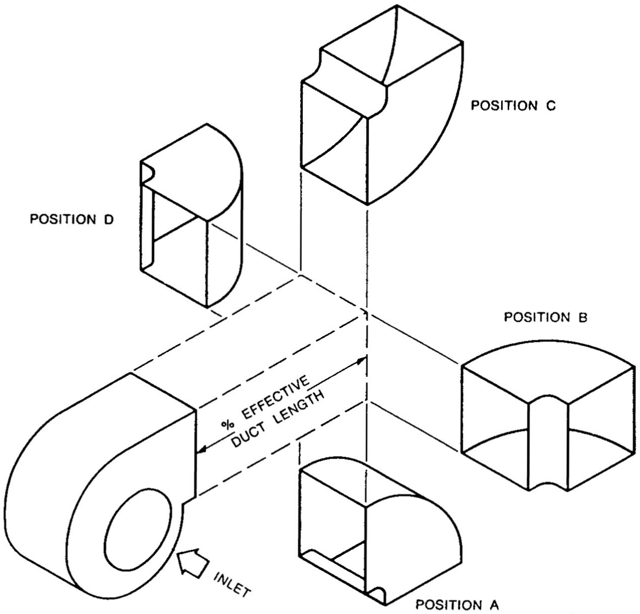
AMERICAN CONFERENCE OF GOVERNMENTAL INDUSTRIAL HYGIENISTS	*SYSTEM EFFECT CURVES FOR OUTLET DUCTS– AXIAL FANS*
	DATE *5–92* FIGURE *6–19*

Figure 6.—System effect curves for outlet ducts—axial fans.



DETERMINE SEF BY USING FIGURES 6–26 AND 6–18

For **DWDI** fans determine SEF using the curve for SWSI fans. Then apply the appropriate multiplier from the tabulation below

MULTIPLIERS FOR DWDI FANS

ELBOW POSITION A = ΔP X 1.00
ELBOW POSITION B = ΔP X 1.25
ELBOW POSITION C = ΔP X 1.00
ELBOW POSITION D = ΔP X 0.85

Blast Area / Outlet Area	Outlet Elbow Position	No Outlet Duct	12% Effective Duct	25% Effective Duct	50% Effective Duct	100% Effective Duct
0.4	A	N	O	P-Q	S	NO SYSTEM EFFECT FACTOR
	B	M-N	N	O-P	R-S	
	C	L-M	M	N	Q	
	D	L-M	M	N	Q	
0.5	A	O-P	P-Q	R	T	
	B	N-O	O-P	Q	S-T	
	C	M-N	N	O-P	R-S	
	D	M-N	N	O-P	R-S	
0.6	A	Q	Q-R	S	U	
	B	P	Q	R	T	
	C	N-O	O	Q	S	
	D	N-O	O	Q	S	
0.7	A	R-S	S	T	V	
	B	Q-R	R-S	S-T	U-V	
	C	P	Q	R-S	T	
	D	P	Q	R-S	T	
0.8	A	S	S-T	T-U	W	
	B	R-S	S	T	V	
	C	Q-R	R	S	U-V	
	D	Q-R	R	S	U-V	
0.9	A	T	T-U	U-V	W	
	B	S	S-T	T-U	W	
	C	R	S	S-T	V	
	D	R	S	S-T	V	
1.0	A	T	T-U	U-V	W	
	B	S-T	T	U	W	
	C	R-S	S	T	V	
	D	R-S	S	T	V	

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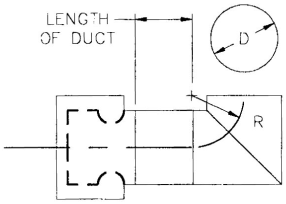
SYSTEM EFFECT CURVES
FOR OUTLET ELBOWS
ON CENTRIFUGAL FANS

DATE *5–92* FIGURE *6–20*

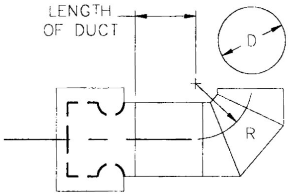
Figure 7.—System effect curves for outlet elbows on centrifugal fans.



SYSTEM EFFECT FACTORS

R/D	NO DUCT	2D DUCT	5D DUCT
–	N	P	R-S

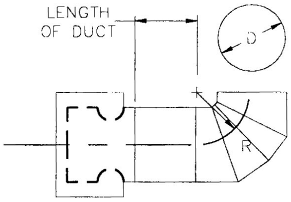
A. TWO-PIECE MITERED 90° ROUND SECTION ELBOW -- NOT VANED



SYSTEM EFFECT FACTORS

R/D	NO DUCT	2D DUCT	5D DUCT
0.5	O	Q	S
0.75	Q	R-S	T-U
1.0	R	S-T	U-V
2.0	R-S	T	U-V
3.0	S	T-U	V

B. THREE-PIECE MITERED 90° ROUND SECTION ELBOW -- NOT VANED



SYSTEM EFFECT FACTORS

R/D	NO DUCT	2D DUCT	5D DUCT
0.5	P-Q	R-S	T
0.75	Q-R	S	U
1.0	R	S-T	U-V
2.0	R-S	T	U-V
3.0	S-T	U	V-W

C. FOUR OR MORE PIECE MITERED 90° ROUND SECTION ELBOW -- NOT VANED

D=Diameter of the inlet collar.
The inside area of the square duct (H X H) should be equal to the inside area of the fan inlet collar.
+The maximum permissible angle of any converging element of the transition is 15°, and for a diverging element 7°.

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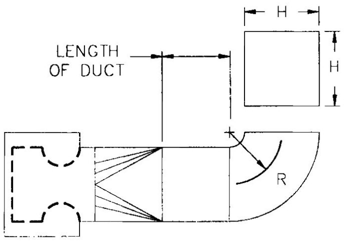
SYSTEM EFFECT CURVES FOR
VARIOUS MITERED ELBOWS
WITHOUT TURNING VANES

DATE *1-98*
FIGURE *6-21*

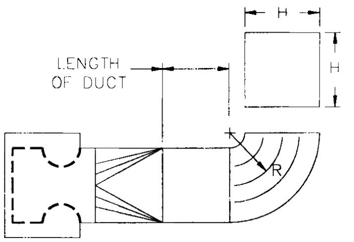
Figure 8.—System effect curves for various mitered elbows without turning vanes.



SYSTEM EFFECT FACTORS

R/D	NO DUCT	2D DUCT	5D DUCT
0.5	O	Q	S
0.75	P	R	S-T
1.0	R	S-T	U-V
2.0	S	T-U	V

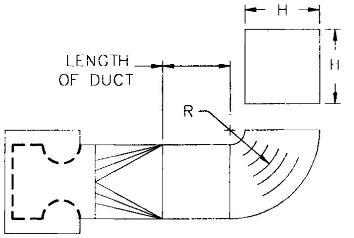
A. SQUARE ELBOW WITH INLET TRANSITION -- NO TURNING VANES.



SYSTEM EFFECT FACTORS

R/D	NO DUCT	2D DUCT	5D DUCT
0.5	S	T-U	V
1.0	T	U-V	W
2.0	V	V-W	W-X

B. SQUARE ELBOW WITH INLET TRANSITION -- 3 LONG TURNING VANES.



SYSTEM EFFECT FACTORS

R/D	NO DUCT	2D DUCT	5D DUCT
0.5	S	T-U	V
1.0	T	U-V	W
2.0	V	V-W	W-X

C. SQUARE ELBOW WITH INLET TRANSITION -- SHORT TURNING VANES.

$D = \frac{2H}{\sqrt{\pi}}$

THE INSIDE AREA OF THE SQUARE DUCT (H X H) IS EQUAL TO THE INSIDE AREA CIRCUMSCRIBED BY THE FAN INLET COLLAR. THE MAXIMUM PERMISSIBLE ANGLE OF ANY CONVERGING ELEMENT OF THE TRANSITION IS 15°, AND FOR A DIVERGING ELEMENT 7.5°.

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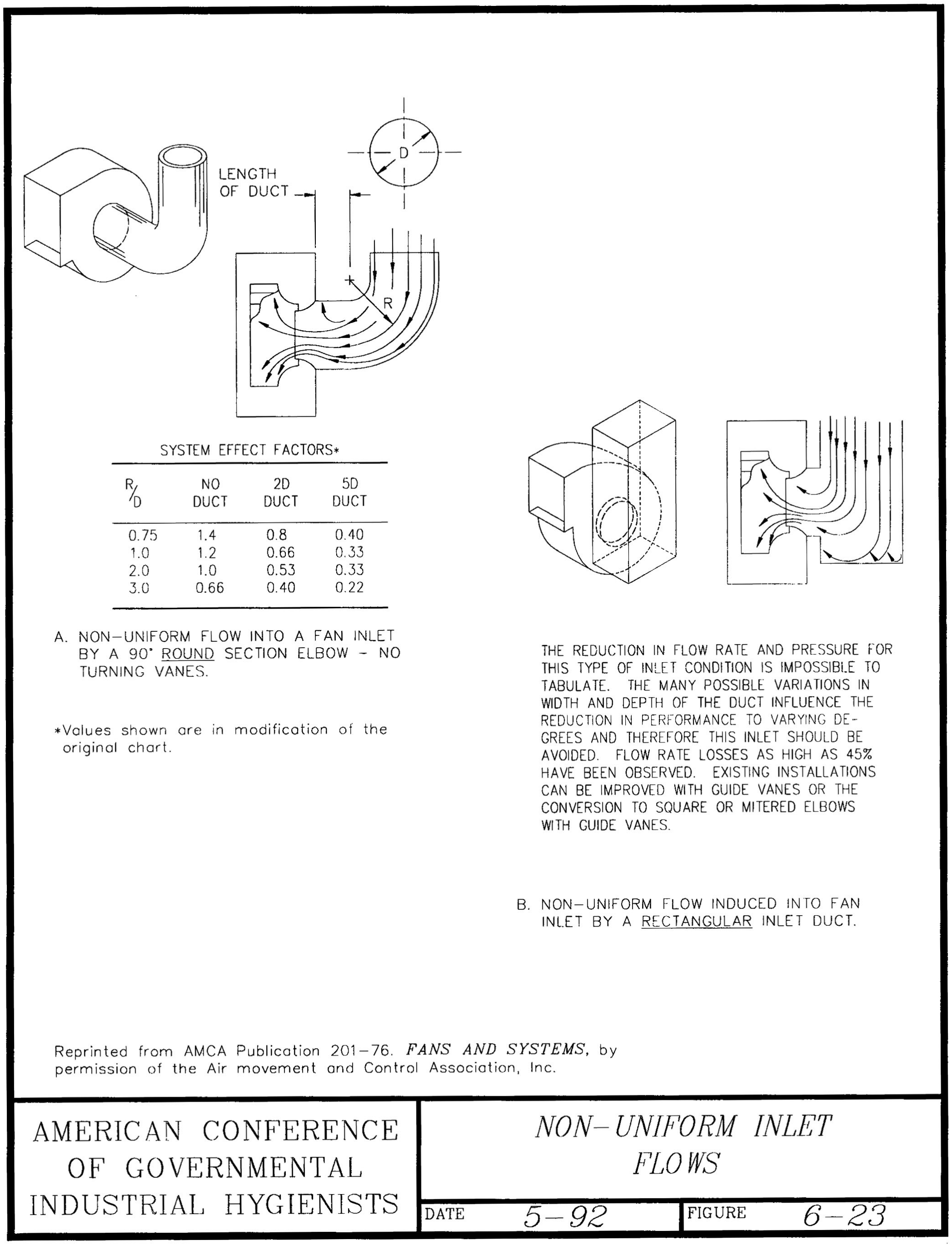
AMERICAN CONFERENCE OF GOVERNMENTAL INDUSTRIAL HYGIENISTS	*SYSTEM EFFECT CURVES FOR OUTLET DUCTS– AXIAL FANS*	
	DATE *5–92*	FIGURE *6–22*

Figure 9.—System effect curves for outlet ducts—axial fans.



SYSTEM EFFECT FACTORS*

R/D	NO DUCT	2D DUCT	5D DUCT
0.75	1.4	0.8	0.40
1.0	1.2	0.66	0.33
2.0	1.0	0.53	0.33
3.0	0.66	0.40	0.22

A. NON-UNIFORM FLOW INTO A FAN INLET BY A 90° ROUND SECTION ELBOW - NO TURNING VANES.

*Values shown are in modification of the original chart.

THE REDUCTION IN FLOW RATE AND PRESSURE FOR THIS TYPE OF INLET CONDITION IS IMPOSSIBLE TO TABULATE. THE MANY POSSIBLE VARIATIONS IN WIDTH AND DEPTH OF THE DUCT INFLUENCE THE REDUCTION IN PERFORMANCE TO VARYING DEGREES AND THEREFORE THIS INLET SHOULD BE AVOIDED. FLOW RATE LOSSES AS HIGH AS 45% HAVE BEEN OBSERVED. EXISTING INSTALLATIONS CAN BE IMPROVED WITH GUIDE VANES OR THE CONVERSION TO SQUARE OR MITERED ELBOWS WITH GUIDE VANES.

B. NON-UNIFORM FLOW INDUCED INTO FAN INLET BY A RECTANGULAR INLET DUCT.

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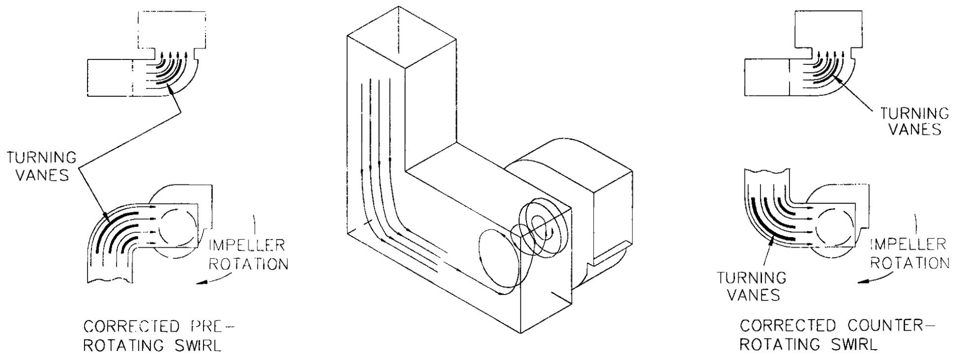
NON-UNIFORM INLET FLOWS

DATE *5-92* FIGURE *6-23*

Figure 10.—Non-uniform inlet flows.



C. NON-UNIFORM FLOW INTO A FAN INLET BY AN INDUCED VORTEX, SPIN OR SWIRL.

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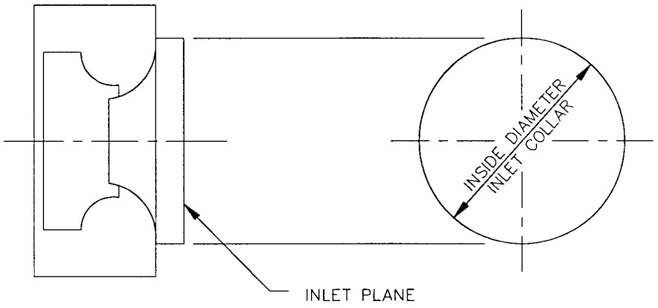
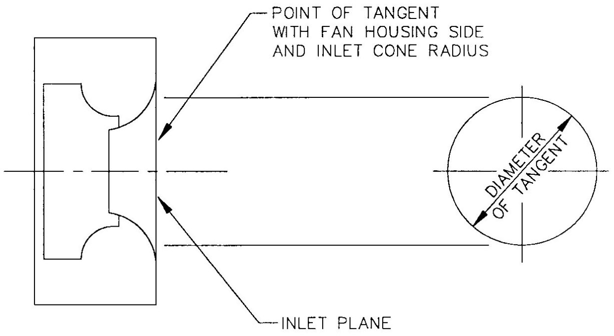
AMERICAN CONFERENCE OF GOVERNMENTAL INDUSTRIAL HYGIENISTS	*NON-UNIFORM INLET CORRECTIONS*			
	DATE	*5-92*	FIGURE	*6-24*

Figure 11.—Non-uniform inlet corrections.



A. FREE INLET AREA PLANE -- FAN WITH INLET COLLAR.



B. FREE INLET AREA PLANE -- FAN WITHOUT INLET COLLAR.

PERCENTAGE OF UNOBSTRUCTED INLET AREA	SYSTEM EFFECT FACTORS
100	NO LOSS
95	0.26
90	0.40
85	0.53
75	0.8
50	1.6
25	2.0

DETERMINE SEF BY CALCULATING INLET VELOCITY AND USING FIGURE 6-26

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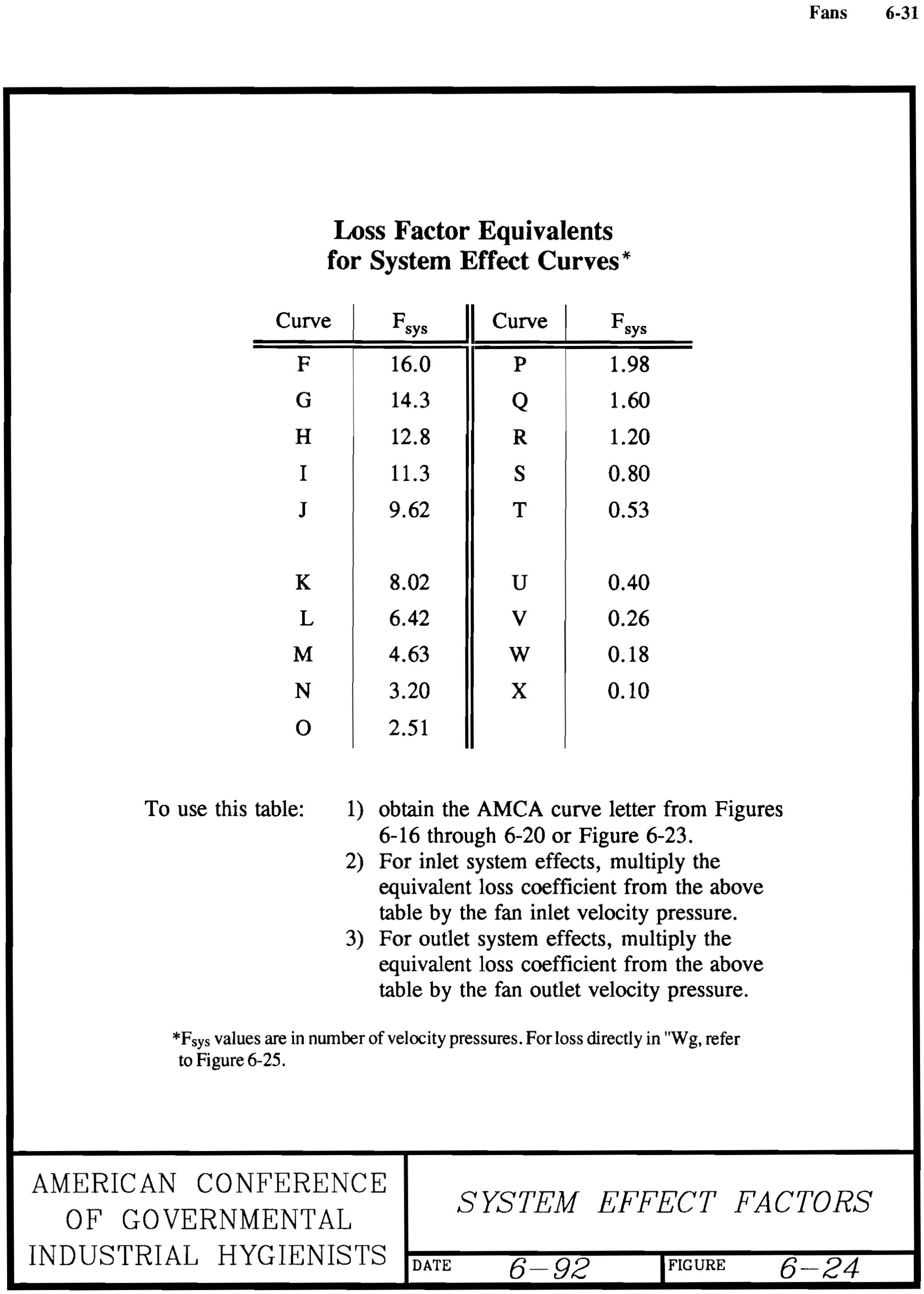
SYSTEM EFFECT CURVES
FOR INLET OBSTRUCTIONS

DATE *5-92* FIGURE *6-25*

Figure 12.—System effect curves for inlet obstructions.



Fans 6-31

Loss Factor Equivalents for System Effect Curves*

Curve	F_{sys}	Curve	F_{sys}
F	16.0	P	1.98
G	14.3	Q	1.60
H	12.8	R	1.20
I	11.3	S	0.80
J	9.62	T	0.53
K	8.02	U	0.40
L	6.42	V	0.26
M	4.63	W	0.18
N	3.20	X	0.10
O	2.51		

To use this table:
1) obtain the AMCA curve letter from Figures 6-16 through 6-20 or Figure 6-23.
2) For inlet system effects, multiply the equivalent loss coefficient from the above table by the fan inlet velocity pressure.
3) For outlet system effects, multiply the equivalent loss coefficient from the above table by the fan outlet velocity pressure.

*F_{sys} values are in number of velocity pressures. For loss directly in "Wg, refer to Figure 6-25.

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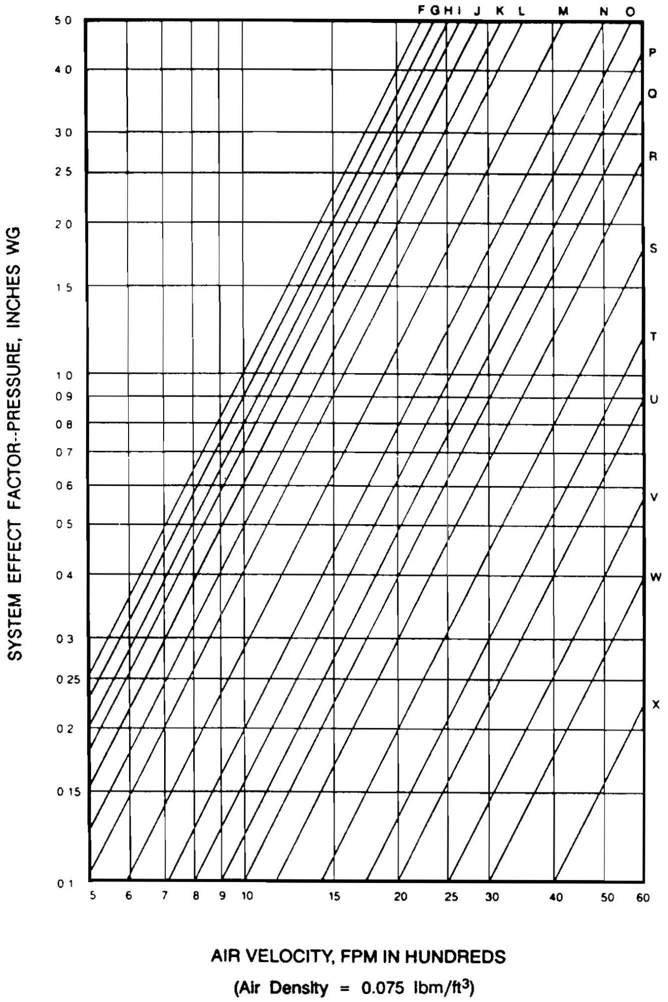
SYSTEM EFFECT FACTORS

DATE *6–92* FIGURE *6–24*

Figure 13.—System effect factors.



Enter the chart at the appropriate air velocity (on the abcissa) read up to the applicable curve, then read across from the curve (to the ordinate) to find the SEF at standard air density.

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SYSTEM EFFECT CURVES

DATE *6–92*
FIGURE *6–25*

Figure 14.—System effect curves.

c. Fan Application and Selection Criteria.—Fans are frequently selected based on airflow requirement, cost, efficient operation, low noise, speed, and low, medium or high pressure. However, other considerations are equally important including: chemical resistance to corrosive gases, vapors, or fumes; ferrous and nonferrous construction for use in potentially explosive applications; space limitations, drive arrangement, rotation, discharge configuration, and motor position should all be considered. These factors should not be disregarded when selecting a fan. The following recommendations are offered:

(1) ***Continuous operation.***—Continuously operating fans (battery room, oil room, paint rooms, plant sumps, air transfer, circulating, superstructure roof or wall exhausters, and main air handling units) should be selected for high efficiency.

(2) ***Intermittent operation***.—Intermittently operating fans (toilets, welding exhaust systems, engine exhaust systems, tunnels, vaults, gate chambers and other confined spaces) should be selected for economic cost.

(3) ***Occupied spaces.***—Fans operating in occupied spaces (offices, control rooms) should be selected for the lowest acceptable noise criteria consistent with the application.

(4) ***Explosive areas.***—Fans used in potentially explosive environments (battery rooms, solvent fume exhaust hoods, propane fired engine generator rooms, tunnels with methane gas) should be nonferrous construction. Explosion proof construction in accordance with *National Fire Protection Association (NFPA) 70* may also be required. Air Movement and Control Association (AMCA) provides three non-sparking construction classifications. All classes must: have the bearings out of the airstream; be grounded; and have non-sparking belts. The classifications are as follows:

(a) AMCA A – Requires all components in the airstream be made of nonferrous materials. Steel shafts are allowed. This the most expensive class.

(b) AMCA B – Requires all components in the airstream be made of nonferrous materials. Housing can be steel.

(c) AMCA C – Nonferrous wearing ring is required on the inlet cone so that, if the impeller shifts, it will rub a nonferrous material. This is the least expensive class.

(5) *Performance ratings*

(a) *Curve ratings.*—When possible, select fans based on the manufacturer's performance curve. By superimposing the system curve on the fan performance curve, the expected operating point can be clearly seen. The performance curves will also show areas where potential unstable operation may occur, i.e. areas where the performance curve breaks or dips. These areas should be avoided. If the system curve results in fan operation to the right of peak efficiency, the fan will be undersized but will tend to operate in a more stable region. If the system curve results in fan operation to the left of peak efficiency, the fan will be oversized and may operate in a less stable region. Fans should be selected to operate slightly to the right of maximum efficiency.

(b) *Table ratings.*—When using table ratings, select fans from the middle range of the cfm rating. Fans in this range will tend to be more efficient and have some additional capacity to accommodate miscalculation of pressure losses. Fans selected near the top of the table may be too small and those near the bottom of the table may be larger than necessary.

(6) *Drive type*

(a) *Direct drive.*—Direct drive fans are more reliable than belt-drive because there is no belt to break. When the fan operating conditions are expected to remain constant, and size permits, direct drive fans are preferred for remote and unmanned applications. Direct drives are also preferred when the fan is located in an area where maintenance will be difficult.

(b) *Belt drive.*—Belt-drive fans offer greater flexibility than direct drive fans because the airflow characteristics can be varied by changing the operating speed. The following should be considered with regard to belt drives:

(i) Drives should be installed with adequate provisions for center distance adjustment to accommodate belt stretch.

(ii) Centers should not exceed 2-1/2 to 3 times the sum of the sheave diameters nor be less than the diameter of the larger sheave.

(iii) The angle of wrap (arc) created by belt contact on the smaller sheave should not be less than 120 degrees.

(iv) Sheave ratios should not exceed 8:1.

(v) The belt speed should not exceed 5,000 ft/min, nor be less than 1,000 ft/min. A speed of 4,000 ft/min is considered best practice.

(vi) Sheaves should be dynamically balanced for rim speeds in excess of 5,000 ft/min.

(vii) A service factor of 1.2 is normal for belts operating between 16 to 24 hours per day. However, Reclamation specifications require belts with a service factor of 1.6.

d. Changing Fan Performance.—Occasionally fan airflow must be increased or decreased to accommodate changes in space loads or use. With “Old” and “New” airflows known, the fan laws are used to calculate the speed, pressure and power requirements for the new operating point. The fan laws stated simply are: [1] Volume varies directly with the speed; [2] Pressure varies with the square of the speed; [3] Horsepower varies as the cube of the speed and directly as the density. Performance changes normally require an increase or decrease in the fan operating speed or the motor.

(1) ***Increasing air flow***.—An increase in speed may also increase the pressure class of the fan and duct system. Refer to the fan performance curve or consult fan manufacturer to verify the new operating point is within the fan pressure class. Refer to *SMACNA Duct Construction Standards* to verify the duct pressure class has not changed. Common pressure classes for standard duct system design are: ½, 1, 2, 3, 4, 6, and 10 inches. These classifications apply duct under positive or negative pressures. Failure to operate the fan and duct system within their pressure class may result in premature failure of this equipment. In general, manufacturers' performance data for fans does not include belt drive losses. These losses can be estimated from figure 15 and added to the fan brake horsepower requirements shown on the manufacturer's tables to determine the actual motor horsepower required.

(2) ***Reducing airflow.***—Reducing airflow can have adverse consequences. Reducing airflow is usually accomplished by changing drives; installing smaller motors and changing drives; installing inlet vane dampers. Before airflows are reduced, the following should be considered:

(a) *Effect on heat transfer through heating and cooling coil.*—Airflow must not be reduced below the minimum required to maintain adequate velocities through electric heating coils.

(b) *Effect on supply air outlets.*—Low airflow may result in poor air distribution causing stagnant conditions in the space.

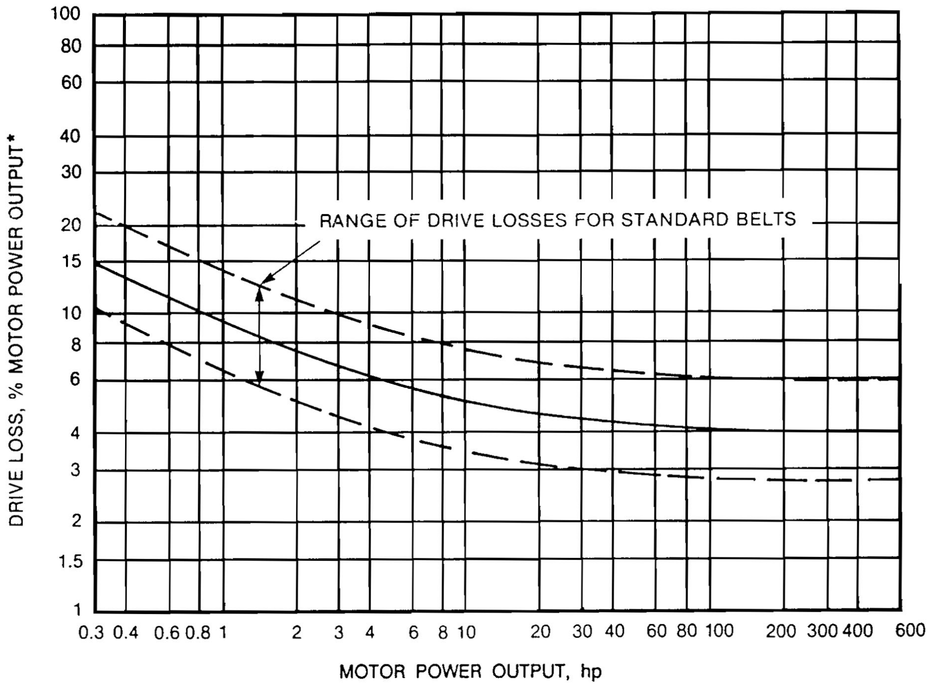
(c) *Effect on outside air and return air.*—Outdoor air must be greater than exhaust air by 1 to 5 percent to maintain slightly positive pressure in the building.

(d) *Effect on ventilation air*.—Minimum outdoor air mandated by code requirements must be maintained at all times.

(e) *Effect on humidity control*.—Reduced airflow may result in increased humidity levels since there is less air available to pick up moisture.

(f) *Effect on fan performance*.—Airflow reduction may shift the fan operating point to the unstable region causing surging.

(g) *Effect on pressure controls.*



HIGHER BELT SPEEDS TEND TO HAVE HIGHER LOSSES
THAN LOWER BELT SPEEDS AT THE SAME HORSEPOWER

*Drive losses are based on the conventional V-belt which has been the "work horse" of the drive industry for several decades.

EXAMPLE

- Motor power output, H_{mo}, is determined to be 13.3 hp
- The belts are the standard type and just warm to the touch immediately after shutdown
- From chart, drive loss = 5.1%
- Drive loss, H_L = 0.051 x 13.3
  = 0.7 hp
- Fan power input, H = 13.3 − 0.7
  = 12.6 hp.

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ESTIMATED BELT
DRIVE LOSS

DATE *5−92*
FIGURE *6−6*

Figure 15.—Estimated belt drive loss.

(3) ***Example***.—The airflow of a fan operating at 9000 cfm and 1-in. w.g. static pressure must be increased to 10,500 cfm. The fan operates at 1500 rpm and requires 5 brake horsepower at the rated conditions. Determine the new operating speed, pressure, and power requirements.

(a) *Speed, N (rpm)*:

$N_2 = N_1(Q_2/Q_1) = 1{,}500(10{,}500/9{,}500) = \underline{1{,}658\ rpm}$

(b) *Static pressure, P (inch w.g.):*

$P_2 = P_1(N_2/N_1) = 1.0(1{,}658/1{,}500)^2 = \underline{1.22\ inch\ w.g.}$

Note: If the duct system was designed for a 1-inch pressure class, the new fan pressure will exceed the design pressure class. Those portions of the duct where the static pressure will exceeded the pressure class should be reinforced or replaced. Failure to account for the excess pressure will result in excess air leakage and noise.

(c) *Brake horsepower:* When possible, the motor actual load amps and voltage should be measured and actual brake horsepower, BHp_a, calculated using the following equation:

$BHp_a = Hp\ (A_a/A_r)(V_a/V_r)$

Where: Hp = Motor horsepower
A_a = Actual motor load amps
A_r = Rated motor load amps
V_a = Actual motor volts
V_r = Rated motor volts

For this example, assume that the actual brake horsepower = 5. Once the actual brake horsepower is determined, the new brake horsepower is estimated as follows:

$BHp_2 = BHp_1\ (N_2/N_1)^3 = 5.0\ (1658/1500)^3 = 6.75\ \text{Hp}$

(d) *Pulley diameter:* to determine pulley diameters to drive the fan, use:
$D_f = D_m(N_m/N_f)$

Where: D_f = Diameter of fan sheave – inches
D_m = Diameter of motor sheave – inches
N_m = Speed of motor – rpm
N_f = Speed of fan – rpm

(e) *Belt length:* to determine pulley belt length, use:

$L_b = 2C + 1.57(D_f + D_m) + [(D_f - D_m)^2/4C]$

Where: L_b = Length of belt – inches
C = Center-to-center distance between motor and fan sheave – inches
D_f = Diameter of fan sheave – inches
D_m = Diameter of motor sheave – inches

(f) *Belt speed:* belt speed can be calculated by:

$S_b = \pi dN/12$

Where: S_b = belt speed – ft/min
d = pitch diameter of pulley – inches
N = rotating speed of pulley – rpm

(g) *Angle of wrap:*

$\alpha_1 = 180 - 2 \sin^{-1}[(R-r)/C]$

$\alpha_2 = 180 + 2 \sin^{-1}[(R-r)/C]$

Where: α_1 = angle of wrap for the small pulley - degrees
α_2 = angle of wrap for the large pulley – degrees
R = radius of large pulley – inches
r = radius of small pulley – inches
C = center-to-center distance of pulleys - inches.

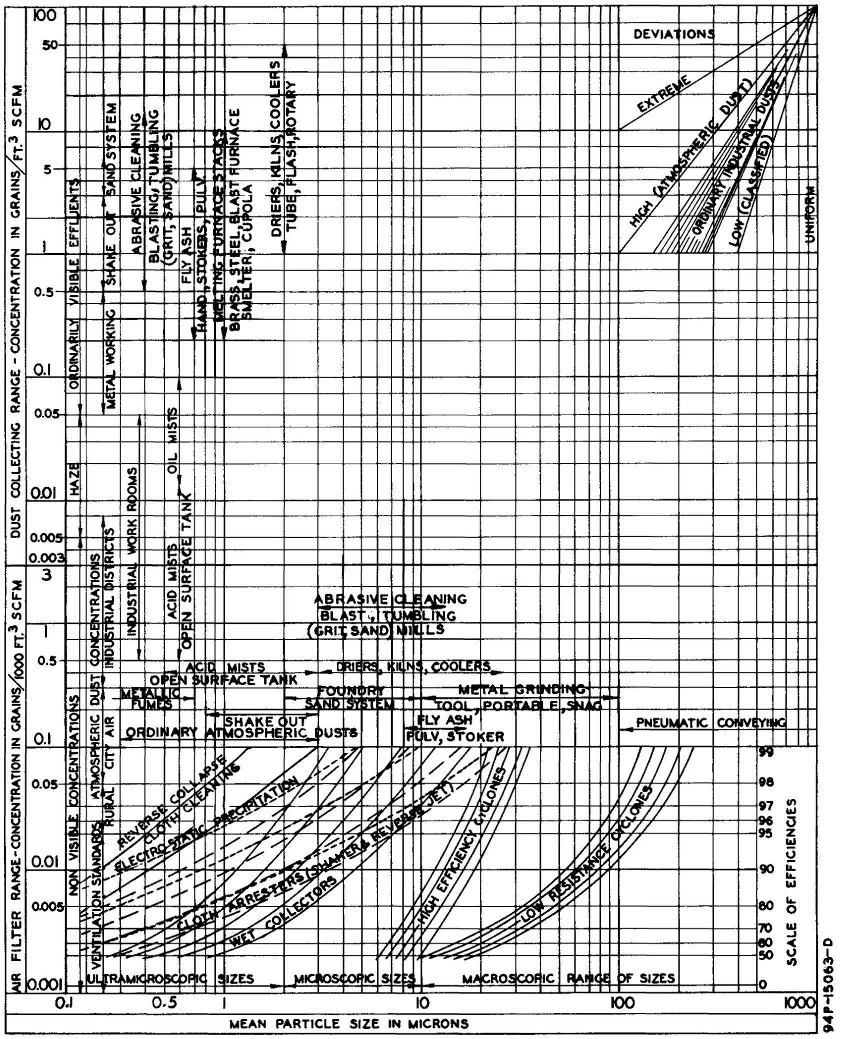
(h) *Drive losses:* Drive losses are usually 3 to 5 percent of the power to the motor and can be estimated from Figure 15. Loss must be added to the fan brake horsepower to determine the actual motor horsepower required.

10. Air Filtration.—Air supplied to the plant must be filtered to remove dust and other particulates that may adversely affect the health of personnel and damage equipment. Various types of filters are available for removing particulates. Figure 16 identifies the range of particle sizes. Table 7 compares air filter characteristics. The most commonly used filters for Reclamation work are: panel, extended surface, renewable media, and electronic. Each type of air filter has an initial (clean) and final (dirty) resistance and a recommended operating airflow speed. For pressure loss calculations, the design resistance for filters should be midway between the initial and final resistance. Carbon filter units have been used for ozone removal.

a. Particulate Filters.—The following are the most frequently used filters in Reclamation applications:

(1) ***Disposable filters.***—Disposable filters are available in panel or extended surface types. Panel filters have the lowest efficiency and their use should be limited. When possible, use medium efficiency extended surface filters. A differential pressure gauge and indicating light should be provided to warn maintenance personnel that the filters need servicing.

(2) ***Renewable media.***—Where plants rely on large modular or built-up AHU or AC systems, plants are remote, or maintenance personnel are not located at or near the plant, renewable media (roll) filters should be considered. These units automatically advance the filter media in response to increased dirt loading. Advancement can be controlled by a photocell light circuit or a differential pressure switch. The photocell control measures the degree of obscurity through the media and advances the media accordingly. The Reclamation has experienced problems with the photocell controllers. Photocell controllers are prone to false indications such as localized and unusually heavy dirt deposit that obscure the light and advance the media prematurely when the remainder of the filter surface is in satisfactory condition. Consequently, Reclamation design



RANGE OF PARTICLE SIZES, CONCENTRATION, & COLLECTOR PERFORMANCE

COMPILED BY S. SYLVAN APRIL 1952 : COPYRIGHT 1952 AMERICAN AIR FILTER CO. INC.

ACKNOWLEDGEMENTS OF PARTIAL SOURCES OF DATA REPORTED:
1 FRANK W.G. - AMERICAN AIR FILTER - SIZE AND CHARACTERISTICS OF AIR BORNE SOLIDS - 1931
2 FIRST AND DRINKER - ARCHIVES OF INDUSTRIAL HYGIENE AND OCCUPATIONAL MEDICINE - APRIL 1952
3 TAFT INSTITUTE AND AAF LABORATORY TEST DATA - 1961 - '63
4 REVERSE COLLAPSE CLOTH CLEANING ADDED 1964

AMERICAN CONFERENCE OF GOVERNMENTAL INDUSTRIAL HYGIENISTS	*RANGE OF PARTICLE SIZE*	
	DATE *1-88*	FIGURE *4-14*

Figure 16.—Range of particle size.

Table 7.—Comparison of some important filter characteristics[*]

Type	Resistance in. w.g.[1,2]		ASHRAE performance[4]		Face velocity fpm	Maintenance[5]	
	Initial	**Final**	**Arrestance**	**Efficiency**		**Labor**	**Material**
Low/medium efficiency							
1. Glass throwaway (2-inches deep)	0.1	0.5	77%	NA[6]	300	High	High
2. High Velocity (permanent units) (2-inches deep)	0.1	0.5	73%	NA[6]	500	High	Low
3. Automatic (viscous)	0.4	0.4	80%	NA[6]	500	Low	Low
Medium/high efficiency							
1. Extended Syrface (dry)	0.15-	0.5-	90-99%	25-95%	300-625	Medium	Medium
2. Electrostatic:	0.60	1.25					
a. Dry Agglomerator/ Roll Media	0.35	0.35	NA[7]	90%	500	Medium	Low
b. Dry Agglomerator/ Extended Surface	0.55	1.25	NA[7]	95%	530	Medium	Medium
c. Automatic Wash Type	0.25	0.25	NA[7]	85-95%	400-600	Low	Low
Ultra high efficiency							
1. HEPA	0.5-1.0	1.0-3.0	Note[3]	Note[3]	250-500	High	High

Note 1: Pressure drop values shown constitute a range or average, whichever is applicable

Note 2: Final pressure drop indicates point at which filter or filter media is removed and the media is either cleaned or replaced. All others are cleaned in place, automatically, manually, or media renewed automatically. Therefore, pressure drop remains constant.

Note 3: 95-99.7% by particle count, DOP Test

Note 4: ASHRAE Standard 52-76 defines (a) A resistance as a measure of the ability to remove injected synthetic dust, calculated as a percentage on a weight basis and (b) Efficiency as a measure of the ability to remove atmospheric duct determined on a light-transmission (dust-spot) basis.

Note 5: Compared to other types within efficiency category.

Note 6: Too low to be meaningful.

Note 7: Too high to be meaningful.

Air filters should be used only for supply air systems or other applications where dust loading does not exceed 1 grain per 1000 cubic feet of air. ACGIH, Industrial Ventilation – A Manual of Recommended Practice, 1992 typically use a differential pressure gage for visual indication of filter condition and a differential pressure switch to advance the media when the pressure loss rises to a predetermined setting recommended by the manufacturer. The controls for the filter should also include an indicating light to warn personnel when the media has fully advanced to the end of the roll and needs servicing.

(3) ***Electronic*.—**Roll filters are not very efficient, therefore, they are frequently combined with electronic filters where maximum particulate removal is desired. In most cases a pre-filter is used to arrest large particulates and a downstream filter is provided to filter agglomerates that occasionally flake off the electronic filter.

(4) ***Washable*.—**Integral filters furnished with roof or wall mounted outdoor air supply fans should be the permanent, washable, 2-inch thick, aluminum mesh type.

b. **Carbon (Ozone) Filters**.—Reclamation generator designs allow a minimal amount of ozone generation that is within acceptable Occupational Safety and Health Administration (OSHA) limits. However, several Reclamation plants have experienced excessive ozone generation due to corona discharge from damage to the windings. When designing a carbon filter installation the following should be given careful consideration:

(1) ***Exposure limits.*—**Prolonged exposure to ozone may be hazardous to personnel depending on the concentration and the length of time personnel are exposed. The permissible ozone concentration within a facility is regulated by OSHA. In addition to health hazards, ozone can cause corrosion of metals and deterioration of rubber seals.

(2) ***Monitoring*.—**Once detected, ozone must be continuously monitored to ensure that acceptable levels are maintained. When the concentration rises to the maximum acceptable OSHA limit, steps must be taken to reduce the levels or to prevent further increase. Expense and downtime usually preclude immediate repairs for ozone problems. Until repairs can be made, carbon filter units should be installed to absorb the ozone. However, it is important to note that the size carbon filter units can not be determined by measuring ozone level in the housing and in the generator room while the unit is operating. No general guidance can be provided to estimate the airflow requirements or the amount of carbon required.

(3) ***Testing and sizing.***—The carbon filter size depends on the ozone generation rate, rate of decay, and background levels. These values must be determined through controlled testing and analysis. Reclamation *Engineering Monograph No. 44 – Ozone Abatement in Air-Cooled Hydroelectric Generators*, outlines the methods and procedures required to size carbon filter units for ozone control. Testing requires coordination with plant personnel since unit downtime will be required at various times throughout the tests. If the generating units can not be taken off-line for testing, the only alternative for controlling an ozone problem is through trial and error. Since this relies strictly on guesswork, the carbon filter unit may or may not control the problem. Furthermore, some downtime will still be required for cutting duct openings in the generator housing and connecting ductwork. If the problem is not controlled, these procedures will have to be repeated at additional expense and downtime until the problem is controlled. Although this approach is not recommended, it has been taken by some Reclamation facilities. Before purchasing carbon filters, personnel should try to borrow carbon units used at other plants such as Montrose, Grand Coulee, and Davis.

Safety factors should be considered. If the ozone problem persists, eventually the filter will not be capable of controlling the concentration within acceptable limits. Therefore, addition of a suitable safety factor (up to 25 percent) should be considered.

(4) ***Types of carbon filters.***—Carbon filters are available in two forms.

(a) *Panel filters.*—Commercially available disposable panel air filters do not contain enough carbon to be effective against the high ozone concentrations in the generating rooms. However, at plants experiencing ozone problems, these filters may be effective at the reduced concentrations away from the generator room. Panel filters should be installed in the air handling units serving occupied areas until the ozone problem is corrected.

(b) *Bulk carbon.*—Typical carbon filters required for ozone absorption consist of an air handling unit and several trays loaded with bulk carbon made from coconut shells. A spare set of trays should be maintained at the plant to reduce down time during replacement. Eventually, the installed carbon will become saturated and ozone levels will begin to rise. The saturated trays should be removed and the spare set is installed. The saturated carbon filters should returned to a supplier for reactivation and return to the plant.

(5) ***Controls.***—Carbon filter unit controls should be designed to operate continuously while the generating unit is operating. If the ozone problem is severe, the carbon filter unit should continue operating during generator shutdown.

(6) ***Potential effects on CO_2 system.***—When carbon filter units are installed the potential effects on the CO_2 system should also be evaluated.

(a) *CO_2 Containment.*—A CO_2 discharge may leak through improperly installed or poorly constructed ductwork. Flexible ductwork is prone to leakage and should not be used. Duct construction and leakage class should be suitable for the intended pressures of a CO_2 discharge.

(b) *Dilution.*—If the carbon filter unit cannot be installed near the generating unit, then the effect of the added volume of the ductwork should to be considered to ensure that the required CO_2 concentration is maintained.

(c) *Controls.*—Once a filter is properly sized, installed, and ozone levels are controlled, there may be tendency to postpone repairs that are causing ozone generation. The temporary fix may become permanent to the degree that it may be required to operate longer than originally anticipated. Therefore filter controls should be designed to automatically shutdown the filter on an impending CO_2 discharge. The degree of operating controls and interlocks required for equipment protection, and use of low leakage dampers to ensure CO_2 concentration is maintained within the housing should be considered.

11. **Service Life of Equipment**.—The service life effective costs of equipment should be considered in the HVAC design. Table 8 shows the estimated service life of various types of HVAC equipment.

Table 8.—Estimated equipment service life

Equipment	Years[1]	Equipment	Years[1]
Air conditioners		Fans	
1. window	10	1. centrifugal	25
2. through-wall	15	2. axial	20
3. packaged water-cooled	15	3. propeller	15
		4. roof-mounted	20
Heat pumps		Coils	
1. air-to-air	15	1. DX, water, steam	20
2. water-to-air	19	2. Electric	15
Roof-top air conditioners		Shell-and-tube heat exchangers	
1. single-zone	15		24
2. multi-zone	15		
Electric unit heater	13	Reciprocating compressors	20
Electric radiant heaters	10	Package reciprocating chillers	
			20
Air terminals		Condensers	
1. Diffuers, grilles, registers	27	1. Air-cooled condensers	20
2. Induction and fan-coil units	20	2. Evaporative Condenser	20
3. VAV boxes	20		
Air washers	17	Cooling tower (Galvanized)	20
Duct work	30	Insulation	
		1. molded	20
		2. blanket	24
Dampers	20	Electric transformers	30
Pumps		Controls	
1. base-mounted	20	1. pneumatic	20
2. pipe-mounted	10	2. electric	16
3. sump and well	10	3. elctronic	15
4. condensate	15		
Electric motors	18	Motor starters	17
Valve actuators			
1. pneumatic	20		
2. self-contained	10		

[1] Median Years
SMACNA: HVAC Systems Duct Design, 1990

C. ENCLOSED STAIRWELLS AND ELEVATOR SHAFTS.

1. **Ventilation.**—Enclosed stairwells and elevator shafts should be ventilated continuously. Ventilation air is rarely accomplished through a supply duct from the main HVAC unit. Air is usually transferred into the stairwell from the lower levels of the plants. Air flows up the stairwell to the highest level and exits back into the superstructure. Air from elevator shafts should be exhausted outdoors.

    a. **Fan.**—Plant stairwells and elevator shafts are usually ventilated with wall mounted propeller fans located at the bottom of the stairs. Typical airflow through an enclosed stairwell or elevator shaft is 900 to 1,000 cfm.

    b. **Control.**—The stairwell and elevator shaft ventilating fans should be operated continuously. Control equipment typically consists of a combination motor starter located near the fan. The controls for the stairwell ventilating system should be interlocked with the stairwell pressure fan controls to prevent simultaneous operation of both fans. When fan installation prevents visual verification of operation, indicating lights should be provided on the motor starter panels to verify system operating status. The lights should be energized through an airflow switch.

    c. **Dampers.**—Stairwell and elevator shaft supply fan openings and air transfer openings must be provided with leakage class I fire/smoke dampers. Motor-operated dampers capable of manual-electric-thermal activation should be provided. This will facilitate testing and enable automatic reset. A smoke detector should be located near the stairwell or elevator to de-energize the ventilating fan and close the dampers if smoke is detected.

2. **Pressurization.**— Elevator shafts are not normally provided with pressurization systems, therefore, the following discussion pertains to enclosed stairwells only. Stairwells should be provided with a dedicated pressurization system that is independent of the main plant HVAC system that is automatically de-energized when fire or smoke is detected in the plant. The fan must be capable of pressurizing the stairwell with 100 percent outdoor air to provide a refuge zone for personnel and to allow safe plant evacuation. The system must be capable of minimizing smoke penetration when a limited number of doors are opened during evacuation.

a. **Enclosure Integrity.**—To maintain pressure, the stairwell enclosure must be adequately sealed. On new plant designs sealing of all openings and penetrations should be coordinated with the design architects. On retrofit applications a thorough inspection of the stairwell should be conducted to identify areas that need to be sealed and to determine the most appropriate method for sealing. *NFPA 92A* discusses method and procedures for estimating leakage through stairwells.

b. **Design.**—Stairwell pressure system should be designed in accordance with the requirements and procedures of *NFPA 92A* and *ASHRAE Design of Smoke Management Systems*. A typical design includes a fan, outdoor air louver, control damper and operator, relief or bypass dampers, ductwork, and controls.

(1) ***Airflow.***—Although propeller fans may be acceptable, Reclamation designs have used centrifugal fans almost exclusively because they are less likely to be adversely affected by wind. Fan should be sized to provide an airflow velocity no less than 50 ft/min through open doors and the required pressure differential between the stairwell and the building. On plants with occupied spaces, i.e. offices, a minimum of three doors (door on occupied floor plus door for the floor above and below the occupied level) should be assumed to be open simultaneously. An airflow of approximately 3,500 to 5,000 cfm should be sufficient in most cases, however, actual code requirements should always be verified.

(2) ***Pressure.***—The fan pressure required depends on a variety of conditions including: airflow; enclosure integrity; losses from all system components; and air temperature both inside and outside the stairwell. In most cases the plant air outside the stairwell will be constant and will have little effect on the pressure system. However in the event of a fire, the indoor temperature outside the stairwell depends on whether the area is protected by a sprinkler system. A fire in a sprinkled area will not have significant effect, however, a fire in an unsprinkled area will result in very high temperatures affecting air buoyancy that will affect the stairwell pressure requirements. Stack effect in the stairwell may result due to seasonal (winter vs. summer) temperature variations and its effects on air buoyancy.

Wind velocity will also produce pressure variations on stairwells that must be considered.

(3) ***Fans.***—Fans may be propeller or centrifugal. Propeller fans have relatively flat performance curves resulting in quick airflow response without significant pressure fluctuations when doors are opened and closed. Propeller fans are also more economical to purchase and install. However, airflow from propeller fans may be reduced if the fan is installed in a location subject to wind. If shielded from the wind, propeller fans are satisfactory.

In areas where wind is a problem or adequate protection against winds can not be assured, centrifugal fans should be considered.

(4) ***Louvers and intake dampers.***—Stairwell pressure systems require high air flows but operate infrequently. Therefore, adherence to normal air velocity criteria intended to prevent moisture entrainment, and reduce noise and pressure loss through intake louvers is not critical. However, preliminary power estimates should ensure that adequate power is available to operate the fan at the higher pressure required to, overcome friction through smaller louvers. The fresh air intake control damper should be a parallel blade, two position, motor-operated type. Full opening time for motor operators should not exceed 75 seconds.

(5) ***Pressure relief dampers.***—The stairwell pressure must be controlled to ensure that the total maximum force required to open egress doors (due to the door closing actuator and the differential pressure across the door) does not exceed the *NFPA 101 Life Safety Code* requirements of 30 lbs. This requires use of relief dampers, bypass dampers, pressure controllers or other suitable methods described in *NFPA 92A*. Pressure relief may be accomplished by relief dampers or open doorways. Relief dampers may be manual such as a barometric or counterbalanced relief dampers or pressure actuated motor-operated type. Because it is relatively failsafe, the counter-balance relief damper is preferred over the motor-operated type. Relief openings should be located on walls away form prevailing winds.

Doorways exiting outside the plant are rare in Reclamation stairwells. This method of relief may also prevent pressurization if the exit door is blocked open. Therefore this method of pressure relief has not been used or recommended.

(6) ***Single and multiple injection*.—**Most of the initial Reclamation stairwell pressure systems were retrofit installations with a single point of air injection into the stairwell. The injection point is typically located near the top of the stairwell and the fan. However, better overall pressure distribution can be maintained if air is supplied through multiple injection points spaced at three story intervals.

c. **Operation.—**The pressure fan should be off when the ventilating system fan is operating. When fire or smoke are detected anywhere in the plant, the stairwell ventilating system should de-energize, the fire/smoke dampers should close, and the pressure fan should start. The fan should continue to operate until manually reset at the Fire Command and Control Panel. A smoke detector should also be provided in the fresh air intake duct to the pressure fan to prevent smoke recirculation by automatically de-energizing the fan.

d. **Controls and Indicating Lights.—**The pressure fan should be controlled through a combination motor starter. The fan must be controlled to enable operation at the desired state in 60 seconds or less. The starter should be interlocked with the stairwell ventilating fan controls to prevent simultaneous operation of both fans. Controls for the stairwell pressure system should be located in a panel near a plant entry point designated for use by the local Fire Marshall or other firefighting entity. An airflow switch should be provided to positively verify airflow at an indicating light on the stairwell pressure control panel. A system test circuit should also be included on the control panel to enable periodic testing of the stairwell pressure system and its interfaces to other equipment and systems.

e. **Testing.—**Stairwell should be tested to verify: stairwell integrity; proper activation by the smoke detectors and the fire detection and alarm system; ability to maintain required pressure differences between stairwell and plant; air flows out of the stairwell through open doors; and door opening forces do not exceed *NFPA 101 Life Safety Code* requirements. Testing should be in accordance with *NFPA 92A*.

D. **SMOKE EXHAUST.**—Except for stairwell pressure fans which are energized when fire or smoke are detected, and battery room exhaust fans where shutdown may pose a greater hazard by allowing hydrogen to accumulate, all HVAC systems and equipment should be designed to automatically shutdown when fire or smoke are detected by the fire detection and alarm system. HVAC systems should also be designed to exhaust smoke after fire and smoke alarms are reset. Smoke exhaust requires additional planning to properly locate equipment to allow using the HVAC system to exhaust all areas of the plant.

1. **Intake and Exhaust Openings.**—Outdoor air and exhaust louvers should be separated as much as possible to prevent smoke re-circulation.

2. **Smoke Chases.**—When possible, smoke exhaust chases should be built into the plant.

3. **Smoke Transfer**.—Smoke transfer between adjacent areas, for smoke removal purposes, should be avoided.

4. **Fire and Smoke Dampers.**—Fire and smoke damper locations must be coordinated with the architect or plant designer. Fire and smoke dampers should be provided with links capable of manual, electric, and thermal activation. Fire and smoke dampers to be used for smoke purging, and those, which would be difficult to manually reset, should be motor-operated to allow remote control. The damper assembly should include red indicating lights initiated through micro switches to provide positive proof of the dampers closed position.

5. **Smoke Exhaust Panel.**—The smoke exhaust panel provided should allow use of the HVAC equipment to purge the plant of smoke. The following should be considered when designing a smoke exhaust panel:

   a. **Override HVAC Controls.**—The smoke exhaust panel should override and enable manual remote control of the plant HVAC systems and equipment to allow smoke purging.

   b. **Indicating Lights.**—The panel should be provided with indicating lights to monitor the operating status of the purge fans.

c. **Zone Purge Control.—**Sequential purging will minimize smoke transfer to unaffected areas. When possible, independent controls should be provided to allow sequential purging of plant areas or spaces.

d. **Location**.—The smoke exhaust panel should be conveniently located near an entry door designated for use by the local fire Marshall or other firefighting entity.

e. **Keyed.—**The smoke exhaust panel should be keyed to prevent use by unauthorized personnel.

f. **Test Feature.—**The smoke exhaust panel should include test circuits that will allow momentary operation of the smoke exhaust system and verification of interfaces with the fire detection and alarm system.

6. **Coordination of HVAC and Fire Detection and Alarm System.—** Coordination of HVAC system controls and smoke exhaust panel controls with the fire detection and alarm system is essential to ensure proper interfaces and operation.

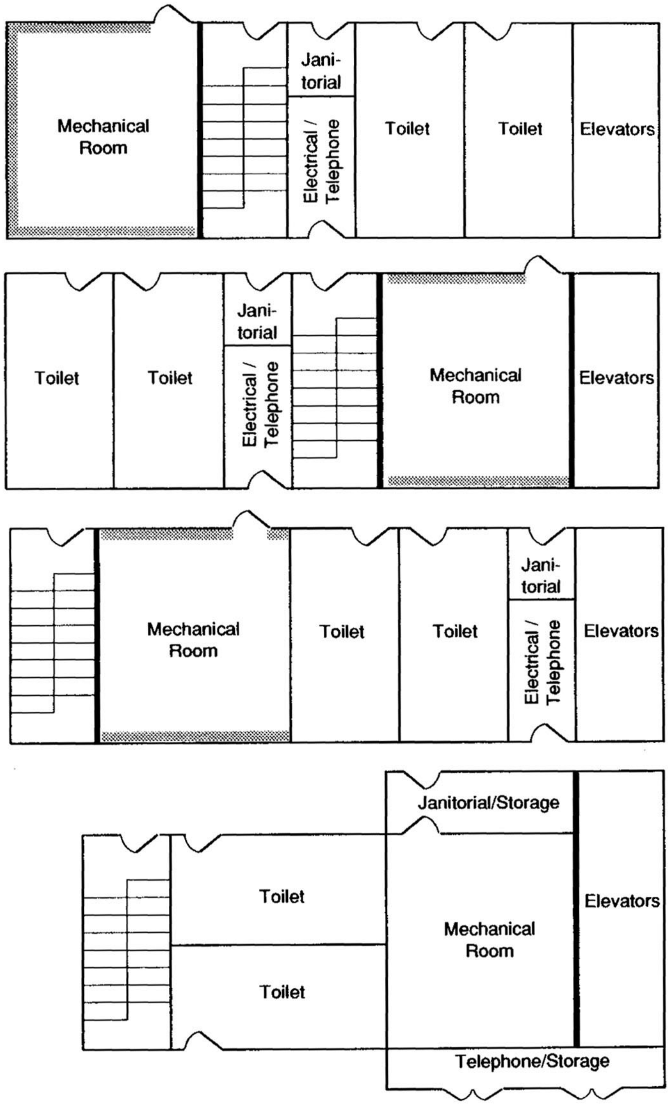
E. **NOISE AND VIBRATION CONSIDERATIONS.**—Poorly designed HVAC systems are prone to noise and vibration. Figure 17 shows common mistakes that increase noise and vibration. Figure 18 illustrates good engineering practices that result in low noise and vibration. Adherence to the following recommendations should help reduce noise and vibration:

1. **Fans and Air Handling Units**

a. **High Efficiency.—**Select the equipment for operation at the highest efficiency consistent with the application. Verify that the operating point is to the right of any unstable areas, i.e. areas where the fan curve dips downward. See figure 19.

b. **Low Velocity.—**Maintain the lowest possible discharge velocity.

c. **Isolate.—**Install flexible connectors between the equipment and ductwork. Provide vibration isolators for equipment with motors exceeding 1 Hp. Locate between the equipment and the floor or ceiling as appropriate. The isolators should be appropriate for the intended application. Figures 20, 20a, 21, 21a, 22, and 23 show the various types of isolators and their applications.



Poor Core Layout

Exposes 3 mechanical room walls to surrounding tenant space. Impenetrable partition between mechanical room and exit stairs results in supply and return air wall openings next to tenant space.

Fair Core Layout

Exposes 2 mechanical room walls to tenant space. Impenetrable mechanical room partition to elevators and exit stairs results in supply and return air wall openings next to tenant space.

Better Core Layout

Exposes 2 mechanical room walls to tenant space. Ceiling over toilets can be used for either supply ducts or return air path.

Best Core Layout

No mechanical room walls exposed to tenant space. No supply or return air openings need be next to tenant space. Ceiling over toilets can be used for supply air ducts or return air path.

Figure 17.—Acoustical comparison of various building core area layouts.

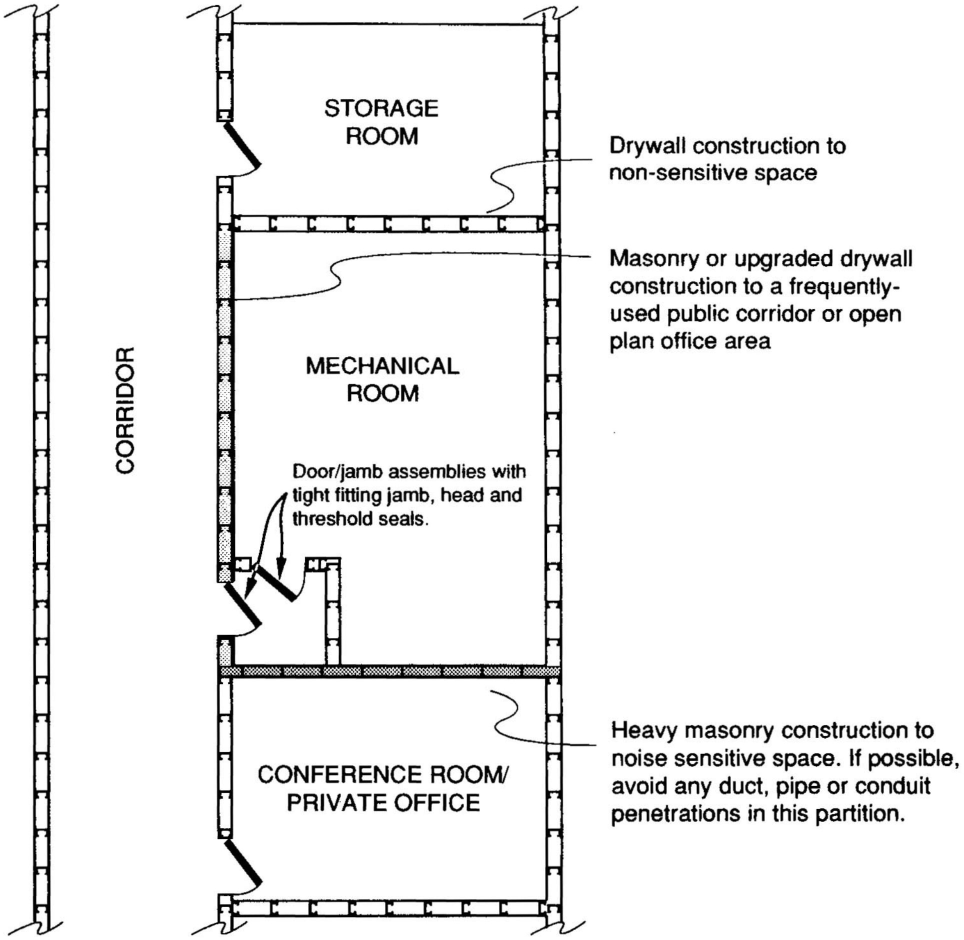
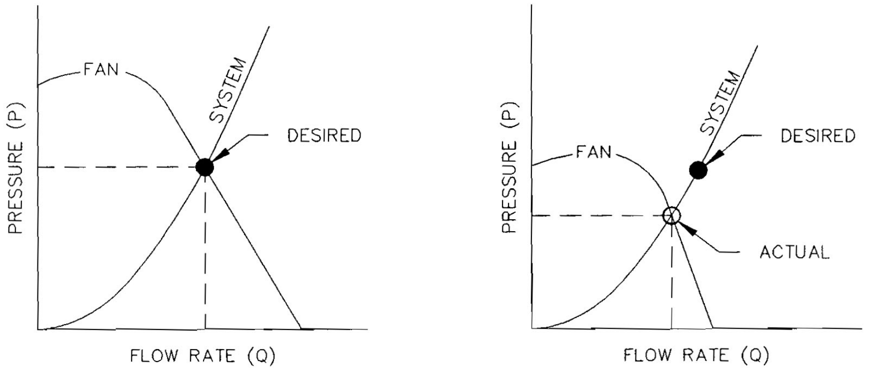
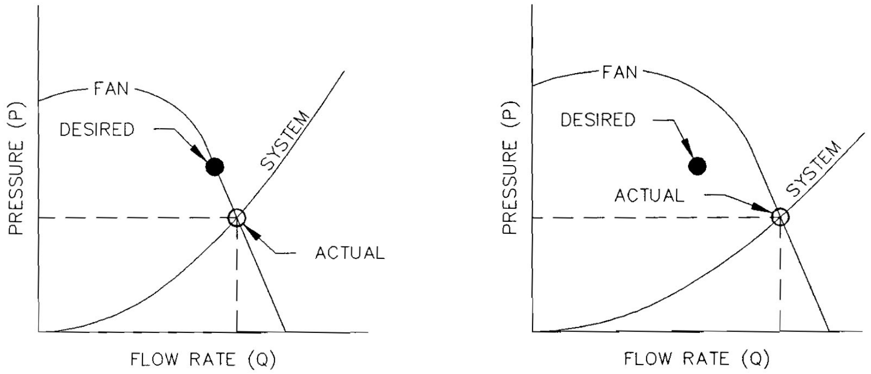


Figure 18.—Guidelines for the preliminary selection of mechanical room walls.



A. FAN AND SYSTEM MATCHED

B. WRONG FAN.



C. WRONG SYSTEM.

D. BOTH FAN AND SYSTEM WRONG

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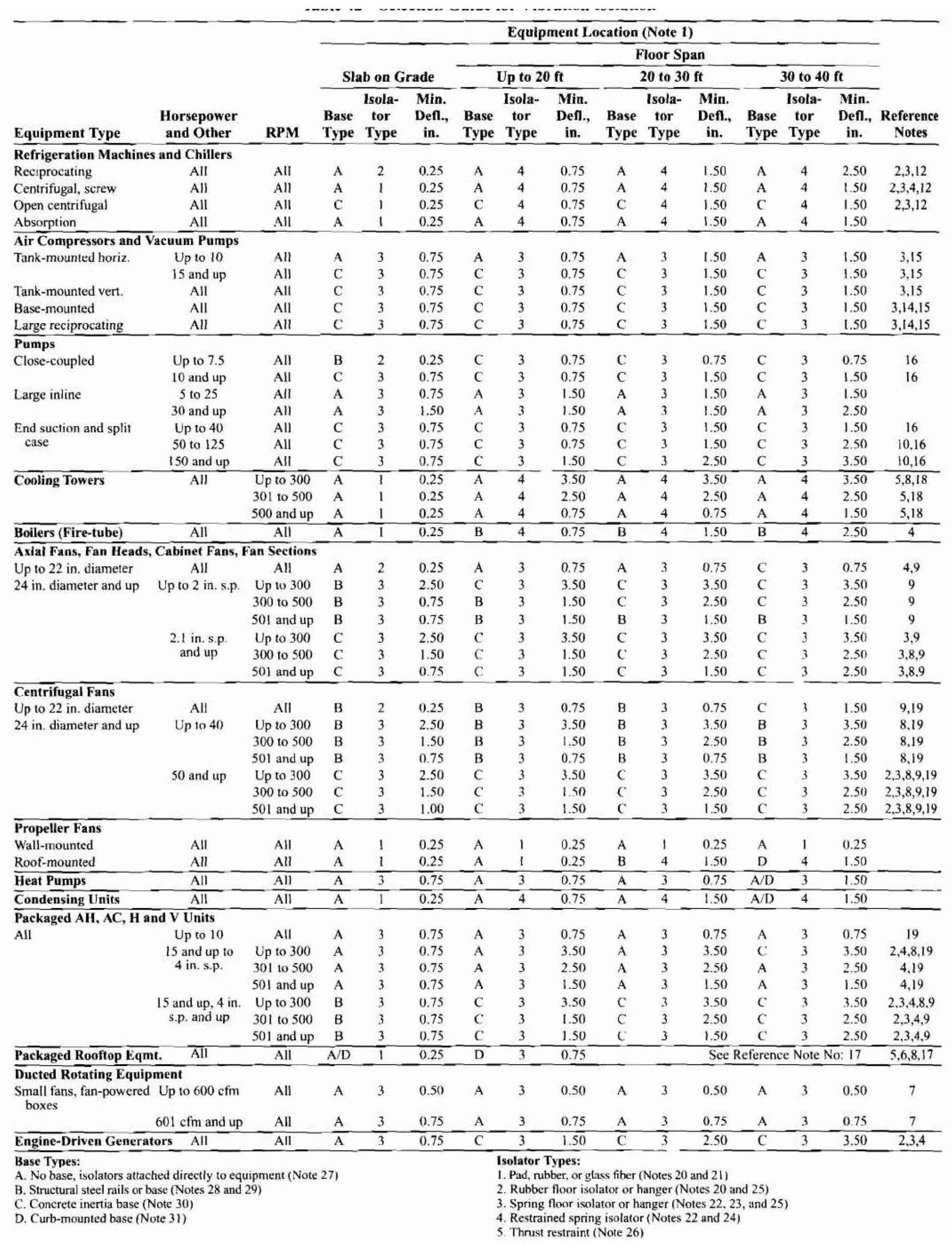
INDUSTRIAL HYGIENISTS

ACTUAL VERSUS DESIRED
POINT OF OPERATION

DATE *1–88*

FIGURE *6–9*

Figure 19.—Actual versus desired point of operation.



			Equipment Location (Note 1)												
						Floor Span									
			Slab on Grade			Up to 20 ft			20 to 30 ft			30 to 40 ft			
Equipment Type	Horsepower and Other	RPM	Base Type	Isolator Type	Min. Defl., in.	Base Type	Isolator Type	Min. Defl., in.	Base Type	Isolator Type	Min. Defl., in.	Base Type	Isolator Type	Min. Defl., in.	Reference Notes
Refrigeration Machines and Chillers															
Reciprocating	All	All	A	2	0.25	A	4	0.75	A	4	1.50	A	4	2.50	2,3,12
Centrifugal, screw	All	All	A	1	0.25	A	4	0.75	A	4	1.50	A	4	1.50	2,3,4,12
Open centrifugal	All	All	C	1	0.25	C	4	0.75	C	4	1.50	C	4	1.50	2,3,12
Absorption	All	All	A	1	0.25	A	4	0.75	A	4	1.50	A	4	1.50	
Air Compressors and Vacuum Pumps															
Tank-mounted horiz.	Up to 10	All	A	3	0.75	A	3	0.75	A	3	1.50	A	3	1.50	3,15
	15 and up	All	C	3	0.75	C	3	0.75	C	3	1.50	C	3	1.50	3,15
Tank-mounted vert.	All	All	C	3	0.75	C	3	0.75	C	3	1.50	C	3	1.50	3,15
Base-mounted	All	All	C	3	0.75	C	3	0.75	C	3	1.50	C	3	1.50	3,14,15
Large reciprocating	All	All	C	3	0.75	C	3	0.75	C	3	1.50	C	3	1.50	3,14,15
Pumps															
Close-coupled	Up to 7.5	All	B	2	0.25	C	3	0.75	C	3	0.75	C	3	0.75	16
	10 and up	All	C	3	0.75	C	3	0.75	C	3	1.50	C	3	1.50	16
Large inline	5 to 25	All	A	3	0.75	A	3	1.50	A	3	1.50	A	3	1.50	
	30 and up	All	A	3	1.50	A	3	1.50	A	3	1.50	A	3	2.50	
End suction and split case	Up to 40	All	C	3	0.75	C	3	0.75	C	3	1.50	C	3	1.50	16
	50 to 125	All	C	3	0.75	C	3	0.75	C	3	1.50	C	3	2.50	10,16
	150 and up	All	C	3	0.75	C	3	1.50	C	3	2.50	C	3	3.50	10,16
Cooling Towers	All	Up to 300	A	1	0.25	A	4	3.50	A	4	3.50	A	4	3.50	5,8,18
		301 to 500	A	1	0.25	A	4	2.50	A	4	2.50	A	4	2.50	5,18
		500 and up	A	1	0.25	A	4	0.75	A	4	0.75	A	4	1.50	5,18
Boilers (Fire-tube)	All	All	A	1	0.25	B	4	0.75	B	4	1.50	B	4	2.50	4
Axial Fans, Fan Heads, Cabinet Fans, Fan Sections															
Up to 22 in. diameter	All	All	A	2	0.25	A	3	0.75	A	3	0.75	C	3	0.75	4,9
24 in. diameter and up	Up to 2 in. s.p.	Up to 300	B	3	2.50	C	3	3.50	C	3	3.50	C	3	3.50	9
		300 to 500	B	3	0.75	B	3	1.50	C	3	2.50	C	3	2.50	9
		501 and up	B	3	0.75	B	3	1.50	B	3	1.50	B	3	1.50	9
	2.1 in. s.p. and up	Up to 300	C	3	2.50	C	3	3.50	C	3	3.50	C	3	3.50	3,9
		300 to 500	C	3	1.50	C	3	1.50	C	3	2.50	C	3	2.50	3,8,9
		501 and up	C	3	0.75	C	3	1.50	C	3	1.50	C	3	2.50	3,8,9
Centrifugal Fans															
Up to 22 in. diameter	All	All	B	2	0.25	B	3	0.75	B	3	0.75	C	3	1.50	9,19
24 in. diameter and up	Up to 40	Up to 300	B	3	2.50	B	3	3.50	B	3	3.50	B	3	3.50	8,19
		300 to 500	B	3	1.50	B	3	1.50	B	3	2.50	B	3	2.50	8,19
		501 and up	B	3	0.75	B	3	0.75	B	3	0.75	B	3	1.50	8,19
	50 and up	Up to 300	C	3	2.50	C	3	3.50	C	3	3.50	C	3	3.50	2,3,8,9,19
		300 to 500	C	3	1.50	C	3	1.50	C	3	2.50	C	3	2.50	2,3,8,9,19
		501 and up	C	3	1.00	C	3	1.50	C	3	1.50	C	3	2.50	2,3,8,9,19
Propeller Fans															
Wall-mounted	All	All	A	1	0.25	A	1	0.25	A	1	0.25	A	1	0.25	
Roof-mounted	All	All	A	1	0.25	A	1	0.25	B	4	1.50	D	4	1.50	
Heat Pumps	All	All	A	3	0.75	A	3	0.75	A	3	0.75	A/D	3	1.50	
Condensing Units	All	All	A	1	0.25	A	4	0.75	A	4	1.50	A/D	4	1.50	
Packaged AH, AC, H and V Units															
All	Up to 10	All	A	3	0.75	A	3	0.75	A	3	0.75	A	3	0.75	19
	15 and up to 4 in. s.p.	Up to 300	A	3	0.75	A	3	3.50	A	3	3.50	C	3	3.50	2,4,8,19
		301 to 500	A	3	0.75	A	3	2.50	A	3	2.50	A	3	2.50	4,19
		501 and up	A	3	0.75	A	3	1.50	A	3	1.50	A	3	1.50	4,19
	15 and up, 4 in. s.p. and up	Up to 300	B	3	0.75	C	3	3.50	C	3	3.50	C	3	3.50	2,3,4,8,9
		301 to 500	B	3	0.75	C	3	1.50	C	3	2.50	C	3	2.50	2,3,4,9
		501 and up	B	3	0.75	C	3	1.50	C	3	1.50	C	3	2.50	2,3,4,9
Packaged Rooftop Eqmt.	All	All	A/D	1	0.25	D	3	0.75	See Reference Note No: 17						5,6,8,17
Ducted Rotating Equipment															
Small fans, fan-powered boxes	Up to 600 cfm	All	A	3	0.50	A	3	0.50	A	3	0.50	A	3	0.50	7
	601 cfm and up	All	A	3	0.75	A	3	0.75	A	3	0.75	A	3	0.75	7
Engine-Driven Generators	All	All	A	3	0.75	C	3	1.50	C	3	2.50	C	3	3.50	2,3,4

Base Types:
A. No base, isolators attached directly to equipment (Note 27)
B. Structural steel rails or base (Notes 28 and 29)
C. Concrete inertia base (Note 30)
D. Curb-mounted base (Note 31)

Isolator Types:
1. Pad, rubber, or glass fiber (Notes 20 and 21)
2. Rubber floor isolator or hanger (Notes 20 and 25)
3. Spring floor isolator or hanger (Notes 22, 23, and 25)
4. Restrained spring isolator (Notes 22 and 24)
5. Thrust restraint (Note 26)

Figure 20.—Selection guide for vibration isolation.

Notes for Table 42: Selection Guide for Vibration Isolation

These notes are keyed to the column titled *Reference Notes* in Table 42 and to other reference numbers throughout the table. Although the guide is conservative, cases may arise where vibration transmission to the building is still excessive. If the problem persists after all short circuits have been eliminated, it can almost always be corrected by increasing isolator deflection, using low-frequency air springs, changing operating speed, improving rotating component balancing, or, as a last resort, changing floor frequency by stiffening or adding more mass. The assistance of a qualified vibration consultant can be very useful in resolving these problems.

Note 1. Isolator deflections shown are based on a reasonably expected floor stiffness according to floor span and class of equipment. Certain spaces may dictate higher levels of isolation. For example, bar joist roofs may require a static deflection of 1.5 in. over factories, but 2.5 in. over commercial office buildings.

Note 2. For large equipment capable of generating substantial vibratory forces and structure-borne noise, increase isolator deflection, if necessary, so isolator stiffness is less than one-tenth the stiffness of the supporting structure.

Note 3. For noisy equipment adjoining or near noise-sensitive areas, see the section on Mechanical Equipment Room Sound Isolation.

Note 4. Certain designs cannot be installed directly on individual isolators (Type A), and the equipment manufacturer or a vibration specialist should be consulted on the need for supplemental support (Base Type).

Note 5. Wind load conditions must be considered. Restraint can be achieved with restrained spring isolators (Type 4), supplemental bracing, snubbers, or limit stops. Also see Chapter 54, Seismic and Wind Resistant Design.

Note 6. Certain types of equipment require a curb-mounted base (Type D). Airborne noise must be considered.

Note 7. See section on Resilient Pipe Hangers and Supports for hanger locations adjoining equipment and in equipment rooms.

Note 8. To avoid isolator resonance problems, select isolator deflection so that resonance frequency is 40% or less of the lowest normal operating speed of equipment (see Chapter 7 in the 2001 *ASHRAE Handbook—Fundamentals*).

Note 9. To limit undesirable movement, thrust restraints (Type 5) are required for all ceiling-suspended and floor-mounted units operating at 2 in. of water or more total static pressure.

Note 10. Pumps over 75 hp may need extra mass and restraints.

Note 11. See text for full discussion.

Isolation for Specific Equipment

Note 12. Refrigeration Machines: Large centrifugal, hermetic, and reciprocating refrigeration machines may generate very high noise levels; special attention is required when such equipment is installed in upper-story locations or near noise-sensitive areas. If equipment is located near extremely noise-sensitive areas, follow the recommendations of an acoustical consultant.

Note 13. Compressors: The two basic reciprocating compressors are (1) single- and double-cylinder vertical, horizontal or L-head, which are usually air compressors; and (2) Y, W, and multihead or multicylinder air and refrigeration compressors. Single- and double-cylinder compressors generate high vibratory forces requiring large inertia bases (Type C) and are generally not suitable for upper-story locations. If such equipment must be installed in an upper-story location or at-grade location near noise-sensitive areas, the expected maximum unbalanced force data must be obtained from the equipment manufacturer and a vibration specialist consulted for design of the isolation system.

Note 14. Compressors: When using Y, W, and multihead and multicylinder compressors, obtain the magnitude of unbalanced forces from the equipment manufacturer so the need for an inertia base can be evaluated.

Note 15. Compressors: Base-mounted compressors through 5 hp and horizontal tank-type air compressors through 10 hp can be installed directly on spring isolators (Type 3) with structural bases (Type B) if required, and compressors 15 to 100 hp on spring isolators (Type 3) with inertia bases (Type C) weighing 1 to 2 times the compressor weight.

Note 16. Pumps: Concrete inertia bases (Type C) are preferred for all flexible-coupled pumps and are desirable for most close-coupled pumps, although steel bases (Type B) can be used. Close-coupled pumps should not be installed directly on individual isolators (Type A) because the impeller usually overhangs the motor support base, causing the rear mounting to be in tension. The primary requirements for Type C bases are strength and shape to accommodate base elbow supports. Mass is not usually a factor, except for pumps over 75 hp, where extra mass helps limit excess movement due to starting torque and forces. Concrete bases (Type C) should be designed for a thickness of one-tenth the longest dimension with minimum thickness as follows: (1) for up to 30 hp, 6 in.; (2) for 40 to 75 hp, 8 in.; and (3) for 100 hp and up, 12 in.

Pumps over 75 hp and multistage pumps may exhibit excessive motion at start-up ("heaving"); supplemental restraining devices can be installed if necessary. Pumps over 125 hp may generate high starting forces; a vibration specialist should be consulted.

Note 17. Packaged Rooftop Air-Conditioning Equipment: This equipment is usually on lightweight structures that are susceptible to sound and vibration transmission. The noise problem is further compounded by curb-mounted equipment, which requires large roof openings for supply and return air.

The table shows Type D vibration isolator selections for all spans up to 20 ft, but extreme care must be taken for equipment located on spans of over 20 ft, especially if construction is open web joists or thin, lightweight slabs. The recommended procedure is to determine the additional deflection caused by equipment in the roof. If additional roof deflection is 0.25 in. or less, the isolator should be selected for 10 times the additional roof deflection. If additional roof deflection is over 0.25 in., supplemental roof stiffening should be installed to bring the roof deflection down below 0.25 in., or the unit should be relocated to a stiffer roof position.

For mechanical units capable of generating high noise levels, mount the unit on a platform above the roof deck to provide an air gap (buffer zone) and locate the unit away from the associated roof penetration to permit acoustical treatment of ducts before they enter the building.

Some rooftop equipment has compressors, fans, and other equipment isolated internally. This isolation is not always reliable because of internal short-circuiting, inadequate static deflection, or panel resonances. It is recommended that rooftop equipment be isolated externally, as if internal isolation was not used.

Note 18. Cooling Towers: These are normally isolated with restrained spring isolators (Type 4) directly under the tower or tower dunnage. High-deflection isolators proposed for use directly under the motor-fan assembly must be used with extreme caution to ensure stability and safety under all weather conditions. See Note 5.

Note 19. Fans and Air-Handling Equipment: The following should be considered in selecting isolation systems for fans and air-handling equipment:

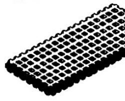
1. Fans with wheel diameters of 22 in. and less and all fans operating at speeds to 300 rpm do not generate large vibratory forces. For fans operating under 300 rpm, select isolator deflection so the isolator natural frequency is 40% or less than the fan speed. For example, for a fan operating at 275 rpm, 0.4 × 275 = 110 rpm. Therefore, an isolator natural frequency of 110 rpm or lower is required. This can be accomplished with a 3 in. deflection isolator (Type 3).
2. Flexible duct connectors should be installed at the intake and discharge of all fans and air-handling equipment to reduce vibration transmission to air duct structures.
3. Inertia bases (Type C) are recommended for all Class 2 and 3 fans and air-handling equipment because extra mass permits the use of stiffer springs, which limit heaving movements.
4. Thrust restraints (Type 5) that incorporate the same deflection as isolators should be used for all fan heads, all suspended fans, and all base-mounted and suspended air-handling equipment operating at 2 in. or more total static pressure. Restraint movement adjustment must be made under normal operational static pressures.

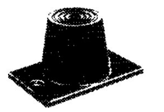
Figure 20a.—Selection guide for vibration isolation.

Vibration Isolators: Materials, Types, and Configurations

Notes 20 through 31 include figures to assist in evaluating commercially available isolators for HVAC equipment. The isolator selected for a particular application depends on the required deflection, life, cost, and compatibility with the associated structures.

RUBBER PADS (Type 1)

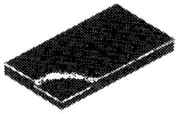




RUBBER MOUNTS (Type 2)

Note 20. Rubber isolators are available in pad (Type 1) and molded (Type 2) configurations. Pads are used in single or multiple layers. Molded isolators come in a range of 30 to 70 durometer (a measure of stiffness). Material in excess of 70 durometer is usually ineffective as an isolator. Isolators are designed for up to 0.5 in. deflection, but are used where 0.3 in. or less deflection is required. Solid rubber and composite fabric and rubber pads are also available. They provide high load capacities with small deflection and are used as noise barriers under columns and for pipe supports. These pad types work well only when they are properly loaded and the weight load is evenly distributed over the entire pad surface. Metal loading plates can be used for this purpose.

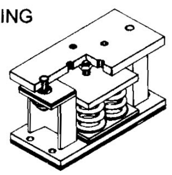
GLASS FIBER PADS (Type 1)



Note 21. Precompressed glass fiber isolation pads (Type 1) constitute inorganic inert material and are available in various sizes, thicknesses of 1 to 4 in., and capacities of up to 500 psi. Their manufacturing process ensures long life and a constant natural frequency of 7 to 15 Hz over the entire recommended load range. Pads are covered with an elastomeric coating to increase damping and to protect the glass fiber. Glass fiber pads are most often used for isolation of concrete foundations and floating floor construction.

SPRING ISOLATOR (Type 3)

RESTRAINED SPRING ISOLATOR (Type 4)



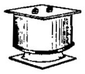
Note 22. Steel springs are the most popular and versatile isolators for HVAC applications because they are available for almost any deflection and have a virtually unlimited life. All spring isolators should have a rubber acoustical barrier to reduce transmission of high-frequency vibration and noise that can migrate down the steel spring coil. They should be corrosion-protected if installed outdoors or in a corrosive environment. The basic types include

1. **Note 23.** *Open spring isolators* (Type 3) consist of a top and bottom load plate with an adjustment bolt for leveling. Springs should be designed with a horizontal stiffness of at least 100% of the vertical stiffness to ensure stability verified by a minimum ratio of 0.8 for the diameter divided by the deflected spring height.

2. **Note 24.** *Restrained spring isolators* (Type 4) have hold-down bolts to limit vertical movement. They are used with (a) equipment with large variations in mass (boilers, refrigeration machines) to restrict movement and prevent strain on piping when water is removed, and (b) outdoor equipment, such as cooling towers, to prevent excessive movement because of wind load. Spring criteria should be the same as open spring isolators, and restraints should have adequate clearance so that they are activated only when a temporary restraint is needed.

3. *Housed spring isolators* consist of two telescoping housings separated by a resilient material. Although not preferred, if this type is used, care should be taken in design and installation to prevent binding and short-circuiting.

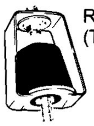
AIR SPRINGS



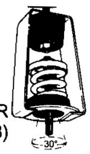
ROLLING LOBE

BELLOWS

Air springs can be designed for any frequency, but are economical only in applications with natural frequencies of 1.33 Hz or less (6 in. or greater deflection). They do not transmit high-frequency noise and are often used to replace high-deflection springs on problem jobs. A constant air supply (with an air dryer) is required.

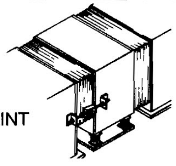


RUBBER HANGER (Type 2)



SPRING HANGER (Type 3)

Note 25. Isolation hangers (Types 2 and 3) are used for suspended pipe and equipment and have rubber, springs, or a combination of spring and rubber elements. Criteria should be the same as for open spring isolators. To avoid short-circuiting, hangers should be designed for 20° to 35° angular hanger rod misalignment. Swivel or traveler arrangements may be necessary for connections to piping systems subject to large thermal movements.



THRUST RESTRAINT (Type 5)

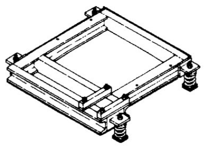
Note 26. Thrust restraints (Type 5) are similar to spring hangers or isolators and are installed in pairs to resist the thrust caused by air pressure.

DIRECT ISOLATION (Type A)

Note 27. Direct isolation (Type A) is used when equipment is unitary and rigid and does not require additional support. Direct isolation can be used with large chillers, packaged air-handling units, and air-cooled condensers. If there is any doubt that the equipment can be supported directly on isolators, use structural bases (Type B) or inertia bases (Type C), or consult the equipment manufacturer.

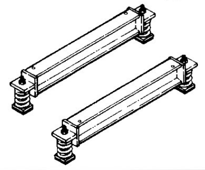
Figure 21.—Vibration isolators.

STRUCTURAL BASES (Type B)



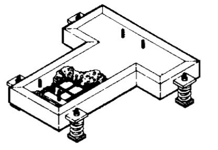
Note 28. Structural bases (Type B) are used where equipment cannot be supported at individual locations and/or where some means is necessary to maintain alignment of component parts in equipment. These bases can be used with spring or rubber isolators (Types 2 and 3) and should have enough rigidity to resist all starting and operating forces without supplemental hold-down devices. Bases are made in rectangular configurations using structural members with a depth equal to one-tenth the longest span between isolators, with a minimum depth of 4 in. Maximum depth is limited to 12 in., except where structural or alignment considerations dictate otherwise.

STRUCTURAL RAILS (Type B)



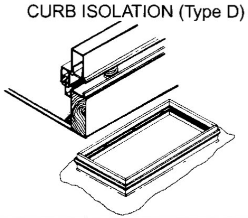
Note 29. Structural rails (Type B) are used to support equipment that does not require a unitary base or where the isolators are outside the equipment and the rails act as a cradle. Structural rails can be used with spring or rubber isolators and should be rigid enough to support the equipment without flexing. Usual practice is to use structural members with a depth one-tenth of the longest span between isolators with a minimum depth of 4 in. Maximum depth is limited to 12 in. except where structural considerations dictate otherwise.

CONCRETE BASES (Type C)



Note 30. Concrete bases (Type C) are used where excess heaving motion may otherwise occur with spring isolators. They consist of a steel pouring form usually with welded-in reinforcing bars, provision for equipment hold-down, and isolator brackets. Like structural bases, concrete bases should be rectangular or T-shaped and, for rigidity, have a depth equal to one-tenth the longest span between isolators with a minimum of 6 in. Base depth need not exceed 12 in. unless specifically required for mass, rigidity, or component alignment.

CURB ISOLATION (Type D)



Note 31. Curb isolation systems (Type D) are specifically designed for curb-supported rooftop equipment and have spring isolation with a watertight and airtight curb assembly. The roof curbs are narrow to accommodate the small diameter of the springs within the rails, with static deflection in the 1 to 3 in. range to meet the design criteria described in Type 3. Flap-type rubber or metal seals are preferable to a continuous sponge at the perimeter to minimize binding and short-circuiting.

systems; they should not be expected to substitute for conventional pipe vibration isolators.

To accommodate pressure thrust, flexible connectors require an end restraint, which is either (1) added to the connector, (2) incorporated by its design, (3) added to the piping system (anchoring), or (4) built in by the stiffness of the system. Connector extension caused by pressure thrust on isolated equipment should also be considered when flexible connectors are used. Overextension will cause failure. Manufacturers' recommendations on restraint, pressure, and temperature limitations must be strictly observed.

Hose Connectors

Hose connectors accommodate lateral movement perpendicular to length and have very limited or no axial movement capability. Rubber hose connectors can be of molded or hand wrapped construction with wire reinforcing and are available with metal-threaded end fittings or integral rubber flanges. Application of threaded fittings should be limited to 3 in. and smaller pipe diameter. The fittings should be the mechanically expanded type to minimize the possibility of pressure thrust blowout. Flanged types are available in larger pipe sizes. Table 43 lists recommended lengths.

Metal hose is constructed with a corrugated inner core and a braided cover, which helps attain a pressure rating and provides end restraints that eliminate the need for supplemental control assemblies. Short lengths of metal hose or corrugated metal bellows, or pump connectors, are available without braid and have built-in control assemblies. Metal hose is used to control misalignment and vibration rather than noise and is used primarily where temperature or the pressure of flow media precludes the use of other material. Table 43 provides recommended lengths.

Expansion Joint or Arched Connectors

Expansion joint or arched connectors have one or more convolutions or arches and can accommodate all modes of axial, lateral, and angular movement and misalignment. When made of rubber, they are commonly called expansion joints, spool joints, or spherical connectors; in PTFE, they are known as couplings or expansion joints.

Rubber expansion joints or spool joints are available in two basic types: (1) hand-wrapped with wire and fabric reinforcing, and (2) molded with fabric and wire or with high-strength fabric only

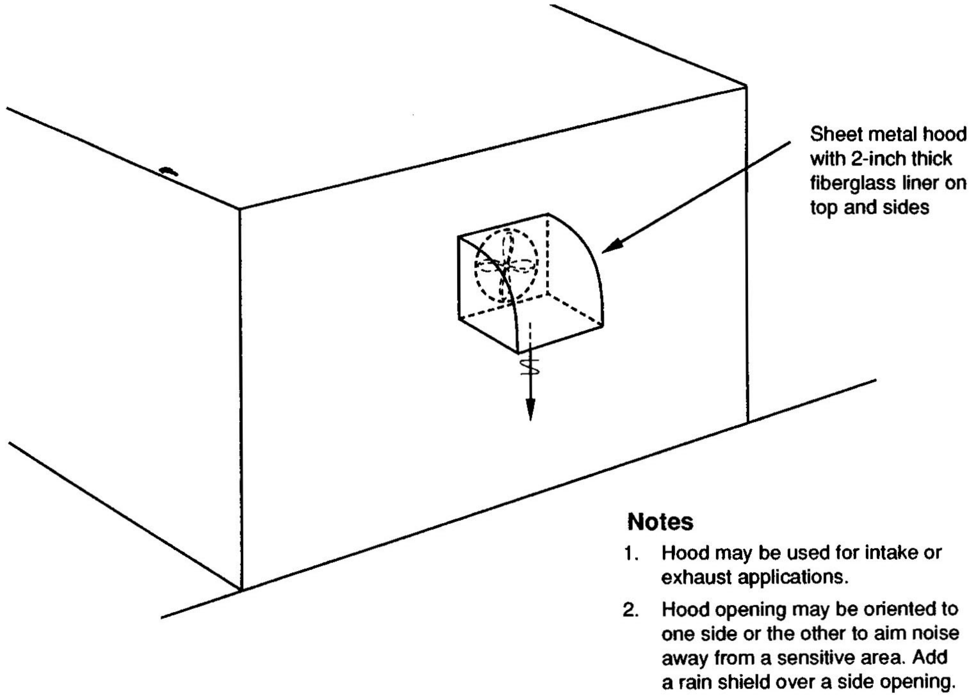
Table 43 Recommended Live Length[a] of Flexible Rubber and Metal Hose

Nominal Diameter, in.	Length,[b] in.	Nominal Diameter, in.	Length,[b] in.
0.75	12	4	18
1	12	5	24
1.5	12	6	24
2	12	8	24
2.5	12	10	24
3	18	12	36

[a] Live length is end-to-end length for integral flanged rubber hose and is end-to-end less total fitting length for all other types.

[b] Per recommendations of Rubber Expansion Division, Fluid Sealing Association.

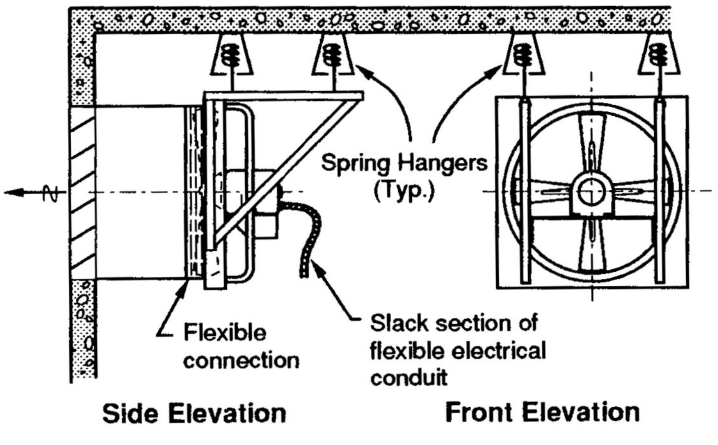
Figure 21a.—Vibration isolators.



Notes

1. Hood may be used for intake or exhaust applications.
2. Hood opening may be oriented to one side or the other to aim noise away from a sensitive area. Add a rain shield over a side opening.

Figure 22.—Lined hood for propeller fan noise control.



NOTE:
Position hangers on line of center of gravity of fan unit. Supplemental sections of steel angle or channel may be secured to fan mounting frame, as required, for support.

Figure 23.—Vibration isolation suspension for propeller fans.

d. **Clearance.**—Provide adequate clearances between unducted equipment and walls. Clearances are usually based on the fan wheel diameter of the equipment. Provide minimum clearance of one diameter at the inlet and 1.5 diameters at the fan outlet. When the mechanical room serves as intake plenum for AHU units, the clearance between the unit's air intake and the wall should be equal to the height of the unit.

e. **Transitions.**—When possible, fan/AHU discharge transition should not exceed a total angle of 15 degrees.

f. **Discharge Duct.**—Provide a minimum of three duct diameters between the fan/AHU inlet and outlet connections and any duct fittings or duct mounted equipment. When space constraints require installation of an elbow within 1.5 diameters of the fan discharge, use long radius elbows without turning vanes and add the appropriate pressure loss due to system effect as discussed elsewhere in this manual.

g. **Speed.**—Select propeller fans, axial fans and vaneaxial fans for the lowest possible speed.

2. **Ductwork**

a. **Velocity.**— Maintain duct velocities as noted in section F.5 below.

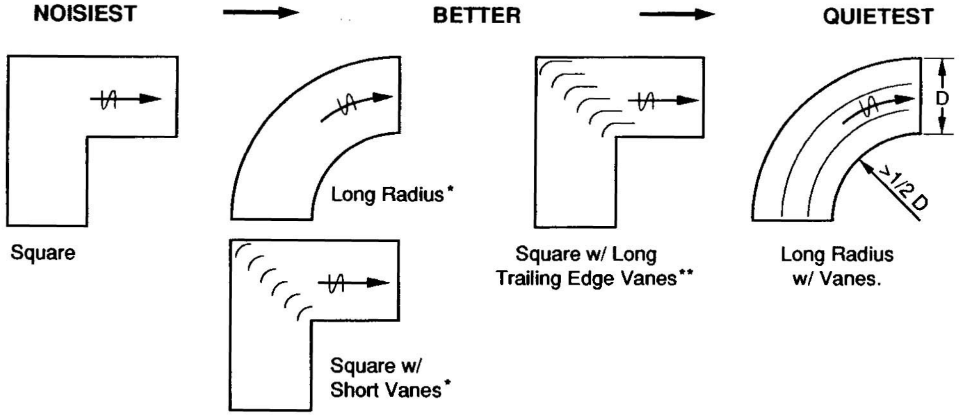
b. **Elbows.**— For maximum noise reduction, use long radius elbows with full radius turning vanes. Long radius elbows without vanes and square elbows with short vanes are satisfactory for most applications. Square elbows without vanes should be avoided. See figure 24.

c. **Turning Vanes**.—The maximum recommended length for single turning vanes is 36-inches. When longer vanes are required, use the double thickness type. Use turning vanes at all diverging T connections.

d. **Branch Takeoffs**.—Radius or bevel branch takeoffs are preferable to Y or straight takeoffs. See figure 25.

e. **Tees**.—See figure 26.

f. **Offsets**.—Gradual offsets (15 degree maximum) are preferable to Z-offsets with vanes. Avoid Z-offsets without vanes. See figure 27.



* Airflow velocity and proximity of upstream and downstream fittings and fans determine which type is preferable.

** Trailing edge length should be at least 3 times the vane spacing.

Figure 24.—Guidelines for minimizing regenerated noise in elbows.

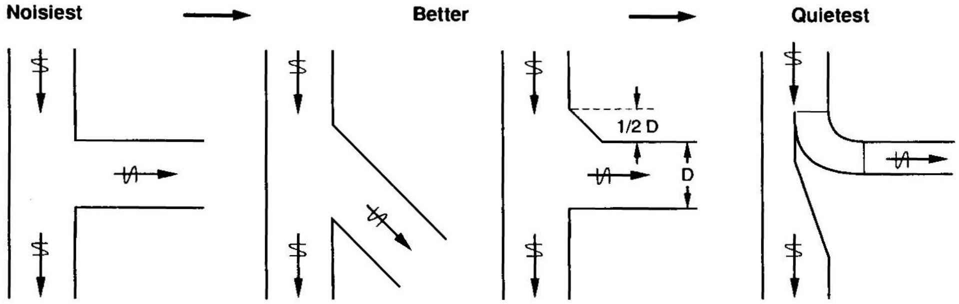


Figure 25.—Guidelines for minimizing regenerated noise in takeoffs.

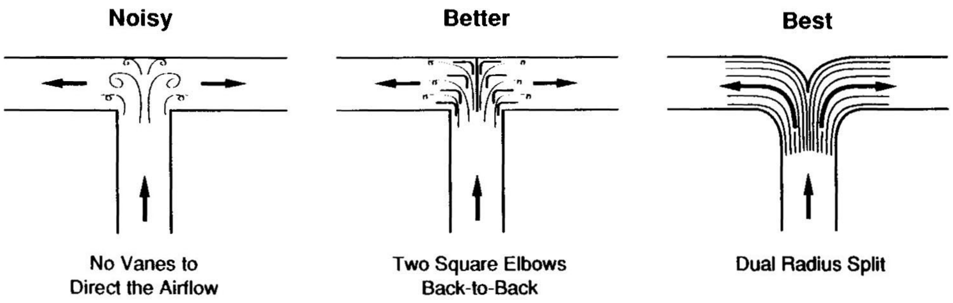


Figure 26.—Guidelines for minimizing regenerated noise in duct tees.

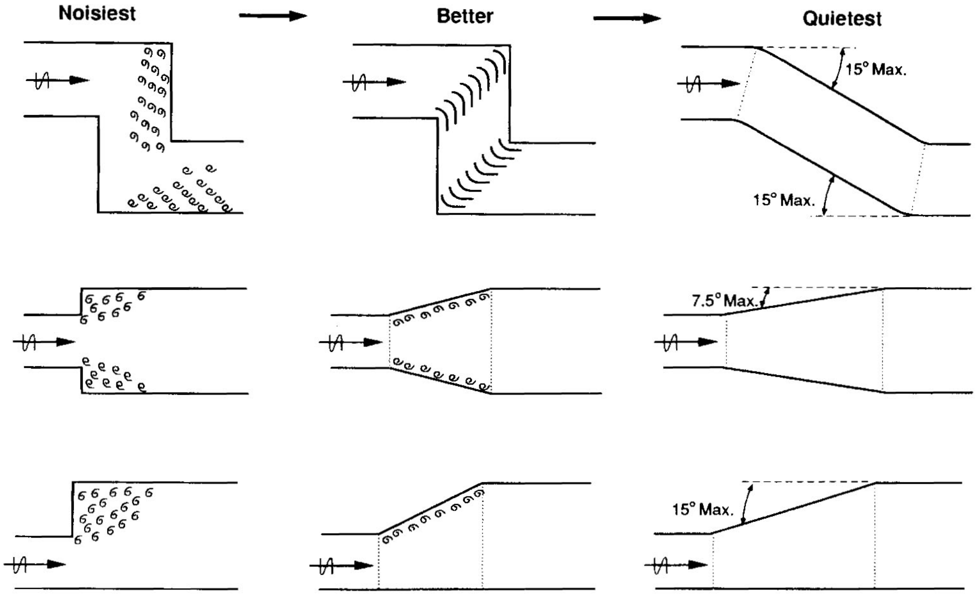


Figure 27.—Guidelines for minimizing regenerated noise in transitions and offsets.

g. **Transitions**.—Provide gradual transitions with 7.5 degree side angle on each side or 15 degree maximum angle on one side. Avoid sudden or sharp edge expansions. See figure 27.

h. **Clearance**.—The clearance between ductwork and walls or ceilings should be equal to 10 percent of the longest side of the duct but in no case less than 6 inches.

i. **Other**.—Figures 28 through 30 illustrate many of the recommendations noted above.

3. **Air Terminal Devices (ATDs)**

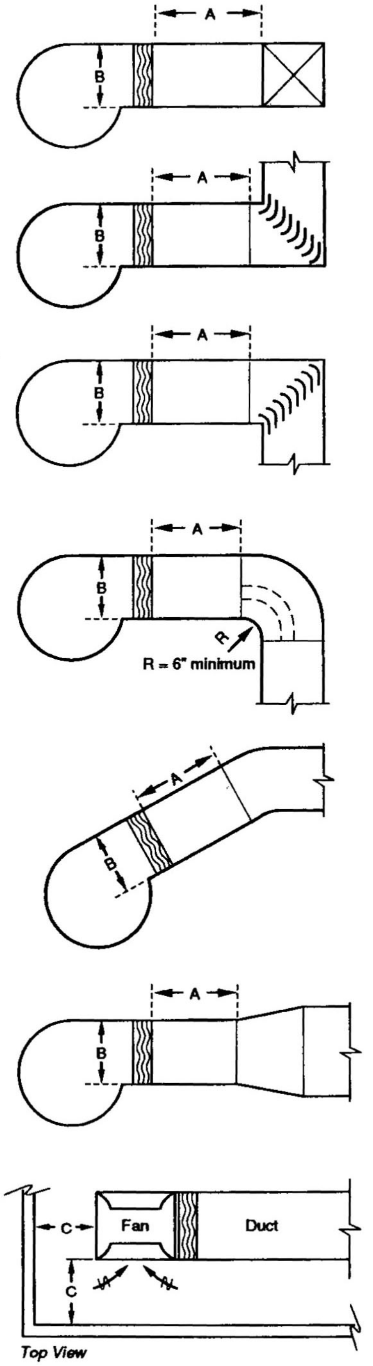
a. **Noise Criteria (NC)**.—Select ATDs for NC 45.

b. **Elbows.—**Avoid elbows at ATDs. Provide three duct diameters of straight duct upstream of the ATDs.

c. **Balancing Dampers.—**When balancing dampers are necessary, install them a minimum of three duct diameters upstream of the ATDs.

4. **Sound Attenuation**.—Good design practice should eliminate the need for use of sound attenuation equipment. The following recommendations are offered to aid in correcting existing problems where system designs failed to adequately account for noise generated by HVAC equipment. In some cases the noise may exit into sensitive areas through openings in the plant walls instead of mechanical room air intake and exhaust louvers. With the exception of acoustical louvers, all the following have been used at Reclamation facilities.

Fan/Duct Layout | **Comment**



Bad if A < 3 B

Bad if A < 3 B
Very Bad if turning vanes are deleted

Fair if A < 3 B
Bad if A < 1.5 B

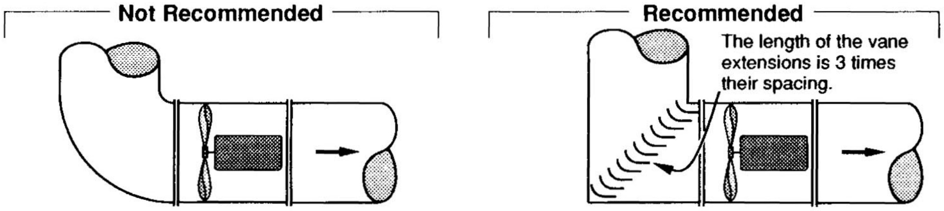
Good if A > 3 B
Fair if A > 1.5 B
Bad if A < 1.5 B
(Delete turning vanes if A < 1.5 B)

Very good if A > 3 B
Fair if A < 3 B

Best if A > 1.5 B
Transition wall slopes of 1:7 preferred.
Slopes of 1:4 permitted if inlet velocity is less than 2000 FPM.

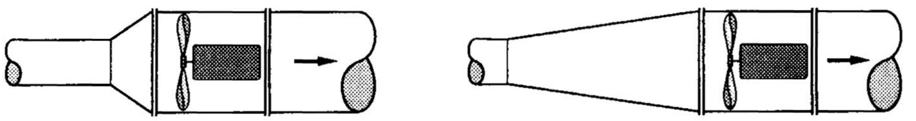
Best if C > 1.5 the fan wheel diameter
Good if C = fan wheel diameter
Poor if C < fan wheel diameter

Figure 28.—Guidelines for centrifugal fan installations.



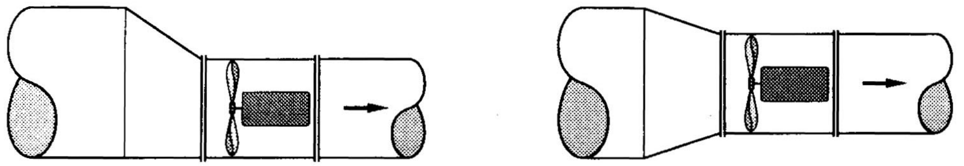
Upstream radius elbow creates imbalance at inlet.

Square inlet elbow with extended trailing edge vanes delivers less turbulent airflow to fan inlet.



Abrupt inlet transition causes turbulence.

Gradual (1:7) expansion of inlet duct avoids impeller turbulence.



Assymmetrical transition creates imbalanced load on fan, with excess turbulence and noise.

Symmetrical transition balances load on fan, which minimizes turbulence and noise.

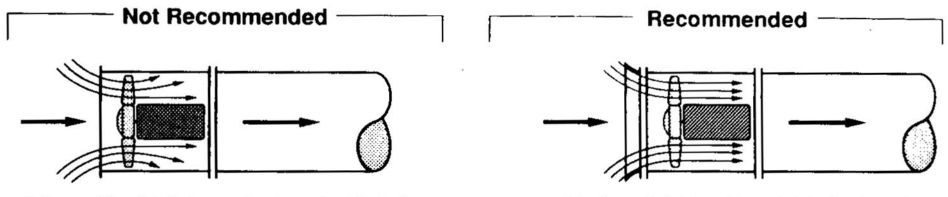
Motor upstream of impeller increases turbulence and noise.

Motor downstream from impeller minimizes turbulence and noise.

Slack or offset flexible connection causes turbulent air flow.

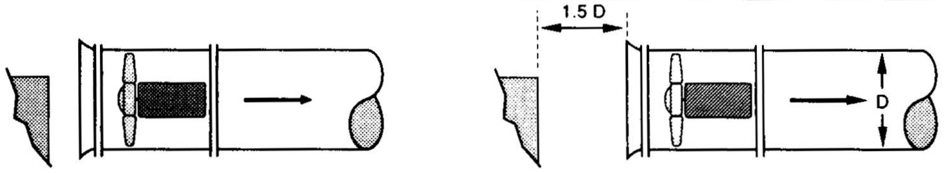
Taut, in-line flexible connections provide optimal vibration isolation without creating turbulence.

Figure 29.—Guidelines for ducted axial flow fan installations.



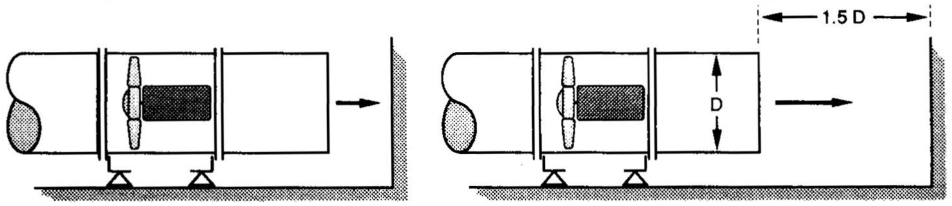
A flanged fan inlet starves the impeller blade tips of air, reducing performance and increasing noise.

A bellmouth inlet guides air into the impeller blade tips.



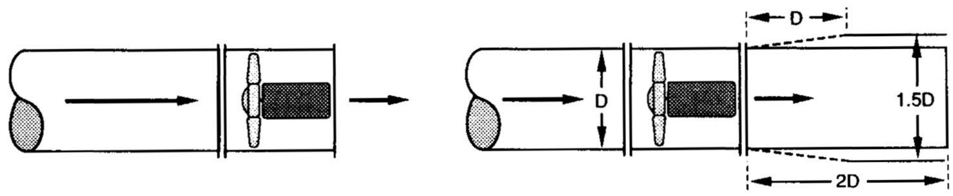
Avoid obstructions close to fan inlets. Part of the impeller is starved for air.

Allow a space of a least 1.5D at the fan inlet.



Avoid obstruction close to the fan outlet.

Allow a space of a least 1.5D at the fan outlet.



Avoid having the fan terminate the discharge end of a system.

Add a duct length of 2D, or an outlet cone of 1.5D at the discharge end of the fan. Extra ducting after the cone is better.

Figure 30.—Guidelines for unducted axial flow fan installations.

a. **Duct Liner.**—Install internal duct liner in first 10-feet of discharge ductwork and return duct connected to a fan, air handling unit, or air conditioning units. One inch liner provides good high frequency sound attenuation but has little or no effect on the low frequencies. For better results at high and low frequencies use 2- or 4-inch duct liner. Table 9 provides recommended duct liner thickness based on the largest duct cross section dimension (length or width):

Table 9.—Recommended duct liner

Duct dimension (inches)	Liner thickness (inches)
Up to 28	1
29 to 48	2
49 to 68	3
69 and larger	4

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b. **Sound Traps**.—Sound traps provide the best acoustical performance however, they can have significant pressure drops. Figure 31 identifies recommended locations for installation of sound traps.

c. **Acoustical Louvers**.—Acoustical louvers can be used when noise may exit into sensitive areas. Designers should verify that recommended louver free area velocities are maintained to prevent moisture entrainment.

d. **Plenum Liners.—**Where space is available, plenum liners 2- to 6-inch thick provide very good sound attenuation over the complete sound spectrum.

e. **Flow Paths.—**Labyrinth airflow paths with acoustical liners can be created to attenuate sound that would escape through louvers into surrounding sensitive areas. Figure 32 illustrates a typical labyrinth configuration.

F. **PLANT HVAC DESIGN GUIDELINES.**—The following guidelines are offered to assist designers with little or no experience in plant HVAC system design. The material presented is only a starting point. The material is based on the experience of previous designers and accepted industry codes and standards. In many cases, especially minimum ventilation requirements, the material presented is subject to change due to periodic revisions of codes and standards. The information or materials presented in these guidelines are not intended to circumvent the requirements of applicable codes or standards. The HVAC system designer is ultimately responsible for

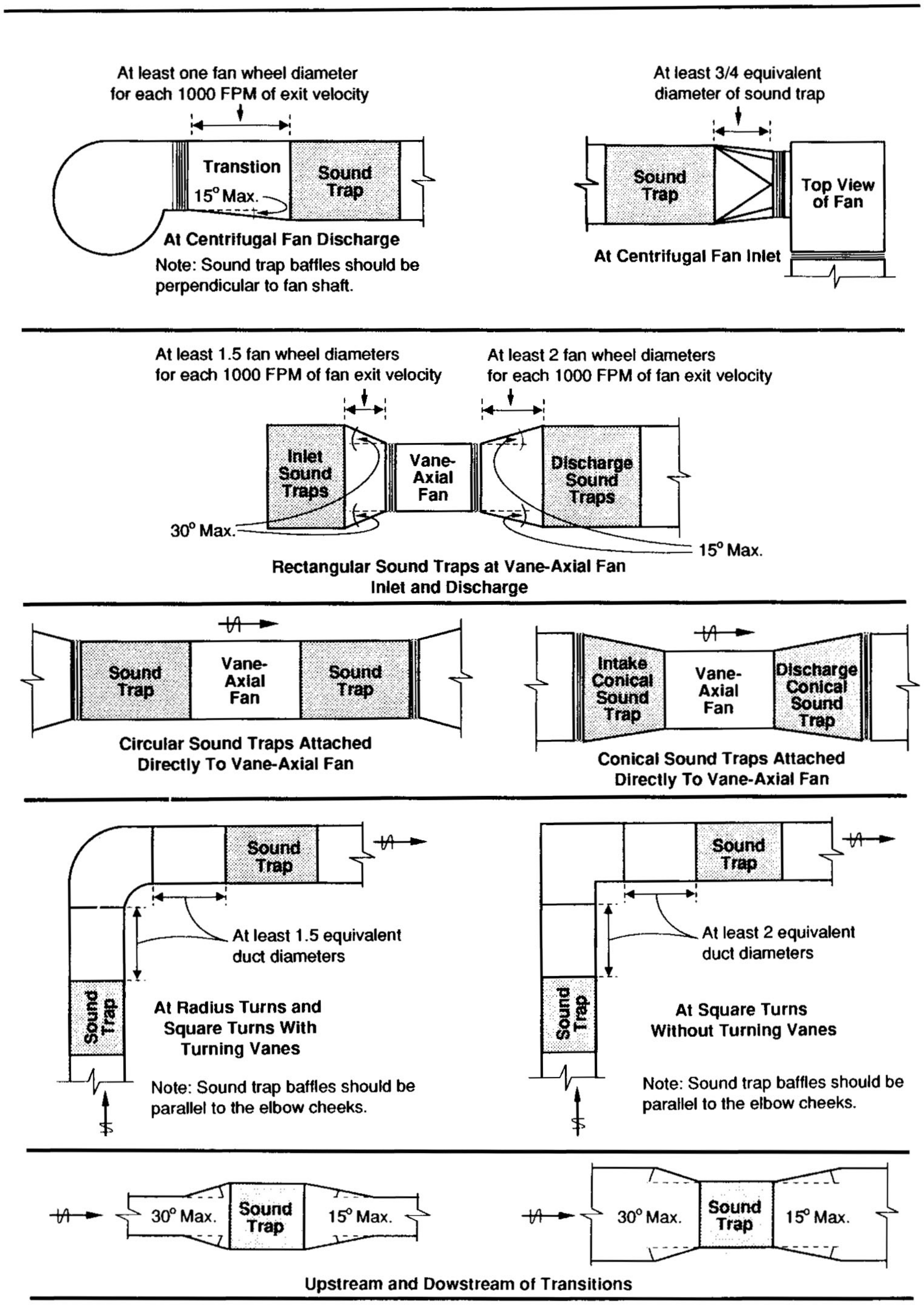


Figure 31.—Guidelines for sound trap placement near fans and duct fittings.

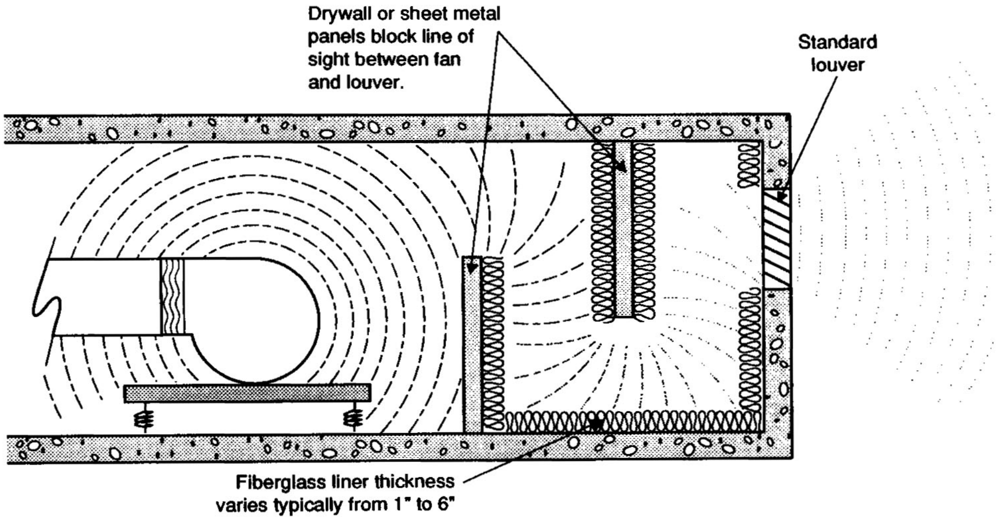


Figure 32.—Labyrinth air path used for sound attenuation.

verifying compliance with the requirements of the latest codes and standards. Where codes and standards differ from these guidelines, the more stringent requirement should be applied in the design.

1. **Human Comfort Health and Safety Applications.**—When human comfort, health, and safety are the primary concern, the HVAC system must control air temperature, humidity, cleanliness, and air distribution to maintain an acceptable environment for personnel. The following conditions should be maintained for the level of activity noted.

   a. **Low Activity and Sedentary Work**

      (1) Temperature above 68 ºF and below 78 ºF.

      (2) Humidity above 30 percent and below 70 percent relative humidity.

      (3) Cleanliness above 85 percent.

      (4) Air motion above 50 ft/min and below 75 ft/min.

      (5) Provide minimum outdoor air ventilation of 15 cfm per person in non smoking areas and 60 cfm per person in smoking areas.

b. **High Activity or Maintenance Work**

(1) Temperature above 45 ºF and below 85 ºF.

(2) Humidity above 20 percent and below 80 percent relative humidity.

(3) Cleanliness above 85 percent.

(4) Air motion above 100 ft/min and below 300 ft/min.

(5) Provide minimum outdoor air ventilation of 15 cfm per person or as required to control contaminants in the space, whichever is higher.

2. **Equipment Protection Applications.**—When equipment protection is the major objective, the air temperature, humidity, cleanliness, and air distribution must be controlled to protect equipment from freezing, corrosion, and high temperatures.

a. **Standing Water or Water Pipes.**—Where equipment contains standing water, or water in small pipes, the ambient temperature should be maintained above 45 ºF.

b. **Oil and Chemical Storage.**—In areas where equipment contains chlorine, oils, and solvents, the ambient temperature should be maintained above 60 ºF and provide minimum ventilation.

c. **Motor Rooms**.—In areas containing motors maintain space temperature below 104 ºF.

d. **Computer Rooms**

(1) Temperature between 72 ºF and 78 ºF.

(2) Humidity between 45 percent and 55 percent relative humidity.

(3) Cleanliness above 90 percent.

(4) Provide minimum outdoor air ventilation of 15 cfm per person.

e. Control Rooms

(1) Temperature between 68 ºF and 80 ºF.

(2) Humidity between 40 percent and 80 percent relative humidity.

(3) Cleanliness above 90 percent.

(4) Provide minimum outdoor air ventilation of 15 cfm per person.

3. Mechanical Equipment Rooms

a. **Location.**—Mechanical equipment rooms should be located away from areas intended for human occupancy such as offices, control rooms, lunch rooms, and computer rooms. Locate to allow reasonable access to outside air and exhaust air louvers.

b. **Floor Space.—**Allow a minimum of 10 to 15 ft^2 of floor space per 1000 cfm of airflow from fans or air handling units. Allow a minimum of 12 to 15 ft of ceiling height.

c. Equipment Clearances.

(1) Allow a minimum of 3 feet of clearance between the equipment and walls. Increase space according to equipment manufacturer's recommendations.

(2) Allow a minimum of 4 feet of clearance in front of electric power and control panel doors. In retrofits, minimum clearances listed in NEMA codes must be provided.

4. Louvers

a. **Style.**—Most Reclamation plant designs specify louvers with 4-inch frame depth, 4-inch blade spacing, and 45º blade angle, and 45 percent free area. However, the louver free areas may vary considerably depending on the size, blade spacing, blade angle, and dimensional orientation. Louvers with dimensions yielding gross areas less than 4 ft^2 will generally have free areas of 35 to 37 percent. For any given dimensions, a taller louver will usually have a larger free area than a wider louver. If higher free areas are required, louvers with wider blade spacing, i.e. 6-inch or smaller, blade angle of 30º can be used.

b. Area and Velocity

(1) ***Outdoor air gross area velocity.*****—**To estimate wall openings required, stationary louver should be sized for a maximum airflow velocity of 400 ft/min.

(2) ***Outdoor air free area.*****—**To prevent moisture entrainment, reduce noise, and minimize pressure loss, the free area velocity should not exceed 800 to 1100 ft/min and a maximum pressure drop of 0.15-in. w.g.

(3) ***Exhaust air.*****—**Size stationary louvers should be sized for a maximum free area velocity of 500 to 600 ft/min and a maximum pressure drop of 0.25-in. w.g.

c. Location.—Locate louvers a minimum of 2 feet above ground level to reduce clogging by debris.

d. Construction

(1) ***Frames and blades.*****—**When louvers are to be installed near ground level or where they may be subject to vandalism, use 16 gage steel construction. Louvers located high enough from the ground, and not subject to vandalism may be aluminum.

(2) ***Screen.*****—**Provide galvanized or stainless steel insect or bird screens on all louvers. Screen may be specified as a single number, i.e. ½ mesh, or two numbers, such as 2 x 2 mesh. The single number (½) format corresponds to the center-to-center distance in inches between the wires. The two digit (2 x 2) format corresponds to the number of openings per inch of screen. Locate insect screens on the outside face of the louver to prevent insect and bird nests. Where bird screens are deemed more desirable than insect screens; use 2 x 2 x 0.063 diameter wire screens. This screen has a free area of approximately 76 percent. The pressure loss due to the reduced free area of the screen is not included in the louver static pressure and must be added to the system losses.

5. Air Flow Velocities

a. Air System Supply

(1) ***Main ducts.***—up to 3000 ft/min not to exceed a friction loss of 0.25 inch w.g. per 100 feet. Normally the duct size is based on 0.1 inch w.g. per 100 feet of duct.

(2) ***Branch ducts***.—up to 2100 ft/min not to exceed a friction loss of 0.2 inch w.g. per 100 feet of duct. Normally the duct size is based on 0.1 inch w.g. per 100 feet of duct.

(3) ***Discharge velocities into rooms***.—vary from 600 ft/min to 1800 ft/min, depending on the air terminal device deflection pattern (0º, 22–1/2º, or 45º) and desired throw (maximum of 3/4 of the distance to the opposite wall). The Noise Criteria (NC) rating of air terminal devices should not be ignored when selecting grilles and registers. NC ratings should not exceed 45.

b. Exhaust System Velocities.—Exhaust velocities, commonly referred to as transport velocities, depend on the type of material to be conveyed by the ductwork. Table 10 identifies the particle sizes and characteristics of various materials commonly conveyed in exhaust systems. Exhaust velocities for the most common applications in Reclamation facilities are provided below. For applications not identified below refer to table 11. For applications where ducts do not require a minimum transport velocity, a pressure drop of 0.10-in/100-ft of duct is normally used to size ducts.

(1) ***Chlorine rooms.***—3,600 ft/m

(2) ***Oil storage rooms.***—3,000 ft/m

(3) ***Toilet rooms.***—1,200 ft/m

(4) ***Paint storage rooms and booths.*****—**2,000 ft/min minimum.

(5) ***Welding fumes.*****—**2,000 ft/min minimum.

(6) ***Battery rooms.*****—**2,000 ft/min minimum.

Table 10.—Types of air contaminants

Airborne materials	Size range µm	Characteristics
Dust	0.10–30.0	Generated by pulverization or crushing of solids. Typical examples are rocks, metal, wood, and coal dust. Particle may be up to 300–400 µm but those above 20–30 µm usually do not remain airborne.
Fumes	0.001–1.0	Small solid particles created by condensation from vapor state, especially volatilized metals as in welding. Fumes tend to coalesce into larger particles as small fume particles collide.
Mists	0.01–10.0	Suspended liquid particles formed by condensation from gaseous state or by dispersion of liquids. Mists occur above open surface electroplating tanks.
Smokes	0.01–1.0	Aerosol mixture form incomplete combustion of organic matter. The size range does not include fly ash.
Vapors	0.005	Gaseous forms of materials that re liquids or solids at room temperature. Many solvents generate vapors.
Gases	0.0005	Materials that do not usually exist as solids or liquids at room temperature, such as carbon monoxide and ammonia. Under sufficient pressure and/or temperature they may be changed into liquids or solids.

McDermott, Henry James: Handbook of Ventilation for Contaminant Control, 1985.

Table 11.—Range of minimum duct design velocities

Nature of contaminant	Examples	Design velocity – ft/min
Vapors, gases, smoke	All vapors, gasses and smoke	Any desired velocity. (Economic optimum velocity usually 1,000 to 2,000 ft/min)
Fumes	Welding	2,000 to 5,000
Very fine light dust	Cotton lint, wood flour, litho powder	2,500 to 3,000
Dry dust and powders	Fine rubber dust, bakelite molding powder dust, cotton dust, light shavings	3,000 to 4,000
Average industrial dust	Grinding dust, granite dust, silica flour, brick cutting,	3,500 to 4,000
Heavy dusts	Sawdust (heavy and wet), metal turnings, sandblast dust	4,000 to 4,500

ACGIH: Industrial Ventilation, 1992

6. **Ventilation Requirements**.—Whenever possible, the airflow requirements should be based on the actual hazard conditions encountered in the space. The air change method is satisfactory for standard or commonly occurring situations. However, caution should be exercised when using the air change method to determine ventilation rates for any application, especially those not identified below. Where significant amounts of hazardous substances are generated in work areas or occupied spaces, the number of air changes may not provide adequate control.

   The sources for the ventilation rates tabulated below have been provided. Because ventilating requirements may change, the sources should be consulted to ensure the recommended ventilation rates reflect current requirements. Where the multiple ventilation criteria are shown, i.e. air changes/hr, cfm/ft^2, and cfm,–the criteria resulting in the highest ventilation requirement should be used.

   a. **Oil Storage and Oil Transfer Rooms.**—Provide 1 cfm/ft^2 of floor area but not less than 150 cfm/min. Ventilate continuously (see *NFPA 850*). Maintain a slight negative pressure in the room. All air from the space must be direct exhausted to outdoors. Provide a fan airflow switch and indicating lights (green–on, red–off) to verify fan operation. Locate monitoring lights and fan motor starter outside the room near entry door to verify fan operating status.

   b. **Toilets, Locker Rooms, and Showers**.—Operate fan when space is occupied and for 15 minutes thereafter. Maintain slight negative pressure. All air from the space must be direct exhausted to outdoors. For each condition below, ventilate at rate producing the highest cfm (see *ASHRAE Standard 62*).

      (1) ***Toilet spaces.***—2 cfm/ft^2 of floor space; 25 cfm/toilet; 200 cfm minimum.

      (2) ***Locker rooms***.—1 cfm/ft^2 of floor space for coat hanging or clean clothes change room. 3 cfm/ft^2 for sweaty or wet clothing

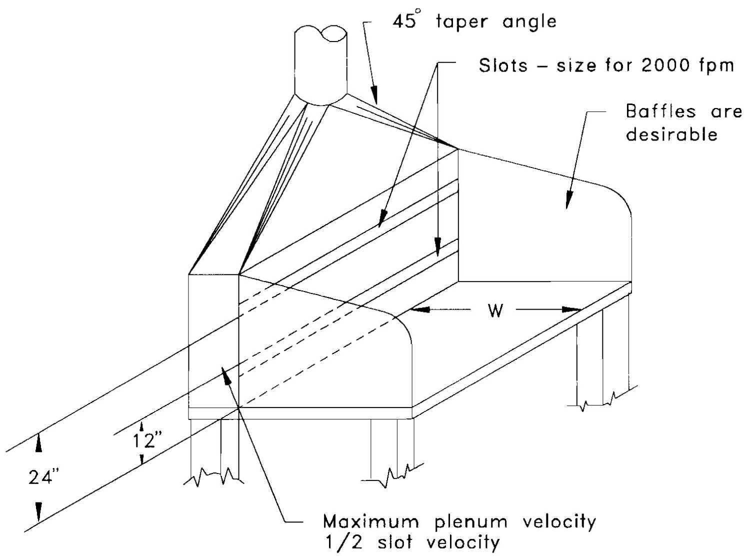
      (3) ***Showers.***—2 cfm/ft^2 of floor space; 50 cfm/shower head minimum; 200 cfm minimum.

c. **Paint Storage Rooms.—**Provide 10 to 12 air changes per hour. Ventilate continuously. Maintain slight negative pressure by making exhaust air equal to 105 percent of supply air (see *American Conference of Governmental Industrial Hygienists (ACGIH) Industrial Ventilation – A Manual of Recommended Practice)*, all air from the space must be direct exhausted to outdoors. Provide fan airflow switch, and indicating lights (green–on, red–off) to verify fan operation. Locate lights and fan motor starter outside room near entry door.

d. **Paint Booths.—**Ventilating air requirements vary depending on type and size of booth, i.e. walk-in type or operator outside, and type of equipment used, i.e. air or airless. Consult *ACGIH* for specific requirements. Ventilate at 150 cfm/min per ft^2 but not less than 20 air changes per minute for preliminary estimates. All air from the space must be direct exhausted to outdoors. Operate fan when booth is occupied. Provide fan airflow switch and indicating lights (green–on, red–off) to verify fan operation. Locate lights and fan motor starter outside room near entry door.

e. **Welding Rooms or Areas.**—Actual amount of air depends on welding rod size and rate of consumption, number of welders used, and type of exhaust system – local or general. For local exhaust systems with fixed bench hoods, see figure 33. For fixed overhead exhaust systems with movable local exhaust hoods, the ventilation rate depends on the distance between the welding rod and the face of the hood, see figure 34. Where local exhaust is not possible, general ventilation rates vary between 1,000 to 4,500 ft^3/min per welder depending on welding rod size see figure 34. Typical welding rod sizes used at Reclamation facilities are 1/4-inch or less. Therefore, a general ventilation rate of 2,500 to 3,500 cfm/welder or 6 to 12 air changes per hour in a welding room should be satisfactory. Provide one portable welding exhaust system with air cleaners capable of removing particulates and odors, and recirculating air.

f. **Battery Rooms**.— Provide continuous ventilation to maintain hydrogen gas concentration below 0.8 percent by volume during maximum gas generation conditions. The following should be noted when determining the required exhaust fan size:



Q = 350 cfm/ft of hood length
Hood length = required working space
W = 24" maximum, if W>24", see Chapter 3
Minimum duct velocity = 2000 fpm
$h_e = 1.78\ VP_s + 0.25\ VP_d$

General ventilation, where local exhaust cannot be used:

Rod, diam.	cfm/welder
5/32	1000
3/16	1500
1/4	3500
3/8	4500

or

A. For open areas, where welding fume can rise away from the breathing zone: cfm required = 800 x lb/hour rod used

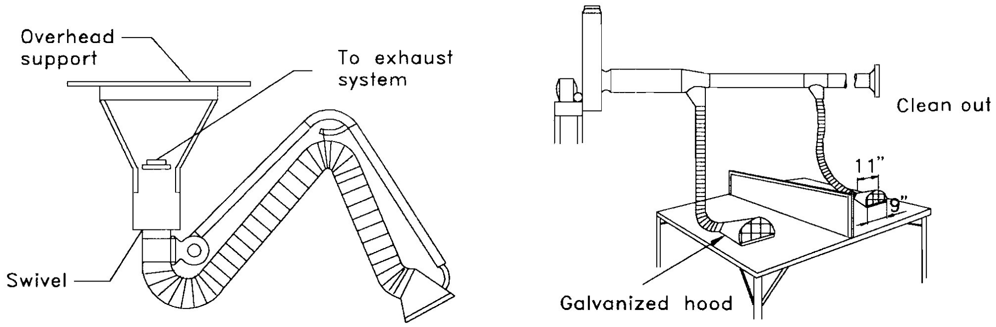
B. For enclosed areas or positions where fume does not readily escape breathing zone: cfm required = 1600 x lb/hour rod used

For toxic materials, higher air flows are necessary, and operator may require respiratory protection equipment.

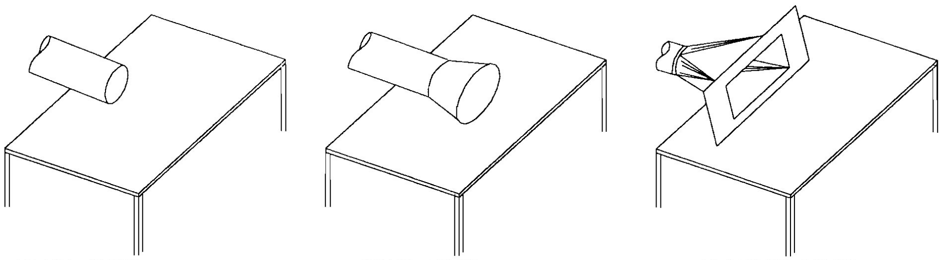
Other types of hoods:
Local exhaust: See VS-90-02
Booth: For design, see VS-90-30
$Q = 100\ \text{cfm/ft}^2$ of face opening
MIG welding may require precise air flow control.

AMERICAN CONFERENCE OF GOVERNMENTAL INDUSTRIAL HYGIENISTS	*WELDING VENTILATION BENCH HOOD*
DATE *1-91*	FIGURE *VS-90-01*

Figure 33.—Weld ventilation hood.



FLEXIBLE EXHAUST CONNECTIONS



RATE OF EXHAUST

X,	Plain duct, cfm	Flange or cone, cfm
Up to 6	335	250
6-9	755	560
9-12	1335	1000

Face velocity = 1500 fpm
Minimum duct velocity = 3000 fpm
Plain duct entry loss = 0.93 VP_d
Flange or cone entry loss = 0.25 VP_d

X= distance of [illegible] to face of hood

Notes:

1. Locate work as close as possible to hood.
2. Hoods perform best when located to the side of the work.
3. Ventilation rates may be inadequate for toxic materials.
4. Velocities above 100–200 fpm may disturb shield gas.

GENERAL VENTILATION, where local exhaust cannot be used :

Rod, diam.	cfm/welder
5/32	1000
3/16	1500
1/4	3500
3/8	4500

OR

A. For open areas where welding fume can rise away from the breathing zone:
cfm required = 800 x lb/hour rod used

B. For enclosed areas or positions where fume does not readily escape breathing zone:
cfm required = 1600 x lb/hour rod used

For toxic materials, higher air flows are necessary and operator should use respiratory protection equipment.

Other types of hoods:
Bench, see VS-90-01
Booth, for design see VS-90-30
Q = 100 cfm/ft^2 of face opening

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INDUSTRIAL HYGIENISTS

WELDING VENTILATION
MOVABLE EXHAUST HOODS

DATE *1-91*
FIGURE *VS-90-02*

Figure 34.—Welding ventilation movable exhaust hoods.

(1) When the battery is fully charged, each charging ampere supplied to the cell produces about 0.016 cubic feet of hydrogen per hour from each cell. This rate of production applies at sea level, when the ambient temperature is about 77 ºF, and when the electrolyte is "gassing or bubbling."

(2) Number of battery cells and maximum charging rate (not float rate) can be obtained from specifications or field inspection.

(3) Hydrogen gas lower explosive limit is 4 percent by volume. Good practice dictates a safety factor of 5, which reduces the critical concentration to 0.8 percent by volume. This large safety factor is to allow for hydrogen production variations with changes in temperature, battery room elevation, and barometric pressure and also allows for deterioration in ventilation systems.

The following example illustrates the procedure for determining battery room ventilation requirements. Assume a battery room volume (V_r) of 900 ft^3, a 60-cell battery with a charge rate of 50 amps per hour, and a maximum H_2 concentration of 0.8 percent by volume.

The total H_2 generation is given by:

$$G_t = G_c N A$$

Where: G_t = total hydrogen generated, ft^3/hr
G_c = hydrogen generation per cell, ft^3/hr/cell
N = number of cells
A = charging rate, amps

$$G_h = (0.016)(60)(50) = \underline{48\ ft^3/hr}$$

The maximum acceptable volume of H_2 is given by:

$$V_h = V_r C$$

Where: V_h = volume of hydrogen, ft^3
V_r = total room volume, ft^3
C = acceptable hydrogen concentration, percent

$$V_h = (900)(0.008) = \underline{7.2\ ft^3}$$

The time, T, to reach critical concentration is given by:

$$T = V_h/G_h = 7.2/48 = \underline{0.15 \text{ hrs or } 9 \text{ minutes}}$$

The minimum number of air changes per hour, N, is determined by:

$$N = 60/9 = \underline{6.7 \text{ changes/hr}}$$

The minimum airflow, Q, required, is given by:

$$Q = V_r N/60 = (900)(6.7)/60 = \underline{100 \text{ cfm}}$$

Ventilation rates for battery rooms are usually small (less than 100 cfm). For preliminary sizing of ventilating requirements, assume 1 cfm/ft^2, or 6 air changes/hr. Maximum hydrogen gas generation occurs when batteries are approaching full charge.

Locate air intakes near floor. Locate exhaust openings near ceilings. Locate air intakes and exhaust openings so that they are diametrically opposed if possible. Avoid routing miscellaneous ducts through battery rooms.

Ventilate continuously with a dedicated exhaust fan. Maintain slight negative pressure. The fan should be spark resistant construction and should be located outside the room if possible. If the fan must be located in the battery room, the fan, controls etc. must comply with *NEC* explosion-proof criteria that will increase cost significantly. Fan controls should not be designed for automatic fan shutdown by the fire detection and alarm system when fire or smoke is detected outside the room. Locate fan motor starter and control switch outside the room adjacent to entry door. Provide a differential pressure switch and indicating lights (green–on, red–off) outside room but near entry door to verify fan operating status.

g. **Chlorine Storage Rooms Inside the Structure.—**Ventilate with 100 percent outdoor air. Provide 15 air changes per hour continuously and 60 air changes per hour when occupied by maintenance personnel. Position air intake near floor and no higher than one duct diameter. Provide an external hazard warning sign, and an internal gas detection device with a red indicating light outside the room or enclosure.

h. **Sewage Rooms.**—Provide 12 air changes per hour continuously (from *NFPA 820*). Exhaust directly to outdoors.

i. **Plant Sumps**.—Provide 12 air changes per hour continuously. (from *NFPA 820*). Exhaust directly to outdoors.

j. **All Other Floors, Rooms and Galleries.**—Ventilate at 1 cfm/ft^2 of floor area or as dictated by the heating or cooling requirements for each space.

k. **Any Room with Combustible Gases.**—The following methods can be used to determine the time required for ventilation, the number of air changes required to dilute a know concentration, and the time to reach a combustible concentration level. The equations, procedures, and examples are extracted from *NFPA 69*:

(1) ***Time required for ventilation.***—The time required to reduce a given concentration of an explosive vapor to a safe concentration can be obtained by solving the following equation for time t:

$$\ln (C/C_o) = (-Q/V)\, Kt$$

Where: C = desired gas concentration, percent
C_o = initial gas concentration, percent
Q = airflow rate, cfm
V = volume of room, ft^3
K = mixing efficiency of ventilating scheme.
t = time required to reach the desired concentration, min.

Values for K are shown in table 12, however, most authorities recommend a K value no greater than 0.25.

Table 12.—Mixing efficiency for various ventilation arrangements

Method of supplying	**Efficiency (K)[1] values**	
	Single exhaust opening	**Multiple exhaust openings**
No positive supply		
1. Infiltration through cracks	0.2	0.3
2. Open doors or windows	0.2	0.4
Forced air supply		
1. Grilles and registers	0.3	0.5
2. Diffusers	0.5	0.7
3. Perforated ceiling	0.8	0.9

[1] Few data exist on defining the degree of mixing. Most authorities recommend a K value no greater than 0.25. NFPA 69.

Example: A 1000 ft^3 room contains a gasoline vapor concentration of 20 percent by volume. Determine the time required to reduce the concentration to the lower flammable limit (LFL) of 1.4 percent if the space is ventilated with 2,000 cfm air.

$$\ln (1.4/20) = (-2{,}000/1{,}000)\ (0.20)\ t$$

$$t = \underline{6.65 \text{ minutes}}$$

If the concentration was reduced to 25 percent of the LFL (0.35 percent concentration), the time required would be 10 minutes. 0.20 mixing efficiency is used as a conservative design assumption.

(2) ***Air changes to inert.***—The number of air changes required for entering a combustible gas is obtained by solving the following equation for N:

$$\ln (C/C_o) = - KN$$

Where: C, C_o, *and* K are as defined above.
N = number of air changes

Example: Using the previous example, determine the number of air changes required to reach the LFL.

$$\ln (1.4/20) = (0.20)\, N$$

$N =$ $\underline{2 \text{ changes per minute}}$

Therefore, the time required to achieve 13.3 air changes is 13.3/2, which is exactly equal to the 6.65 minutes determined in the previous example.

(3) ***Time to reach a concentration.***—Time to reach a given concentration can be obtained by rewriting equation 2 as follows:

$$C = (G/Q)(1 - e^{-KN})$$

This equation can be rewritten as:

$$-KN = \ln [1 - (CQ/G)]$$

Where: C = target concentration, percent
G = combustible gas entering space, cfm
Q = air entering space, cfm
K = mixing efficiency factor
N = number of theoretical air changes

Example: Assume a gas leak, $Q_{L,}$ of 100 ft^3/min of a 15 percent combustible gas/air mixture into a 1000 ft^3 room. Determine the time to reach a 5 percent concentration throughout the room.

Based on the statement of the problem, the amount of gas entering the space is 15 percent of the total leak rate or 0.15(100) = $\underline{15\ ft^3/min}$. The amount of air is 100–15 = $\underline{85\ ft^3/min}$. Assume K = 0.2.

The number of air changes required to achieve a 5 percent concentration of the combustible gas is determined as follows:

$$-0.02N = \ln [1 - (0.05(85)/15)]$$

$$N = \underline{1.67}$$

Since the room volume and leakage rate are known, the time to reach the expected concentration is determined as follows:

$$t = (V/Q_L)\ N = (1000/100)1.67 = \underline{16.7\ \text{min.}}$$

7. **Recommended Space Temperatures.—**Recommended ambient space temperatures are shown on table 13.

8. **Recommended Relative Humidity**.—Recommended relative humidity values are shown on table 14. In general, Reclamation plants are considered unmanned structures and no attempt is made to control humidity through addition of water to the supply air. Computer rooms are the exception where minimum and maximum humidity levels are critical and closely controlled. Maximum humidity is controlled when using evaporative cooling systems because of the significant amount of moisture added to cool the supply air.

9. **Building Pressurization**.—Negative pressures may cause unsatisfactory conditions in the plant. See table 15. When appropriate, air conditioning systems should be designed to pressurize the structure to restrict the entry of dust, and control temperature and humidity due to drafts through door cracks and other openings. The amount of outdoor air induced into the building through the main air conditioning units must be equal to the amount of contaminated air exhausted from the main air handling systems and all continuously operating exhaust fans.

Table 13.—Recommended space temperatures

Space	Minimum		Maximum	
	ºF	ºC	ºF	ºC
Oil storage and transfer rooms	45	7.2	80	26.6
Toilets and janitor rooms	65	18.3	85	29.4
Paint storage rooms	45	7.2	80	26.6
Paint booths	65	18.3	80	26.6
Welding rooms	65	18.3	80	26.6
Battery rooms	50	10	80	26.6
Chorine storage indoors	45	7.2	90	32.2
Generator room level	45	7.2	80	26.6
Generator room ceiling level			120	48.9
Pipe galleries	45	7.2	80	26.6
Electrical galleries	45	7.2	104	40
Visitor facilities	68	20	80	26.6
Control rooms	68	20	80	26.6
Computer rooms	72	22.2	78	25.5
UPS rooms	65	18.3	90	32.2
Lunch rooms	68	20	80	26.6
Offices	68	20	80	26.6
HVAC machinery	55	12.8	90	32.2
Air compressor and pump rooms	45	7.2	90	32.2

Table 14.—Recommended relative humidity

Spaces	Minimum	Maximum
Pipe Galleries		80
Electrical Galleries	40	80
Visitors facilities	30	80
Control rooms	40	80
Computer rooms	45	55
Offices	30	70
Lunch rooms	30	70
Intake air supply		80

Table 15.—Unsatisfactory conditions due to negative pressures within buildings

Negative pressure inch w.g.	Adverse conditions which may result
0.01 to 0.02	Worker draft complaints – High velocity drafts through doors and windows
0.01 to 0.05	Natural draft stacks ineffective – Ventilation through roof exhaust ventilators, flow through stacks with natural draft greatly reduced
0.02 to 0.05	Carbon monoxide hazard – Back drafting will take place in hot water heaters, unit heaters, and other combustion equipment not provided with induced draft
0.03 to 0.10	General mechanical ventilation reduced – Airflows reduced in propeller fans and low pressure supply and exhaust fans
0.05 to 0.10	Doors difficult to open – Serious injury may result from nonchecked, slamming doors
0.10 to 0.25	Local exhaust ventilation impaired – Centrifugal fan fume exhaust flow reduced

ASHRAE Handbook: HVAC Applications, 1991.

a. **Recommended Pressure.**—To provide positive pressure within the building, the amount of outdoor air should be increased to approximately 105 percent of the amount exhausted or be balanced to give a positive gage building pressure difference of 0.08 to 0.16 inch of water.

b. **Effect of Pressure on Door Opening Forces.**—Positive pressure may affect occupants' ability to open doors or may prevent doors from completely closing; therefore, excessive positive pressure should be avoided. According to the *NFPA 101 Life Safety Code*, the forces required to fully open any door manually in a means of egress shall not exceed 30 lbs to set the door in motion. For additional discussions concerning the effects of positive building pressurization and procedures for calculating door opening forces refer to *NFPA 92A*.

10. Water Piping

a. **Design Guides.**—Water distribution systems should be designed in accordance with the *UPC (Uniform Plumbing Code)* and the *ASHRAE Handbook of Fundamentals*.

b. **Materials.**—Copper piping conforming to *ASTM B33* Type K should be used to prevent corrosion. Water flow velocities in copper piping should not exceed 5 ft/sec to prevent erosion.

c. **Dielectric Joints.**—Where dissimilar metal piping must be connected, insulating flange sets or insulating unions must be provided between the piping.

d. **Valve Strainers.**—Control valves should be protected by properly sized strainers immediately upstream of the valve. Provide blow-off valve on strainer.

e. **Freeze Protection.**—Piping located outdoors, or indoors in unheated areas, is subject to freezing. This is especially true for pipe systems containing standing water. Insulation will delay but not prevent freezing of still water or water flowing at a rate insufficient for the available heat content to offset the heat loss. The time required for standing water to cool to 32 °F can be estimated with the following equation from *1997 ASHRAE Fundamental Handbook*:

$$\theta = \rho c_p \pi (D_i/2)^2 (R_T) \ln [(t_i - t_a)/(t_f - t_a)]$$

Where: θ = time for water to cool to freezing, h
ρ = density of water = 62.4 lb/ft^3
c_p = specific heat of water = 1.0 Btu/lb F
D_i = inside diameter of pipe, ft
D_p = outer diameter of pipe or inner diameter of insulation, ft
D_l = outer diameter of insulation, ft
R_T = $R_p + R_l + R_a$ = total resistance: pipe, insulation, air film, ft · °F · h/Btu
R_a = $1/(h_a \pi D_l)$ = resistance between ambient air and outer surface of insulation per foot of pipe, ft · °F · h/Btu
h_a = air heat transfer coefficient
R_l = $\ln(D_l/D_p)/(2\pi k_l)$ = resistance of thermal insulation per foot of pipe, ft · °F · h/Btu
R_p = $\ln(D_p/D_i)/(2\pi k_p)$ = resistance of pipe per foot of pipe, ft · °F · h/Btu ($R_p = 0$ for metal pipe)
k_l = thermal conductivity of insulation, Btu/h · ft · ºF

k_p = thermal conductivity of pipe material, Btu/h ft ºF
t_a = ambient air temperature, ºF
t_i = initial water temperature, ºF
t_f = freezing temperature, ºF

The water flow required to prevent freezing can be estimated as follows:

$$L/W = c_p\,(R_T + R_W)\ \ln\left[\,(R_T)(t_i - t_a) / (R_T + R_W)(t_f - t_a)\right]$$

Where: W = flow rate required to keep pipe free of ice – lb/h
L = length of exposed pipe – ft
R = $1/(\pi k_w Nu)$ = resistance between water and inner surface of pipe per foot of pipe = 0.23 ft · ºF · h/Btu.
k_w = thermal conductivity of water = 0.32 Btuh/h · ft · ºF
Nu = Nusselt number for water = 4.36 for fully developed laminar flow and constant heat flux.

When appropriate, the piping system design should include thermostatically controlled valves to automatically drain the system when a low-limit temperature sensor detects a fall in temperature below a predetermined set point – usually 35 ºF. Once outdoor temperature falls below freezing, cooling water is rarely required to maintain ambient conditions. In these cases, the HVAC controls should include a manual reset to prevent refilling the system after a shutdown initiated by the freeze protection system. Occasionally, some components such as pump casings and strainers cannot be completely drained. If complete draining is not possible, or desirable, the piping and components should be wrapped with electric heat tape. Heat tape and sizing requirements are discussed below.

f. Heat Tracing.—Heat tracing may be accomplished with fluids such as hot water and steam, or electric cable. Due to lack of steam, or hot water in sufficient quantities, heat tracing in most Reclamation facilities is accomplished with electric cables.

(1) *Applications*:

(a) *Freeze protection*.—A self-regulating electric cable is used to prevent freezing of water pipes. This is primary consideration for outdoor piping applications and indoor applications where space heating is not required or is impractical. Most Reclamation applications fall under this category.

(b) *Maintain temperature*.—A constant Watt cable with temperature controls is used to maintain a specific fluid temperature. This application has very limited use in Reclamation facilities because most systems tolerate fairly wide operating ranges.

(c) *Snow and ice melting*.—Cable can be installed to prevent undesirable ice formation on or around equipment.

(2) *Sizing heat cable*:

(a) *Establish design conditions*:

T_m = maintenance temperature, °F
T_s = startup temperature, °F
T_a = ambient temperature, °F
D_p = nominal pipe diameter
Insulation type: See table 16 – K factors for various insulation types
Insulation thickness: inches
Maximum wind speed
Heat loss safety factor
Location of pipe: indoor, outdoor, underground

Table 16.—K factor chart for various insulation types[1]

Temperature (° F)	0	50	100	150	200	250	300	350	400
Fiberglass	0.23	0.25	0.27	0.29	0.32	0.34	0.37	0.39	0.41
Calcium silicate	0.35	0.37	0.40	0.43	0.45	0.47	0.50	0.53	0.55
Urethane	0.18	0.17	0.18	0.22	0.25	–	–	–	–
Cellular glass	0.38	0.40	0.46	0.50	0.55	0.58	0.61	0.65	0.70

[1] Select the K factor equal to or below the maintenance temperature for K_{Tm} or the K factor equal to or below the ambient temperature for K_{Ta}.

(b) *Heat loss from the pipe:*

$$Qp = [2\pi K (T_m - T_a)] \div [z \ln (D_o / D_i)]$$

Where: Q_p = heat loss, Watts/ft of pipe
K = thermal conductivity of insulation, Btu in/hr ft^2 °F
D_o = outside diameter of insulation, inches
D = inside diameter of insulation, inches
z = 40.994 Btu in/W hr ft

Q_p can also be determined from table 17. To determine the heat loss/ft of pipe, multiply the table value by the difference between the maintenance and ambient temperatures $(T_m - T_a)$.

(c) *Correction factors:*

(i) Location.—Note that table 17 is based on outdoor applications. For indoor installation apply a correction factor of 0.9 to heat losses determined by table 17, i.e. multiply Q_p by 0.9. For underground installations (including below frost line), or for plastic pipe, apply a 25 percent safety factor, i.e. multiply by 1.25. For underground plastic pipe apply a 50 percent safety factor to account for poor heat transfer characteristics, i.e. multiply by 1.50.

(ii) Wind speed.—Note that table 17 is based on 20 mph wind velocities. Apply a wind speed correction factor of 5 percent/5 mph wind speed over 20 mph not to exceed 10 percent to Q_p.

Table 17.—Heat losses from insulated metal pipes w/ft of pipe per temperature differential (° F)

Pipe size (IPS)	Insulation thickness (inches)							
	½	¾	1	1 ½	2	2 ½	3	4
½	0.054	0.041	0.035	0.028	0.024	0.022	0.020	0.018
¾	0.063	0.048	0.040	0.031	0.027	0.024	0.022	0.020
1	0.075	0.055	0.046	0.036	0.030	0.027	0.025	0.022
1 ¼	0.090	0.066	0.053	0.051	0.034	0.030	0.028	0.024
1 ½	0.104	0.075	0.061	0.046	0.038	0.034	0.030	0.026
2	0.120	0.086	0.069	0.052	0.043	0.037	0.033	0.029
2 ½	0.141	0.101	0.080	0.059	0.048	0.042	0.037	0.032
3	0.168	0.118	0.093	0.068	0.055	0.048	0.042	0.035
3 ½	0.189	0.113	0.104	0.075	0.061	0.052	0.046	0.038
4	0.210	0.147	0.115	0.083	0.065	0.056	0.050	0.041
4 ½	0.231	0.161	0.125	0.090	0.072	0.061	0.054	0.044
5	0.255	0.177	0.137	0.098	0.078	0.066	0.058	0.047
6	0.300	0.207	0.160	0.113	0.089	0.075	0.065	0.053
7	0.342	0.235	0.181	0.127	0.100	0.084	0.073	0.059
8	0.385	0.263	0.202	0.041	0.111	0.092	0.080	0.064
9	0.427	0.291	0.222	0.156	0.121	0.101	0.087	0.070
10	0.474	0.323	0.247	0.171	0.133	0.110	0.095	0.076
12	0.559	0.379	0.290	0.200	0.155	0.128	0.109	0.087
13	0.612	0.415	0.316	0.217	0.168	0.168	0.118	0.093
16	0.696	0.471	0.358	0.246	0.189	0.155	0.133	0.104
18	0.781	0.527	0.401	0.274	0.210	0.172	0.147	0.115
20	0.865	0.584	0.443	0.302	0.231	0.189	0.161	0.125
24	0.134	0.696	0.527	0.358	0.274	0.226	0.189	0.147
Tank	0.161	0.107	0.081	0.054	0.040	0.032	0.027	0.020

(iii) K factor efficiency.—Multiply Q_p by a correction factor of $2(k_{Tm} + k_{Ta})$ where k_{Tm} and k_{Ta} are the thermal conductivities at the maintenance and ambient temperatures respectively.

(d) *Select a cable.*—Refer to manufacturers product data and select a cable with a heat output that exceeds the heat loss. If the heat loss exceeds the heat output of the cables available, the cable must be wrapped around the pipe as shown in figure 35.

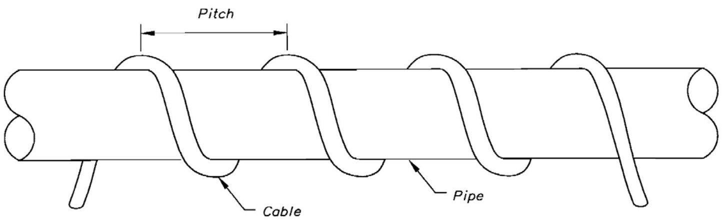
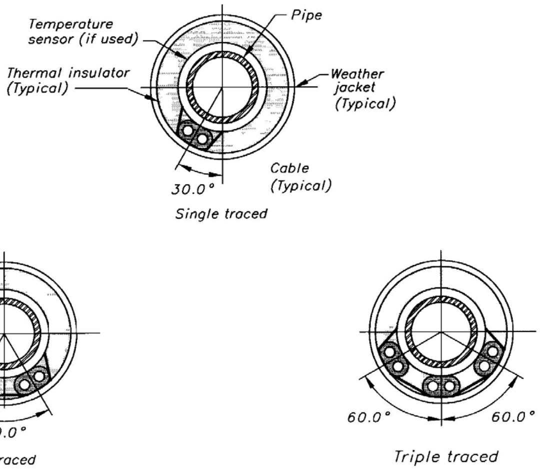


FIGURE A: Spiral–wrap arrangement



Single traced

Duel traced

Triple traced

Figure 35.—Spiral wrap.

(e) Determine length of cable required by adding the following to the length of pipe:

(i) Component allowances for each occurrence of the following items:

Flange pair: 1.5 ft
Pipe support: 2.0 ft
Butterfly valve: 2.5 ft
Ball valve: 2.7 ft
Globe valve: 4.0 ft
Gate valve: 5.0 ft

(ii) Wrapping allowance.—If the cable is wrapped around the pipe, refer to table 18 to determine the additional length of cable and the pitch.

Table 18.—Wrapping factor (feet of cable per foot of pipe)

Pipe size	**Pitch (inches)**																	
	2	**3**	**4**	**5**	**6**	**7**	**8**	**9**	**10**	**11**	**12**	**14**	**16**	**18**	**24**	**30**	**36**	**42**
½	1.90	1.47	1.29	1.19	1.14	1.10	1.08	1.06	—	—	—	—	—	—	—	—	—	—
¾	2.19	1.64	1.40	1.27	1.19	1.14	1.11	1.09	1.07	1.06	—	—	—	—	—	—	—	—
1	2.57	1.87	1.55	1.38	1.27	1.21	1.16	1.13	1.11	1.09	1.07	—	—	—	—	—	—	—
1 ¼	3.07	2.18	1.76	1.53	1.39	1.30	1.24	1.19	1.16	1.13	1.11	1.08	1.06	—	—	—	—	—
1 ½	3.43	2.41	1.92	1.65	1.48	1.37	1.29	1.24	1.20	1.16	1.14	1.10	1.08	1.06	—	—	—	—
2	4.15	2.86	2.25	1.90	1.67	1.52	1.42	1.34	1.28	1.24	1.20	1.15	1.12	1.10	1.05	—	—	—
2 ½	4.19	3.36	2.61	2.17	1.89	1.70	1.56	1.46	1.39	1.33	1.28	1.21	1.17	1.13	1.08	1.05	—	—
3	5.88	3.99	3.06	2.52	2.17	1.93	1.76	1.63	1.53	1.45	1.39	1.30	1.23	1.19	1.11	1.07	1.05	—
4	7.43	5.01	3.82	3.11	2.65	2.33	2.09	1.92	1.78	1.67	1.58	1.45	1.36	1.29	1.17	1.11	1.08	1.06
5	9.09	6.10	4.63	3.75	3.17	2.77	2.47	2.24	2.06	1.92	1.81	1.63	1.51	1.42	1.25	1.17	1.12	1.09
6	10.75	7.20	5.44	4.40	3.70	3.22	2.86	2.58	2.36	2.19	2.04	1.83	1.67	1.55	1.34	1.23	1.16	1.12
8	13.88	9.28	6.99	5.63	4.72	4.08	3.60	3.23	2.94	2.17	2.51	2.22	2.00	1.83	1.53	1.36	1.26	1.20
10	17.20	11.49	8.65	6.94	5.81	5.01	4.41	3.95	3.58	3.28	3.03	2.65	2.37	2.15	1.75	1.52	1.38	1.29
12	20.34	13.58	10.21	8.19	6.85	5.89	5.18	4.62	4.18	3.83	3.53	3.07	2.73	2.40	1.97	1.68	1.51	1.39
14	22.30	14.89	11.18	8.97	7.49	6.44	5.66	5.05	4.57	4.17	3.85	3.34	2.96	2.67	2.11	1.79	1.59	1.46
16	25.44	16.98	12.75	10.22	9.53	7.33	6.42	5.74	5.18	4.72	4.35	3.77	3.33	3.00	2.34	1.97	1.73	1.57
18	28.58	19.07	14.31	11.47	9.57	8.22	7.21	6.42	5.80	5.29	4.86	4.20	3.71	3.33	2.58	2.15	1.88	1.69
20	31.71	21.16	15.88	12.72	10.61	9.11	7.99	7.11	6.42	5.85	5.38	4.64	4.09	3.66	2.82	2.34	2.03	1.81
24	37.99	25.34	19.02	15.22	12.70	10.90	9.55	8.50	7.66	6.98	6.41	5.52	4.85	4.34	3.32	2.72	2.33	2.07

Note: To determine the wrapping factor, divide the calculated heat loss by the heat output of the cable. Locate the value that is equal to or the next highest in the row for the pipe size in your application. The value at the top of the column is the pitch or spacing from center to center of the cable along the pipe.

G. HVAC SYSTEM CONTROLS

1. **General Considerations.**—Most plant heating, ventilating, and air conditioning equipment should be provided with "ON-OFF-AUTO" controls. In the "ON" mode the ventilating fans operate continuously and heating and cooling equipment remain de-energized. In the "AUTO" mode the fan operates continuously but the heating and cooling equipment are automatically controlled by temperature sensors. Typical control equipment includes temperature sensors for: low limit (outside air freeze protection), supply air, mixed air, room air, and high limit (electric heating coil discharge air) temperatures; temperature averaging sensors; humidity sensors; dry bulb and enthalpy economizer controls; control dampers and operators; control valves and operators; and smoke detectors.

   HVAC equipment used in Reclamation plants is usually basic and does not require sophisticated control equipment. The following should be considered when designing or specifying a control system:

   a. **Simplicity.**—Reclamation plants are located in remote areas where a high degree of technical service may not be available. Keep the system as simple as possible to ensure reliability and facilitate repairs. Tight control schemes are not necessary except in computer rooms. Offices should be maintained at normal comfort levels. Plant temperatures are usually allowed to fluctuate between the normal low and high temperature range.

   b. **Energy Recovery.**—Most Reclamation plants are designed for unmanned operation, therefore, the primary HVAC objective is equipment protection. Since wide temperature fluctuations are permitted in Reclamation plants, energy recovery systems and equipment are seldom economically justifiable. However, every design should be evaluated for opportunities to save or recover energy.

c. **Free Cooling.**—Where outdoor winter design temperatures are below 35 ºF (based on the *ASHRAE Handbook of Fundamentals* 97.5 percent criteria) an economizer can provide significant energy savings. Economizers can be controlled by dry bulb sensors or enthalpy controllers.

(1) ***Dry bulb controller.***—A dry bulb temperature controller measures and compares return air and outdoor air temperature. The outside air (OA) and return air (RA) dampers modulate between minimum position and 100 percent OA whenever the OA temperature is below a predetermined high-limit set point, typically 65 to 75 degrees. Once the OA temperature exceeds the high-limit, the economizer reverts to the minimum OA ventilation and maximum RA positions. A psychrometric chart is used to determine the high-limit set point as follows: Assume return conditions of 80 degree db and 40 percent relative humidity, and an average outdoor relative humidity of 70 percent. Locate the RA point on a psychrometric chart. The enthalpy at this point is 29 Btu/lb of dry air. Follow the constant enthalpy line to its intersection with 70 percent outdoor air relative humidity line. Read the high-limit dry bulb temperature of 70 degrees. The actual high-limit should be slightly lower than 70 degrees to account for higher morning relative humidity and sensor drift. Typically, the high-limit set point ranges from 60 to 70 ºF in areas with 60 to 80 percent relative humidity. Since most plants are located in dry climates, dry bulb economizer controls will usually provide satisfactory temperature control.

(2) ***Enthalpy controller.***— Enthalpy controllers require four sensors: a dry bulb sensor and humidity or dew point sensor to measure the enthalpy of each (OA and RA) air stream. The controller selects the airstream with the lowest enthalpy (energy level) to control the economizer. When the outdoor air enthalpy is lower than the indoor enthalpy, the controller maximizes use of outdoor air for free cooling. When the indoor enthalpy is lower, the controller reduces the outdoor air to the minimum ventilation requirements.

Because enthalpy controllers actually measure the total energy level of the two airstreams, better economizer control is achieved, especially in humid climates.

d. **Average Temperature.**—Plants normally have large rooms with high localized heat loads. Random operation of equipment causes wide ambient temperature variations that cannot be properly sensed or controlled with one room or return air temperature sensor. A general rule is to use one sensor for every 5,000 ft^2 of floor space. Usually three or four temperature sensors should be located throughout the room. The average temperature signal from the sensors should be used for control purposes.

e. **Freeze Protection.**—Provide freeze-protection low-limit thermostats or sensors on outside air intake openings or adjacent to water bearing equipment. If necessary the thermostat should open an automatically controlled drain valve to drain water from the system.

f. **High-Limit Thermostats.**—Provide high-limit thermostat or sensor downstream of electric heating coils to de-energize the coils if discharge temperature is too high (135 to 165 ºF).

g. **Step and Proportional Controllers.**—The output of electric heating coils should be controlled through multistage step controllers or proportional controllers. Provide a differential pressure switch or an airflow switch to prevent energizing the coil if airflow is not verified or to de-energize the coil if the airflow fails to maintain the minimum velocity through the coil.

h. **Integral Controls.**—In some cases decisions will be required concerning use of factory furnished integral controls or independent external controls furnished by a control system supplier. Each has its advantages and disadvantages. Many packaged AHUs and AC units are now available with factory installed, programmable, microprocessor based unitary controllers that contain the algorithms necessary for controlling and monitoring the equipment. Factory assembled controllers can reduce design time and facilitate installation of the HVAC equipment. These controllers are especially advantageous when installing a single HVAC unit or multiple identical HVAC units by the same manufacturer.

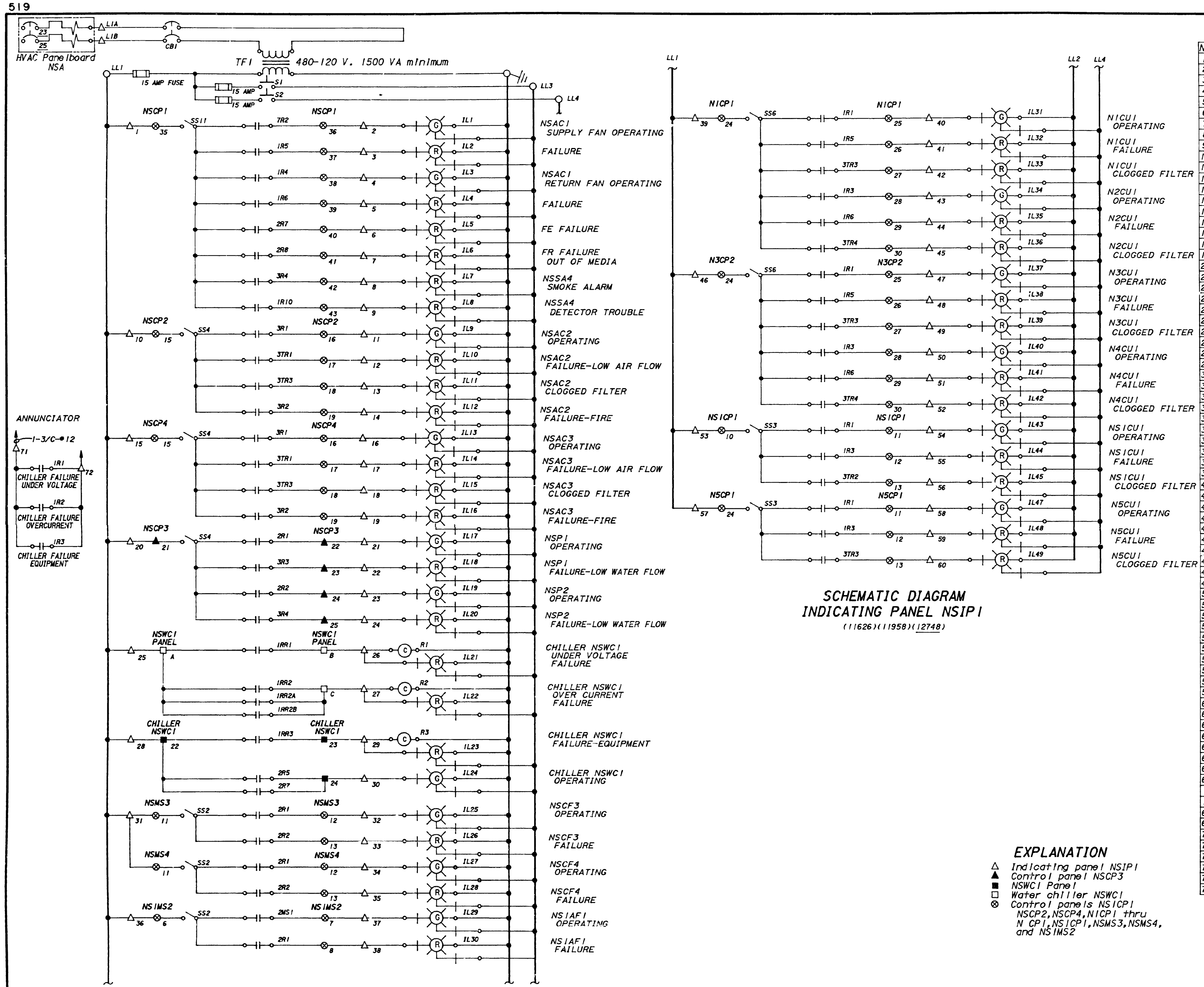
i. **Independent Controls.—**When HVAC equipment is furnished by several manufacturers, the controllers will probably be different. If the HVAC units are not integrated into an energy management system, compatibility is not a significant issue. However, if these HVAC units are to be integrated with an energy management system, incompatibility problems may appear despite the trend towards open protocols. Under these conditions the use of external unitary controllers produced by one control manufacturer may be advantageous since the risk for incompatibility is virtually eliminated. Maintenance and adjustment is also simplified since a single protocol is used.

j. **Indicating Panels.—**An indicating panel should be provided to monitor the operating status of the main HVAC system and equipment. The indicating panel should be independent of the Main HVAC system control panel to enable remote location if necessary. The panel should include sufficient indicating lights to identify specific equipment failures, warn maintenance personnel when equipment servicing is required, an assist maintenance personnel in isolating problems. A contact should also be provided to energize a warning light on the Plant Control Room display panel to warn operators of an HVAC system failure. Where the Plant is monitored from a remote control center, the indicating panel contact should be interfaced with the Remote Terminal Unit panel or Supervisory Control and Data Acquisition system. Figure 36 shows a typical indicating panel layout and identifies the equipment to be monitored.

Auxiliary equipment such as air transfer and exhaust fans should be monitored locally by installing indicating lights on their motor starter panels as discussed under the specific system requirements elsewhere in this manual.

k. **Thermostat.—**The hearing, ventilating, and cooling states of a main air conditioning unit should be controlled by one sensor (thermostat) so that overlapping of the stages cannot be caused by manual adjustments of separate sensors.

2. **Heating.**—The electric heater in each main air conditioning unit should be sized to handle the space loads and heating of the minimum quantity of outdoor air. The heater elements should be the finned sheath tube type made of Monel, ceramic-coated steel, or copper-plated steel to resist corrosion due to moisture in the air. The capacity of the heater should be divided into stages such that each



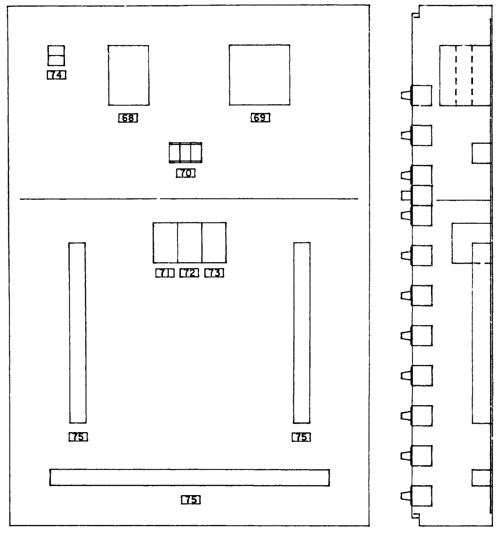
SCHEMATIC DIAGRAM
INDICATING PANEL NSIP1
(11626)(11958)(12748)

EXPLANATION

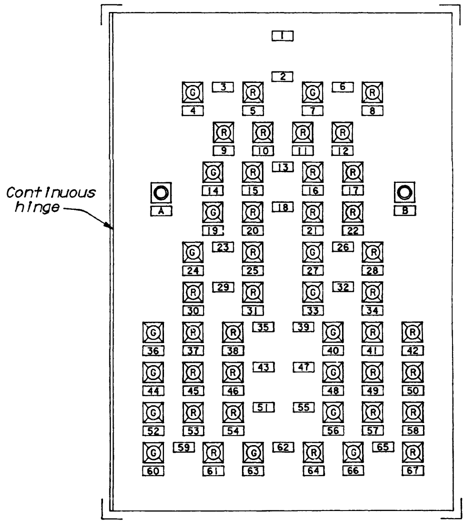
- △ Indicating panel NSIP1
- ▲ Control panel NSCP3
- ■ NSWC1 Panel
- □ Water chiller NSWC1
- ⊗ Control panels NS1CP1 NSCP2,NSCP4,N1CP1 thru N CP1,NS1CP1,NSMS3,NSMS4, and NS1MS2

NAMEPLATES

No	STATEMENT
1	AIR CONDITIONING SYSTEM INDICATING PANEL NSIP1
2	AIR CONDITIONING UNIT NSAC1
3	SUPPLY FAN
4	IL1-OPERATING
5	IL2-FAILURE-LOW AIR FLOW
6	RETURN FAN
7	IL3-OPERATING
8	IL4-FAILURE-LOW AIR FLOW
9	IL5-ELECTRONIC FILTER FAILURE
10	IL6-ROLL FILTER FAILURE-OUT OF MEDIA
11	IL7-NSSA4-SMOKE ALARM
12	IL8-NSSA4-DETECTOR TROUBLE
13	AIR CONDITIONING UNIT NSAC2
14	IL9-OPERATING
15	IL10-FAILURE-LOW AIR FLOW
16	IL11-CLOGGED FILTER BANK
17	IL12-FAILURE-FIRE
18	AIR CONDITIONING UNIT NSAC3
19	IL13-OPERATING
20	IL14-FAILURE-LOW AIR FLOW
21	IL15-CLOGGED FILTER BANK
22	IL16-FAILURE-FIRE
23	CHILLED WATER PUMP NSP1
24	IL17-OPERATING
25	IL18-FAILURE-LOW WATER FLOW
26	CHILLER WATER PUMP NSP2
27	IL19-OPERATING
28	IL20-FAILURE-LOW WATER FLOW
29	PANEL NSWC1
30	IL21-CHILLER FAILURE-UNDER VOLTAGE
31	IL22-CHILLER FAILURE-OVER CURRENT
32	WATER CHILLER NSWC1
33	IL24 OPERATING
34	IL23-FAILURE-EQUIPMENT
35	COOLING UNIT N1CU1
36	IL31-OPERATING
37	IL32-FAILURE-FAN
38	IL33-CLOGGED FILTER
39	COOLING UNIT N2CU1
40	IL34-OPERATING
41	IL35-FAILURE-FAN
42	IL36-CLOGGED FILTER
43	COOLING UNIT N3CU1
44	IL37-OPERATING
45	IL38-FAILURE-FAN
46	IL39-CLOGGED FILTER
47	COOLING UNIT N4CU1
48	IL40-OPERATING
49	IL41-FAILURE FAN
50	IL42-CLOGGED FILTER
51	COOLING UNIT N5CU1
52	IL47-OPERATING
53	IL48-FAILURE-FAN
54	IL49-CLOGGED FILTER
55	COOLING UNIT NS1CU1
56	IL43-OPERATING
57	IL44-FAILURE
58	IL45-CLOGGED FILTER
59	BATTERY ROOM EXHAUST NSCF3
60	IL25-OPERATING
61	IL26-FAILURE-FAN
62	OIL STORAGE ROOM EXHAUST NSCF4
63	IL27-OPERATING
64	IL28-FAILURE-FAN
65	TRANSFER FAN NS1CF1
66	IL29-OPERATING
67	IL30-FAILURE-FAN
A	S1-PUSH TO TEST INDICATING LIGHTS IL1 THRU IL30
B	S2-PUSH TO TEST INDICATING LIGHTS IL31 THRU IL49
68	CIRCUIT BREAKER CB1
69	TRANSFORMER TF1
70	120 VOLT FUSES
71	RELAY R1
72	RELAY R2
73	RELAY R3
74	TERMINAL BLOCKS 480 VOLT
75	TERMINAL BLOCKS 120 VOLT



PANEL INTERIOR



FRONT DOOR
PANEL LAYOUT
INDICATING PANEL NSIP1

NOTES

For general notes, reference drawings, symbols and designations, see Dwg. 344-D-10670.

ALWAYS THINK SAFETY

UNITED STATES
DEPARTMENT OF THE INTERIOR
BUREAU OF RECLAMATION
CENTRAL ARIZONA PROJECT
TUCSON DIVISION-ARIZONA

RED ROCK PUMPING PLANT

HEATING, VENTILATING, AND AIR CONDITIONING SYSTEMS

INDICATING PANEL NSIP1
SCHEMATIC DIAGRAM AND PANEL LAYOUT

DESIGNED A. L. Miller TECHNICAL APPROVAL A. L. Miller
DRAWN K. Schwalry SUBMITTED V. B. Yokes
CHECKED A. L. Miller APPROVED T. W. Rogan
Act. CHIEF, MECHANICAL BRANCH

DENVER, COLORADO
COMPUTER DRAFTING
JULY 12, 1985
344-D-12747

STRUCTURAL AND ARCHITECTURAL BRANCH
DESIGN SUPPORT SECTION
QS1:[52,30]12747C.DGN
32 8-21-85

stage will heat the volume of air between 3 ºF and 8 ºF. The heater should be automatically controlled through a step controller or proportional controller to stage the heater from no load to full load. The actual number of steps depends on the occupancy and equipment available. More steps should be used if the space being heated is for human comfort, and fewer steps should be used for equipment protection. In many applications a multistage heater may combine step and proportional control.

When heating is no longer required the outdoor air damper and return air damper should be controlled to maintain the supply air temperature above the minimum required space temperature. A sensor located in the mixed air section of each main air conditioning unit, and a sensor in the outdoor air intake, are required to control the supply air temperature. The mixed air sensor modulates the outdoor air damper and the return air damper to maintain the temperature setting. As the mixed air temperature rises, the outdoor air damper should be modulated from the minimum opening position towards the full open position and the return damper should be modulated from the full open position towards the full closed position. The exhaust dampers and associated fans should be controlled to operate as the outdoor air damper opens.

The exhaust damper should be modulated from the full closed position to full open position as the mixed air temperature rises, outdoor air damper opens. The exhaust fans should be energized as the pressurization of the structure increases or at set points in the exhaust damper operation.

3. **Cooling.**—When the free cooling cannot maintain the space temperature below the high temperature limit, the cooling system should be energized and the outdoor air damper returned to the minimum position. If provided, roof exhausters and dampers should de-energize or close once the refrigeration system is energized. When refrigerated cooling is required for each main air conditioning unit, a water chiller is normally used. The chilled water supply should be modulated through the cooling coil by use of a three way mixing valve. This allows the chilled water flow to bypass the cooling coil and allows the circulating pumps to maintain a constant volume and head. As the space sensor senses a rise in temperature the flow of water through the cooling coil should increase from no flow to full flow.

When the cooling system is energized, the chilled water pumps and condenser water pumps should be started before energizing the water chiller. After water flow is established by the pumps, the water chiller should be capable of being staged from no load to full load in a minimum of four steps using a reciprocating type water chiller or modulated from 10 percent load to full load using a centrifugal type water chiller.

4. **Roof Exhaust Systems.**—Many plants are provided with superstructure ventilation systems consisting of multiple-wall mounted air-intake louver and control-damper assemblies and exhaust-damper and fan assemblies. These systems are automatically controlled by remote bulb controllers. In plants that are air conditioned by refrigeration equipment, the controls should enable the roof exhaust fan to de-energize and the intake and exhaust dampers close when the plant indoor temperature rises to the set point calling for refrigeration cooling. This will allow stratification to take place and reduce the cooling load.

H. CONFINED SPACE HEATING AND VENTILATION – GATE CHAMBERS, TUNNELS, SHAFTS, AND VAULTS

1. **General.**—Continuous mechanical ventilating systems must be provide for confined space such as gate chambers of reservoir outlet works; outlet-pipe tunnels or access shafts; valve vaults and penstock tunnels and connecting adits, when these spaces are occupied. Spaces with explosive gases must be continuously ventilated at all times whether or not the spaces are occupied. The ventilating system may be either a supply or exhaust type. Ventilating systems may be fixed or temporary. Fixed systems are permanently installed and are designed for inspection activities only. These systems are relatively small and fairly easily accommodated in the limited space available. Because of the significantly greater airflow requirements to cope with gases, vapors, fumes, and dust, temporary systems should be used as-needed for maintenance and constructions activities.

2. **Supply Systems.**—Supply systems provide positive ventilation, create turbulence and tend to the agitate air and gases in a confined space. Supply systems are most appropriate where contaminant levels are relatively low and the air displaced from the space will not pose a health hazard to others. These conditions are fairly typical in Reclamation applications where the atmospheric hazards are oxygen deficiency, hydrogen sulfide and methane. Supply systems are also used when the outdoor air must be tempered to prevent freezing of

small pipes in the gate chambers. In supply air systems, the fan draws/forces fresh outdoor air through an air intake louver or pipe assembly and supplies it to a duct heater where it is tempered to a minimum temperature of 45 ºF. The fan forces air through a duct into the gate chamber. Stale or contaminated air from the gate chamber and tunnel is displaced by the fresh incoming air and moved downstream in the tunnel until it finds its way outdoors through suitably placed vents located at the downstream end of the tunnel. When possible, the ventilating system should be designed with fresh air flowing into the face of personnel exiting the tunnel or adit.

3. **Exhaust Systems.—**Exhaust systems are most appropriate where hazardous fumes and vapors can and should be captured at the point of occurrence to prevent diffusion into the surrounding area. Exhaust systems are also typically used for gate chambers located at the upstream ends of outlet tunnels and for penstock tunnels including connecting adits. Where exhaust systems are used, the fan exhausts the stale or contaminated air from the gate chamber or from the section of the tunnel to be ventilated and discharges it to the outside through suitable ducts. The quantity of air to be changed per hour for an exhaust system will vary according to the length and diameter of the tunnel, and the presence of contaminants such as gasses, vapors, or fumes. The cross-sectional area occupied by the penstock or outlet pipe located in the tunnel should be deducted from the gross area of the tunnel when computing the net area. However, if a penstock or outlet pipe is provided with an access door in the tunnel, the entire cross-sectional area of the tunnel should be used. Some penstock tunnels are sectionalized, and are provided with branch ducts located at appropriate points in each section of the tunnel. These branch ducts exhaust air directly from the areas to be ventilated. When branch ducts are used, the outdoor airflow may have to be increased, relative to the same tunnel with a single exhaust opening, to ensure that the minimum airflow velocity of 35 ft/min is maintained throughout all sections of the tunnel. Each branch must be provided with a balancing damper to adjust air flows for that.

4. **Air Quality Standards.—**Acceptable air quality must be maintained in tunnels and gate chambers at all times when the spaces are occupied. Adherence to the following criteria is absolutely essential:

    (1) ***Oxygen concentration*** – maintain in the range of 19.5 to 23.5 percent. Concentrations above and below these levels are hazardous. Physiological effects of various oxygen concentrations are shown in table 19.

Table 19.—Physiological effects of oxygen at different levels

% O_2	Symptoms
23.5	Oxygen enriched atmosphere
21.0	Normal oxygen level
19.5	Minimum safe entry level
12–16	Disturbed respiration, emotional upset, abnormal fatigue on exertion, flames are extinguished
10–11	Increased respiration, and heart coordination may be disturbed, some euphoria, possible headache
6–10	Nausea and vomiting, inability to move freely, possible unconsciousness, possible collapse while remaining conscious but helpless
below 6	Gasping respiration: respiration stops, followed by cardiac arrest, death in minutes

Complete Confined Space Handbook, John F. Rekus, Lewis Publishers, 1994

(2) ***Carbon monoxide concentration*** – not to exceed 25 ppm (0.0025 percent). The physiological effects of exposure to CO are shown in table 20.

(3) ***Carbon dioxide concentration*** – not to exceed 5,000 ppm (0.5 percent)

(4) ***Nitrogen dioxide concentration*** – not to exceed 3 ppm (0.0003 percent)

(5) ***Hydrogen sulfide*** – concentration not to exceed 10 ppm (0.001 percent). Test air every 4 hours when concentration exceeds 5 ppm (0.0005 percent). Provide continuous sampling and alarm when concentration exceeds 10 ppm (0.001 percent). Increase ventilation to reduce concentrations exceeding 10 ppm (0.001 percent) (time weighted average [TWA] – A workers average exposure to a contaminant over the work period or shift) for an 8-hour period. The physiological effects of exposure to H_2S are shown in table 21.

Table 20.—Signs and symptoms of exposure to carbon monoxide

CO_2 Level –ppm	Exposure Time	Signs and Symptoms
50	8 hrs.	OSHA PEL
20	8 hrs.	ACGIH TLV-TWA
200	2 to 3 hrs.	Possible mild frontal headache
400	1 to 2 hrs.	Frontal headache and nausea
400	2.5 to 3.5 hrs.	Occipital headache
800	20 min.	Headache, dizziness and nausea
800	2 hrs.	Collapse, possible death
1,600	20 min.	Headache, dizziness and nausea
1,600	2 hrs.	Collapse, possible death
3,200	5 to 10 min.	Headache and dizziness
3,200	10 to 15 min.	Unconsciousness, danger
128,000	1 to 3 min.	Immediate effect, Unconsciousness, danger of death

Complete Confined Space Handbook, John F. Rekus, Lewis Publishers, 1994

Table 21.—Signs and symptoms of exposure to hydrogen sulfide

H_2S Level–ppm	Exposure Time	Signs and Symptoms of Exposure
0.1	–	Odor threshold
5.0	–	Moderate odor
20	8 hrs.	OSHA PEL
10	8 hrs.	ACGIH TLV
15	15 min.	
25	–	Tolerable, but strong, unpleasant odor
100	2 to 5 min.	Eye irritation, coughing, loss of sense of smell
200–300	1 hr.	Marked eye irritation and respiratory tract
500–700	30 to 60 min	irritation
700–1,000	minutes	Loss of consciousness and possible death
1,000–2,000	–	Rapid unconsciousness, respiratory distress, death
		Unconsciousness almost immediately. Respiration stops, death in a few minutes. Death may occur even if the victim is removed to fresh air.

Complete Confined Space Handbook, John F. Rekus, Lewis Publishers, 1994.

(6) ***Methane gas*** – concentration not to exceed 20 percent of the (lower explosive limit [LEL] – The lower limit of flammability or explosiveness of a gas or vapor at ordinary ambient temperatures expressed in percent of the gas or vapor in air by volume. This limit is assumed constant for temperatures up to 250 ºF. Above this temperature it should be decreased by a factor 0.7 since explosiveness increases with higher temperatures (see *ACGIH*). Increase ventilation whenever the concentration of methane or other flammable gas is 5 percent or more of the LEL.

(7) ***Other flammable gases or vapors*** – concentration not to exceed 20 percent of LEL.

(8) ***Airborne contaminants*** – Comply with the most stringent requirement of *OSHA* personnel exposure limits (PEL); *ACGIH* threshold limit values (TLV) which are the values for airborne toxic materials which are used as guides in the control of health hazards and represent time-weighted concentrations to which nearly all workers may be exposed 8 hours per day over extended periods of time without adverse affects; State or local standards; or the industrial hygienist overseeing the job. Control programs are required when airborne contaminants equal or exceed the action levels established in specific occupational health standards. In the absence of specific action levels, 50 percent of the appropriate PEL/TLV shall be used as the action level.

5. Airflow Requirements.—The quantity of air to be supplied or exhausted will vary according to the size of the gate chamber, the length and size of the tunnel or access shaft, and the level of oxygen or presence of harmful gases such as carbon monoxide, carbon dioxide and hydrogen sulfide, or explosive gases such as methane. The following criteria are provided to assist in determining the minimum ventilating system airflow requirements. The actual design airflow must satisfy all the criteria; therefore, the criteria that yields the highest airflow should be used to size the fan and other ventilating system equipment.

a. Fixed Ventilating System Airflow Design Criteria.—Fixed ventilating systems are normally designed, by Reclamation personnel, for permanent installation and are intended for intermittent operation during inspection of tunnels, gate chambers, gates, gate controls, and other equipment contained in these areas. These systems are not designed or intended to provide adequate ventilation during maintenance or construction activities. Refer to heading Temporary Ventilating Systems for operating criteria for ventilating systems intended for maintenance and construction activities.

Various methods for determining confined space ventilation requirements are available. Unfortunately, these methods do not yield the same results. These methods are presented for reference only. In the absence of more stringent criteria, confined space ventilating systems for Reclamation projects should be designed to comply with *Reclamation Safety and Health Standards* (RSHS) or *OSHA* requirements, whichever is more stringent. Designers should avoid using of rules-of-thumb except for quick estimates and only when the basis of the rule-of-thumb and its applicability to the job is known. Reclamation drawings and documents such as *Design Summary* and *Designer's Operating Criteria* should cautions field personnel to measure gas levels with calibrated direct reading instruments when entering and while occupying confined spaces.

(1) ***OSHA minimum per person.***—The absolute minimum volume of air to be supplied to the gate chamber or confined space should be 200 cfm per person (see *OSHA 29CFR, CH XVII, 1926.800*). A minimum of three persons, 600 cfm should be assumed to occupy the space simultaneously.

(2) ***OSHA tunnel ventilation.***—Maintain a minimum airflow velocity of 30 ft/min in the tunnel or shaft (see *OSHA 29CFR Chapter XVII , 1926.800)*

(3) ***National safety council.***—The *Complete Confined Space Handbook* recommends a minimum airflow based on the following equation:

$$Q = 7.5\ V/t$$

Where: Q = fan airflow capacity, cfm
V = total volume of the confined space, ft^3
t = desired purge time, minutes

(4) ***Reclamation Safety and Health Standards (RSHS).***—Ventilating systems for adits, tunnels, gate chambers, and water conveyance structures should be designed to provide 35 ft/min airflow velocity over the cross sectional area if no atmospheric contaminants are anticipated. If atmospheric contaminants are anticipated, the airflow should be increased to provide a velocity of 100 ft/min. Entry to the space should be permitted only after 2 complete air changes of the entire volume are completed. A system designed for these minimum requirements will usually result in long purge times. When possible, Reclamation ventilating systems should be designed to produce the maximum airflow consistent with the available power supply, space requirements and the desire to reduce the purge time for inspection personnel. Usually the power and space available are sufficient for a ventilating system sized to complete the purge requirements in 30 to 60 minutes.

b. **Temporary Ventilating Systems Airflow Design Criteria.—** Temporary ventilating systems are used for maintenance and construction activities in tunnels and gate chambers. Unless the work is to be accomplished by government personnel, these systems are rarely designed by Reclamation engineers. However, contract/specifications will require that the contractor performing work in a confined space submit their proposed design to the Reclamation for approval before commencing work. Reclamation engineers should design, when necessary, or verify that any contractor proposed ventilating system complies with the greater of the following criteria from *RSHS*:

(1) The cfm required to maintain a minimum airflow velocity of 100 ft/min over the gross bore area of all sections of the tunnel.

(2) The cfm required to control airborne contaminants or toxic and flammable gas/vapor within prescribed limits and/or values identified under the required air quality standards referenced above.

(3) When diesel or internal combustion engines are used, refer to *RSHS* for ventilation requirements. In no case shall the ventilation system be less than 150 cfm times the sum of the manufacturers rated horsepower for all engines when operating at the maximum fuel/air ratio.

Once the design airflow rate is determined, the duct and fan required to deliver the air may be too large for the available space. These criteria may have to be re-evaluated to find a compromise between the airflow rate, acceptable lead times before entering the gate chamber, tunnel or shaft, and the size of the equipment. In some circumstances, workers may have to use special breathing equipment to ensure personnel safety.

The volume occupied by the equipment located in the gate chamber and outlet pipes located in the tunnel should be deducted from the gross volume of the gate chamber and tunnel when computing the net volume of air required. Figure 37 shows a typical Reclamation gate chamber ventilation system.

6. Ductwork.

a. Design Criteria.

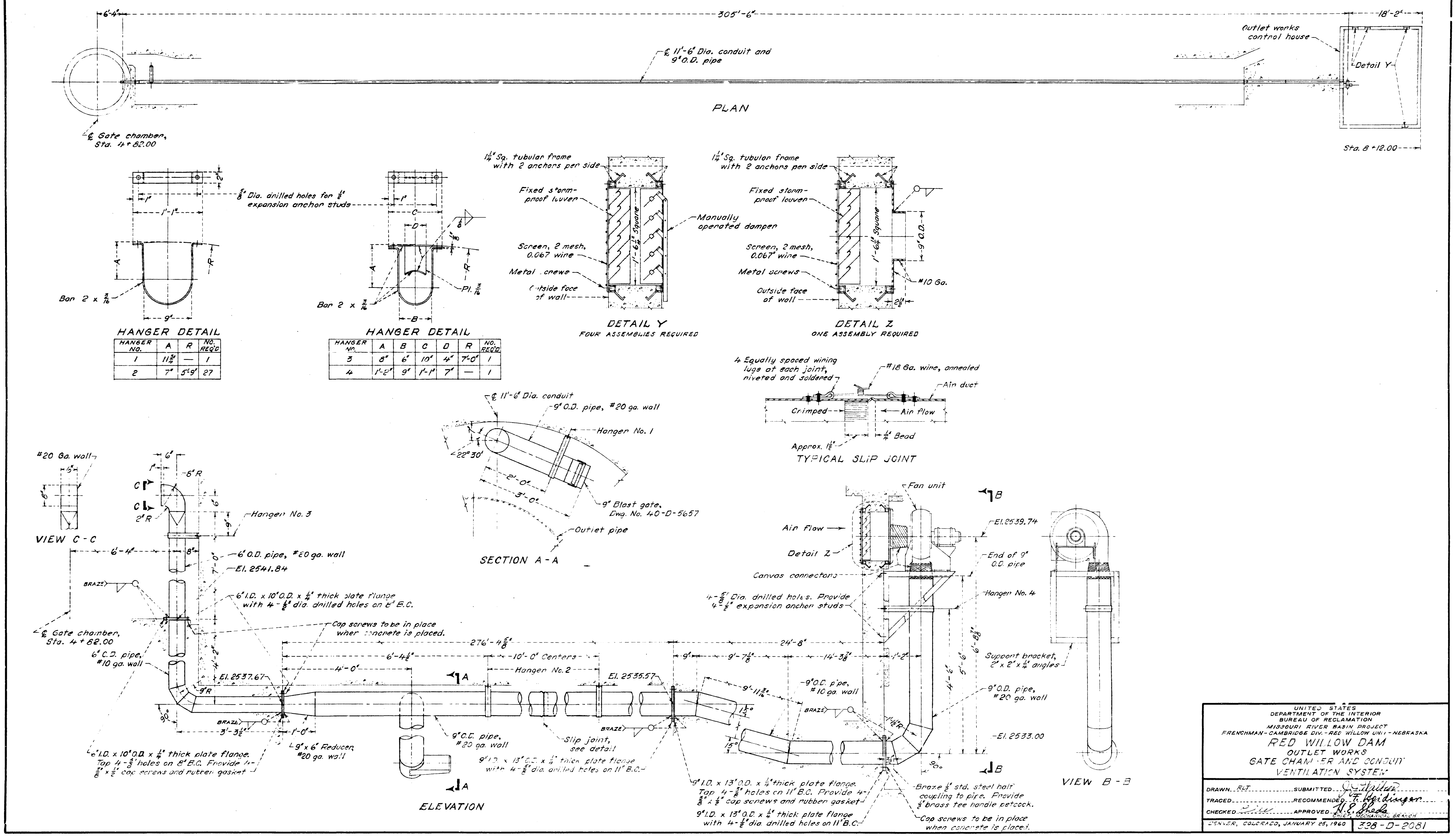
(1) ***Fixed systems.***—The quantity of air passing through the ducts is equal to the total quantity of air required at the points of discharge. Once the required cfm is known, the static pressure, or resistance of the duct system to the flow of the air, is calculated and a fan capable of providing the required flow at the design pressure is selected. Static pressures are expressed in inches of water.

(a) *Duct* s*izing*.—Ducts for gate chamber ventilating should be sized for an airflow resistance of 0.3 to 0.35-inch w.g./100 ft of duct. The main duct in the penstock tunnels should be sized for an air velocity of 1,300 to 2,000 ft/min, and branch ducts in the penstock tunnels should be sized for an air velocity of 800 to 1,000 ft/min.

For a given volume of air, the resistance depends on the size and length of ducts, and the type and number of fittings (elbows, tees, transitions), obstructions (dampers, heating coils, screens), and louvers. Elbows should be made with an inside radius not less than the diameter of the duct and is commonly referred to as an R/D = 1. Such elbows (90º turns) offer approximately the same resistance to flow as a straight duct having a length in feet equal to the diameter of the duct in inches. If space permits, an R/D of 1.5 should be used.

(b) *Exhaust air intakes/supply hood.*—The location of exhaust air intake openings is dictated by the hazard present in the vault or gate chamber. If gases heavier than air (hydrogen sulfide and propane) are expected, and no floor recesses are provided for gates, the exhaust air intake must be located close to floor level.

FORM 7-1289
(OCT. '58)



HANGER DETAIL

HANGER NO.	A	R	NO. REQ'D
1	11¾"	—	1
2	7"	5'-9"	27

HANGER DETAIL

HANGER NO.	A	B	C	D	R	NO. REQ'D
3	8"	6"	10"	4"	7'-0"	1
4	1'-2"	9"	1'-1"	7"	—	1

UNITED STATES
DEPARTMENT OF THE INTERIOR
BUREAU OF RECLAMATION
MISSOURI RIVER BASIN PROJECT
FRENCHMAN-CAMBRIDGE DIV.-RED WILLOW UNIT-NEBRASKA

RED WILLOW DAM
OUTLET WORKS
GATE CHAMBER AND CONDUIT
VENTILATION SYSTEM

DRAWN RLT SUBMITTED J. J. Walker
TRACED RECOMMENDED F. Heidinger
CHECKED APPROVED H. E. Sheda
CHIEF, MECHANICAL BRANCH
DENVER, COLORADO, JANUARY 25, 1960
328-D-2081

FEB 23 '62

Job No. 7-328-01-04-01

GPO 831395

21" x 36"

MICROFILMED AUGUST 17 1973

Where floor recesses are provided, exhaust air intakes must be installed near the floor level of each separate recess. The intake or air supply is usually a round expansion cone, but may be a rectangular transition without grilles. The total angle of divergence for hoods should be no greater than 30 degrees, i.e. 15 degrees per side. The clearance between the floor and the face of the hood should be neither less than 0.5 equivalent diameters, nor greater than 1.0 equivalent diameter. When gases lighter than air (carbon monoxide, hydrogen, methane) are anticipated, the air exhaust opening should be located as close to the ceiling as possible. The air intake opening on supply or exhaust ducts in tunnels or gate chambers should be provided with a 2x2 galvanized or stainless steel wire mesh bird screen. Insect screens should not be used except at the outdoor air intake louver or pipe opening. Air discharge or exhaust openings should not be oriented up towards the ceiling. After prolonged exposure to wet environments insect screens oriented towards the ceiling can be completely clogged with calcium deposits and dirt. Screen orientation should be approximately 30 degrees from vertical.

(2) ***Temporary systems.***—Air flows must be mechanically induced. Passive ventilation is not acceptable. Electric motors, fans, drives, and auxiliary equipment, including wiring, starters, and controls, must comply with NEC class 1, division 1, explosion proof requirements. Size duct for a pressure loss of 0.30 to 0.35-inch w.g./100 feet of duct. Noise from the fan must not exceed 85 dBA measured at the closest point of worker exposure. The airflow must be reversible from a surface location without rewiring the fan controls. The system must be designed to operate in the exhausting mode. The primary duct system inlets must be maintained within 3 duct diameters of the tunnel face or shaft bottom when operating in the exhaust mode and within 10 duct diameters of the tunnel face when operating in the supply mode.

With the above information established, it is possible to determine the frictional resistance and the size of the duct, and to consider the size and speed of the fan required for the system. The friction loss charts shown in *SMACNA HVAC Systems Duct Design* are useful in determining the static pressure in inches of water required to overcome the friction due to the passage of air in the ducts.

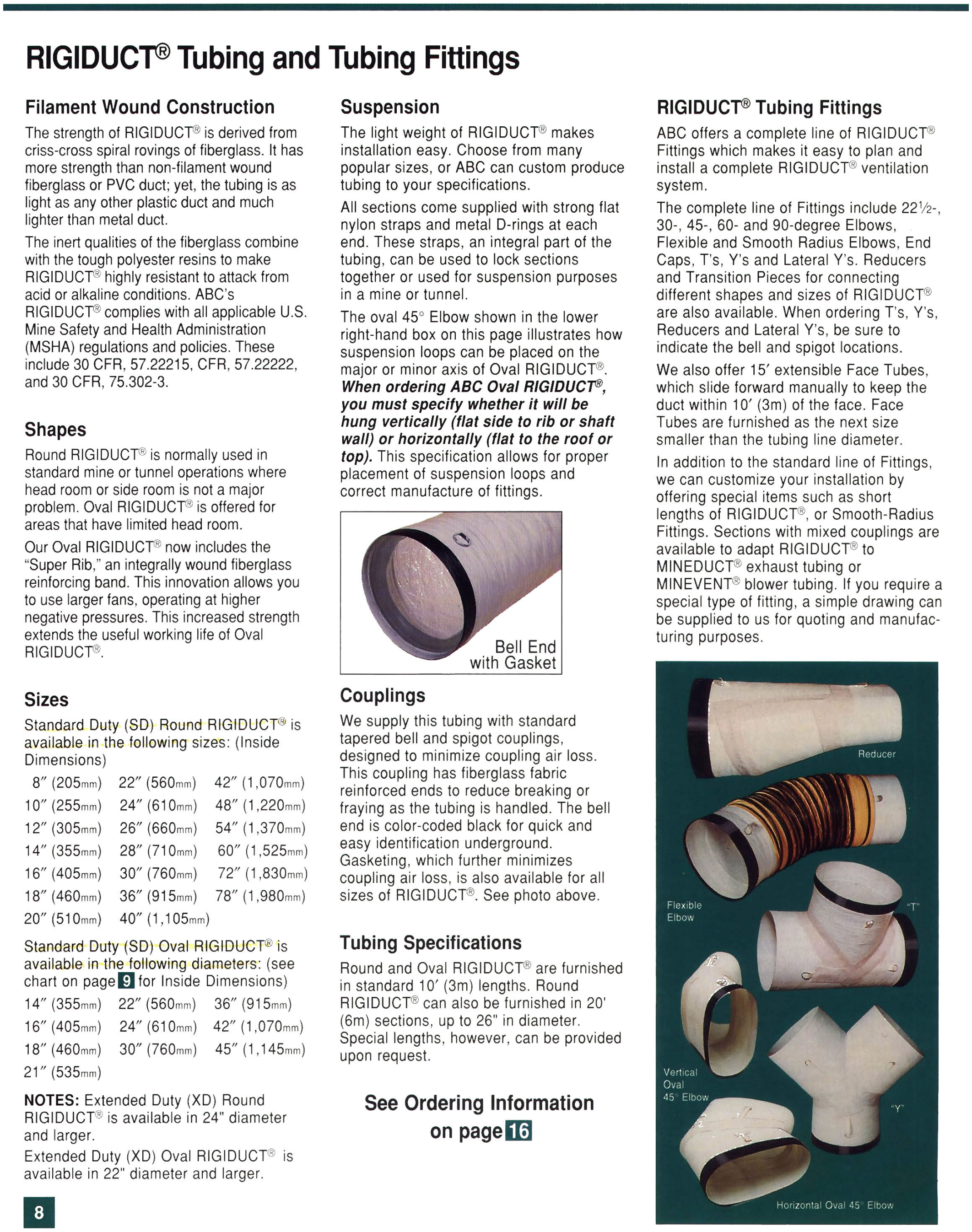
b. Duct Construction.

(1) ***Galvanized ductwork.***—Galvanized duct is very common in tunnel and gate chamber applications. Steel sheet should be G-90 coated (or better) galvanized steel of lock forming grade conforming to *ASTM A653* and *A924* standards.

Air ducts should be outside diameter slip-joint duct or spiral duct made from galvanized-steel sheets. The longitudinal and circumferential duct seams (joints) of slip-joint duct should be of the grooved- or double-seam type with the seam formed on the outside of the duct, the ends of the longitudinal seams riveted, and all seams soldered on the outside. Longitudinal joints of rectangular ducts should be formed on a corner of the duct and be riveted on the ends and soldered on the outside. Laying lengths of ducts should not exceed 14 feet 7-1/2 inches. Slip joints should be of the crimped type with bead that should be located on laying length ends of duct, and each joint should be complete with wiring lugs riveted and soldered to the duct. For slip-joint duct and spiral duct, all support bars, frame angles, wiring lugs, and flanges attached in the shop to the duct, should be riveted and soldered to the duct.

To prevent transmission of vibration, the ventilating fan must be isolated from all ductwork or the concrete structure. Flexible connectors should be provided between the fan exhaust and intake openings and other ventilating system components or the concrete structure. Galvanized steel transitions should be provided where required. Sudden enlargements or contractions in the duct system should be avoided by allowing a maximum angle of 15° in the transition when possible. Hangers and supports should be made from hot-rolled galvanized steel strip. Branch ducts should be provided with balancing dampers of the butterfly type. Shutoff gates of the blast type should be avoided. In tunnels, the ducts should be located to clear overhead lights, conduits, and piping. Duct bends should be designed for smooth airflow. When elbows must be provided near the fan intake or discharge openings, allow at least 2.5 equivalent duct diameters between the elbow and the fan supply or connections. There are many ways to lay out ductwork for a given application. There are also many precautions, which should be observed to prevent unnecessary pressure losses, noise, and vibrations which are frequently present in poorly designed systems. For additional precautions and procedures related to duct design refer to *SMACNA HVAC System Duct Design*. The metal thickness gauge, reinforcement and hanger requirements, and weight estimates for a given pressure can be determined from the tables in the *SMACNA Duct Construction Standards*.

(2) ***FRP duct.***—Although very common in the mining industry, until recent years FRP has not been used extensively in Reclamation projects. FRP duct is formed over a mandrel from filament wound fiberglass coated with polyester resins. FRP duct is manufactured to comply with MSHA (Mine Safety and Health Administration) regulations. The interior surface is very smooth and the duct is light, and available in round sizes from 8 to 30-inches in 2-inch increments, and from 30 to 78-inches in 6-inch increments. Many standard fittings are available. Oval sizes are also available. For Reclamation applications, the greatest advantage of FRP duct over galvanized duct is the lack of corrosion. The greatest disadvantage is higher cost compared to metal duct. When standard fittings cannot be used, the cost rises significantly. Because of the lack of corrosion, FRP is becoming the preferred choice for wet and high humidity installations that tend to corrode metal duct. The cost associated with maintenance in a confined space environment can be offset by using FRP duct. See figures 38 and 39 for additional information.



RIGIDUCT® Tubing and Tubing Fittings

Filament Wound Construction

The strength of RIGIDUCT® is derived from criss-cross spiral rovings of fiberglass. It has more strength than non-filament wound fiberglass or PVC duct; yet, the tubing is as light as any other plastic duct and much lighter than metal duct.

The inert qualities of the fiberglass combine with the tough polyester resins to make RIGIDUCT® highly resistant to attack from acid or alkaline conditions. ABC's RIGIDUCT® complies with all applicable U.S. Mine Safety and Health Administration (MSHA) regulations and policies. These include 30 CFR, 57.22215, CFR, 57.22222, and 30 CFR, 75.302-3.

Shapes

Round RIGIDUCT® is normally used in standard mine or tunnel operations where head room or side room is not a major problem. Oval RIGIDUCT® is offered for areas that have limited head room.

Our Oval RIGIDUCT® now includes the "Super Rib," an integrally wound fiberglass reinforcing band. This innovation allows you to use larger fans, operating at higher negative pressures. This increased strength extends the useful working life of Oval RIGIDUCT®.

Sizes

Standard Duty (SD) Round RIGIDUCT® is available in the following sizes: (Inside Dimensions)

8″ (205mm)	22″ (560mm)	42″ (1,070mm)
10″ (255mm)	24″ (610mm)	48″ (1,220mm)
12″ (305mm)	26″ (660mm)	54″ (1,370mm)
14″ (355mm)	28″ (710mm)	60″ (1,525mm)
16″ (405mm)	30″ (760mm)	72″ (1,830mm)
18″ (460mm)	36″ (915mm)	78″ (1,980mm)
20″ (510mm)	40″ (1,105mm)	

Standard Duty (SD) Oval RIGIDUCT® is available in the following diameters: (see chart on page 9 for Inside Dimensions)

14″ (355mm)	22″ (560mm)	36″ (915mm)
16″ (405mm)	24″ (610mm)	42″ (1,070mm)
18″ (460mm)	30″ (760mm)	45″ (1,145mm)
21″ (535mm)		

NOTES: Extended Duty (XD) Round RIGIDUCT® is available in 24" diameter and larger.

Extended Duty (XD) Oval RIGIDUCT® is available in 22" diameter and larger.

Suspension

The light weight of RIGIDUCT® makes installation easy. Choose from many popular sizes, or ABC can custom produce tubing to your specifications.

All sections come supplied with strong flat nylon straps and metal D-rings at each end. These straps, an integral part of the tubing, can be used to lock sections together or used for suspension purposes in a mine or tunnel.

The oval 45° Elbow shown in the lower right-hand box on this page illustrates how suspension loops can be placed on the major or minor axis of Oval RIGIDUCT®. ***When ordering ABC Oval RIGIDUCT®, you must specify whether it will be hung vertically (flat side to rib or shaft wall) or horizontally (flat to the roof or top).*** This specification allows for proper placement of suspension loops and correct manufacture of fittings.

Couplings

We supply this tubing with standard tapered bell and spigot couplings, designed to minimize coupling air loss. This coupling has fiberglass fabric reinforced ends to reduce breaking or fraying as the tubing is handled. The bell end is color-coded black for quick and easy identification underground. Gasketing, which further minimizes coupling air loss, is also available for all sizes of RIGIDUCT®. See photo above.

Tubing Specifications

Round and Oval RIGIDUCT® are furnished in standard 10′ (3m) lengths. Round RIGIDUCT® can also be furnished in 20' (6m) sections, up to 26" in diameter. Special lengths, however, can be provided upon request.

See Ordering Information on page 16

RIGIDUCT® Tubing Fittings

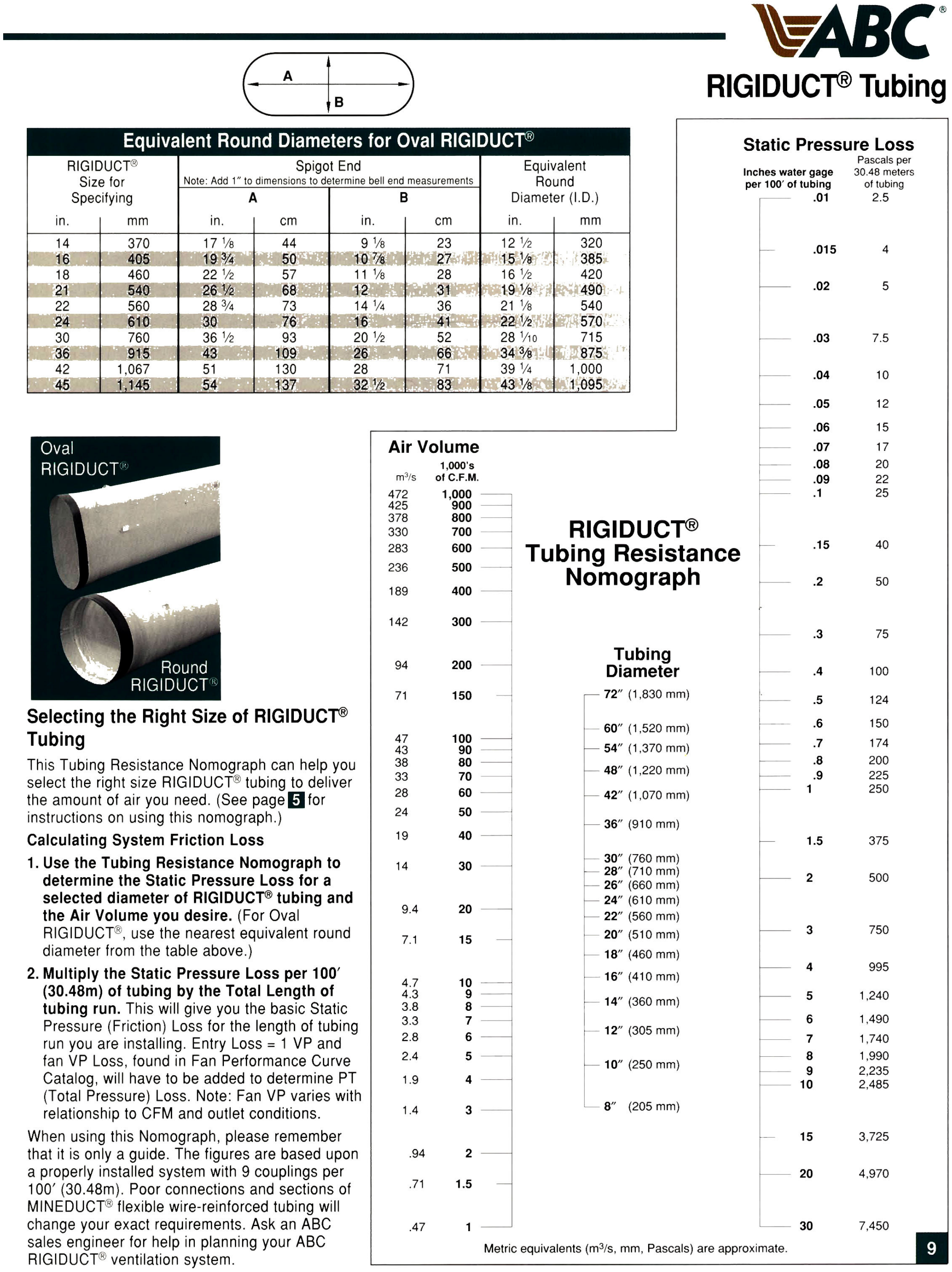
ABC offers a complete line of RIGIDUCT® Fittings which makes it easy to plan and install a complete RIGIDUCT® ventilation system.

The complete line of Fittings include 22½-, 30-, 45-, 60- and 90-degree Elbows, Flexible and Smooth Radius Elbows, End Caps, T's, Y's and Lateral Y's. Reducers and Transition Pieces for connecting different shapes and sizes of RIGIDUCT® are also available. When ordering T's, Y's, Reducers and Lateral Y's, be sure to indicate the bell and spigot locations.

We also offer 15′ extensible Face Tubes, which slide forward manually to keep the duct within 10′ (3m) of the face. Face Tubes are furnished as the next size smaller than the tubing line diameter.

In addition to the standard line of Fittings, we can customize your installation by offering special items such as short lengths of RIGIDUCT®, or Smooth-Radius Fittings. Sections with mixed couplings are available to adapt RIGIDUCT® to MINEDUCT® exhaust tubing or MINEVENT® blower tubing. If you require a special type of fitting, a simple drawing can be supplied to us for quoting and manufacturing purposes.

Figure 38.—FRP duct – sizes and specifications.



RIGIDUCT® Tubing

Equivalent Round Diameters for Oval RIGIDUCT®

RIGIDUCT® Size for Specifying		Spigot End Note: Add 1″ to dimensions to determine bell end measurements				Equivalent Round Diameter (I.D.)	
		A		B			
in.	mm	in.	cm	in.	cm	in.	mm
14	370	17 1/8	44	9 1/8	23	12 1/2	320
16	405	19 3/4	50	10 7/8	27	15 1/8	385
18	460	22 1/2	57	11 1/8	28	16 1/2	420
21	540	26 1/2	68	12	31	19 1/8	490
22	560	28 3/4	73	14 1/4	36	21 1/8	540
24	610	30	76	16	41	22 1/2	570
30	760	36 1/2	93	20 1/2	52	28 1/10	715
36	915	43	109	26	66	34 3/8	875
42	1,067	51	130	28	71	39 1/4	1,000
45	1,145	54	137	32 1/2	83	43 1/8	1,095

Selecting the Right Size of RIGIDUCT® Tubing

This Tubing Resistance Nomograph can help you select the right size RIGIDUCT® tubing to deliver the amount of air you need. (See page **5** for instructions on using this nomograph.)

Calculating System Friction Loss

1. **Use the Tubing Resistance Nomograph to determine the Static Pressure Loss for a selected diameter of RIGIDUCT® tubing and the Air Volume you desire.** (For Oval RIGIDUCT®, use the nearest equivalent round diameter from the table above.)
2. **Multiply the Static Pressure Loss per 100′ (30.48m) of tubing by the Total Length of tubing run.** This will give you the basic Static Pressure (Friction) Loss for the length of tubing run you are installing. Entry Loss = 1 VP and fan VP Loss, found in Fan Performance Curve Catalog, will have to be added to determine PT (Total Pressure) Loss. Note: Fan VP varies with relationship to CFM and outlet conditions.

When using this Nomograph, please remember that it is only a guide. The figures are based upon a properly installed system with 9 couplings per 100′ (30.48m). Poor connections and sections of MINEDUCT® flexible wire-reinforced tubing will change your exact requirements. Ask an ABC sales engineer for help in planning your ABC RIGIDUCT® ventilation system.

RIGIDUCT® Tubing Resistance Nomograph

Air Volume

m³/s	1,000's of C.F.M.
472	1,000
425	900
378	800
330	700
283	600
236	500
189	400
142	300
94	200
71	150
47	100
43	90
38	80
33	70
28	60
24	50
19	40
14	30
9.4	20
7.1	15
4.7	10
4.3	9
3.8	8
3.3	7
2.8	6
2.4	5
1.9	4
1.4	3
.94	2
.71	1.5
.47	1

Tubing Diameter

- **72″** (1,830 mm)
- **60″** (1,520 mm)
- **54″** (1,370 mm)
- **48″** (1,220 mm)
- **42″** (1,070 mm)
- **36″** (910 mm)
- **30″** (760 mm)
- **28″** (710 mm)
- **26″** (660 mm)
- **24″** (610 mm)
- **22″** (560 mm)
- **20″** (510 mm)
- **18″** (460 mm)
- **16″** (410 mm)
- **14″** (360 mm)
- **12″** (305 mm)
- **10″** (250 mm)
- **8″** (205 mm)

Static Pressure Loss

Inches water gage per 100′ of tubing	Pascals per 30.48 meters of tubing
.01	2.5
.015	4
.02	5
.03	7.5
.04	10
.05	12
.06	15
.07	17
.08	20
.09	22
.1	25
.15	40
.2	50
.3	75
.4	100
.5	124
.6	150
.7	174
.8	200
.9	225
1	250
1.5	375
2	500
3	750
4	995
5	1,240
6	1,490
7	1,740
8	1,990
9	2,235
10	2,485
15	3,725
20	4,970
30	7,450

Metric equivalents (m³/s, mm, Pascals) are approximate.

9

Figure 39.—FRP duct – selection data.

(3) ***PVC duct.***—PVC duct is not commonly used in Reclamation work. PVC ductwork is used primarily in fan systems where corrosive chemicals must be exhausted such as laboratories, or water treatment plants. PVC can also be used in wet environments where galvanized ductwork experiences excessive corrosion. However, most facilities prefer not to use PVC due to its brittle nature and the toxic fumes generated by the solvents and glues when repairs are required.

(4) ***Fans.***—Fans for both supply and exhaust systems should be of the centrifugal, radial, or backward-incline blade type, direct-connected to the motor. The unit should be integrally mounted on a suitable steel or cast iron base, to be supported on the floor or on wall brackets. After the quantity of air and the static pressure necessary have been determined, the fan can be selected. Fan manufacturer's data generally include full operating characteristics of any particular fan and indicate a recommended range of operation. Avoid operating points in the unstable region of the performance curves. Unless the fan will operate continuously, an economical fan selection should be adequate. If the fan will operate continuously, select the fan based on best efficiency. The manufacturer's data are presented for sea level conditions. For applications 2000 ft above sea level or higher and temperatures above 70 ºF, the manufacturer's data must be corrected for altitude and temperature. See table 6 for applicable corrections.

(5) ***Heating.***—Except in the coldest climates where freeze protection is absolutely necessary, heating is rarely required or provided. Local area personnel should be consulted regarding low temperatures and the need or desire for heating spaces, protecting equipment, or both. Heat loss is greatest from exposed vault covers and concrete walls and diminishes with increasing depth below grade level. *ASHRAE Handbook of Fundamentals* contains procedures and data for estimating heat loss from below grade structures.

(a) *Tunnels.*—When possible, tunnel and gate chamber ventilating system supply air temperature should be maintained at 45 ºF.

(b) *Vaults*.—In mildly cold climates, heat tape or insulation applied to the vault ceiling, walls, and vent pipes is sufficient to prevent freezing. Low leakage control dampers should be provided to prevent loss of warm air. In very cold climates where freezing is probable and heating is not possible or desired, valve vault covers should be insulated with 2 to 4-inch rigid Styrofoam board. The insulation should also be extended along the walls from the roof cover to minimum of 4-feet below ground level or frost level whichever is greater. Heat tape should be applied to pipes to prevent freezing. When heat taping is not practical or when heating is necessary for personnel comfort, unit heaters should be installed. In moderate climates one unit heater should be sufficient to prevent freezing. For severe outdoor climates two unit heaters should be provided to prevent freezing if one heater fails. With two heaters operating, the units should be capable of maintaining 45 ºF in the vault with the outdoor temperature at the *ASHRAE Handbook of Fundamentals* 1 percent winter conditions for the location. In addition, each heater should be capable of maintaining 35 ºF when one unit is inoperable.

(6) ***Controls and operation.***—Gate chamber and tunnel ventilating fans are usually controlled by a manual motor starter through an ON-OFF switch located near the entry door to the control house. Frequently, the fan switch is interlocked with the light switch to energize both simultaneously. If power to the fan cannot be locked-out, the *NEC* requires a disconnect switch be provided near the fan to electrically isolate it from the power source during maintenance. Regardless of the type of fan drive used (direct or belt) an adjustable airflow switch or differential pressure switch with a momentary time delay should be provided to initiate an audible alarm upon loss of airflow when the ventilating system is energized. Loss of power to the tunnel lights will probably result in loss of power to the ventilating system fan and alarm. Designers should insert cautions to this effect in documents such as Standard Operating Procedures (SOPs) and Designer's Operating

Criteria (DOC) prepared for the facility. Occupants should be warned to evacuate the vault, tunnel, or gate chamber under these conditions.

I. TESTING AND COMMISSIONING

1. **Testing, Adjusting and Balancing (TAB).**—Testing, adjusting and balancing is a procedure that enables systematic checking and adjusting all the environmental systems in a plant or other building to ensure operation in accordance with the design objectives. TAB procedures are generally performed after the equipment is completely installed and serviced. TAB procedures involve quantifying and reporting the following:

    a. **Balancing Air and Water Flow.**—Balancing valves on coil or water circuits and balancing dampers on air distribution systems must be adjusted to provide the required flows.

    b. **Adjust System.**—Fans and other equipment may require speed adjustments to provide the design quantities.

    c. **Electrical Measurements.**—Voltage and current measurements are required to verify power requirements and ensure the motors are not overloaded.

    d. **Equipment Ratings.**—Record data to verify the performance of all components complies with the intent of the design.

    e. **Controls.**—Verify settings and operation of automatic controls.

    f. **Noise and Vibration.**—Measure sound and vibration on critical systems or systems that exhibit abnormal noise and vibration levels. However, noise and vibration measurements are not normally included in TAB procedures for Reclamation projects.

2. **Procedures and Workmanship -** Successful testing, adjusting and balancing of a system is highly dependent on the degree of care exercised in the procedures and workmanship of the TAB contractor.

    a. **Standard Procedures.**—Contractors should follow one of the several standard procedures available, such as those published by *ASHRAE*, *SMACNA*, *NEBB (National Environmental Balancing Bureau)* and *ABCC (Air Balancing and Control Council)*.

b. **Certification.**—All TAB firms or technicians should provide evidence of current certification by the entity sanctioning the TAB procedures.

c. **Standard Reports.**—Data should be reported on standard report forms generally provided by certification agencies. These reports identify the types of measurements required for each piece of equipment, thereby ensuring that all pertinent data are recorded and reducing the opportunity for error. Sample test reports for various types of equipment commonly encountered in Reclamation designs are shown in figure 40.

d. **Instrument Calibration.**—All TAB instrument calibrations should be current when the testing is conducted.

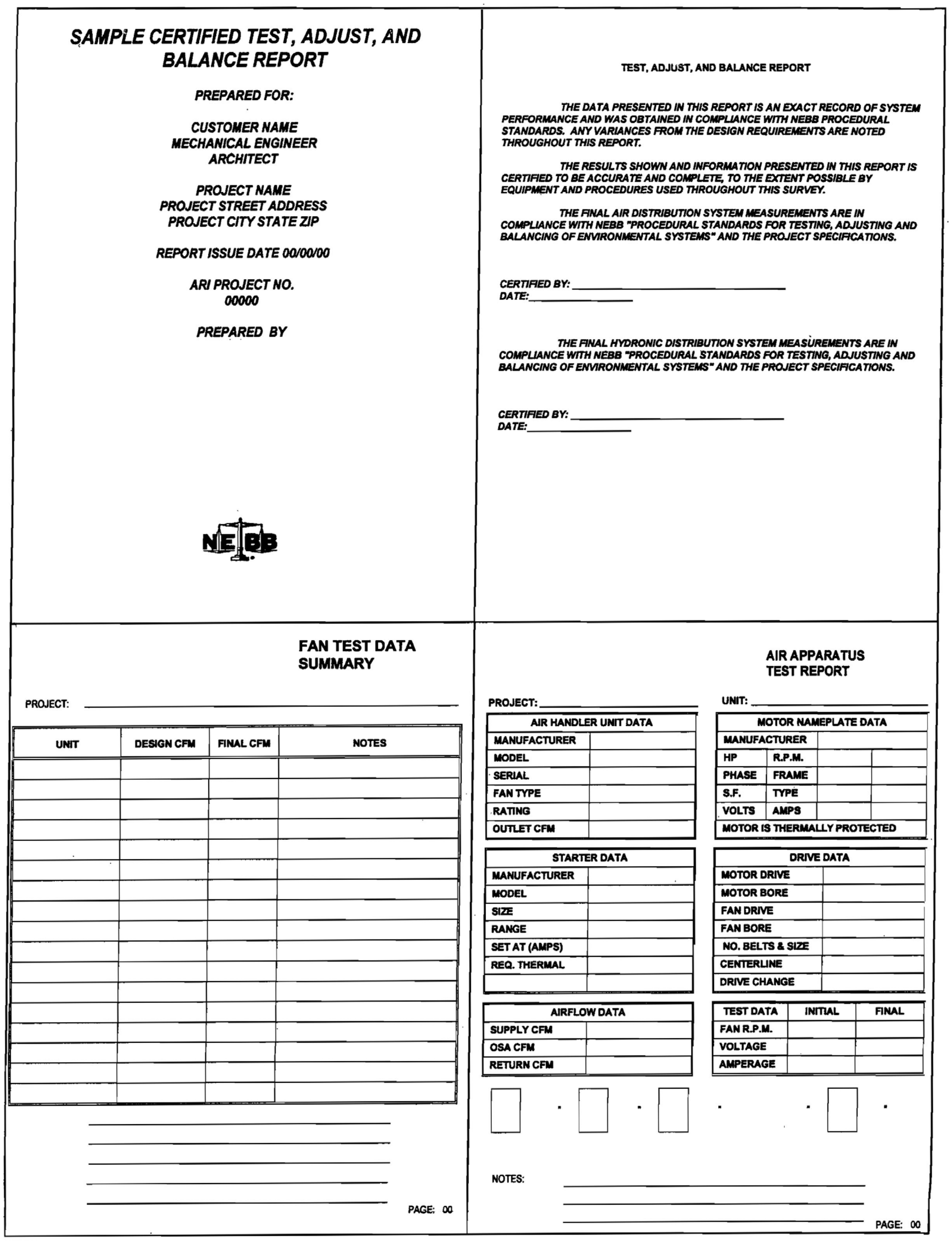
3. **System Design.**—Successful testing, adjusting and balancing of a system is also highly dependent on the degree of care exercised in the design of a system. Designers are encouraged to follow the recommended design guidelines provided in this manual and those recommended by other recognized sources referenced in the appendix. Designers should verify their designs include the following information:

a. **Balancing Dampers.**—Provide sufficient balancing dampers and valves at branch connections throughout the system. These devices should be located the required number of equivalent diameters away from elbows or other fittings or components that may adversely affect the type of measurements to be made.

b. **Transition Requirements.**—Stipulate the minimum length or maximum divergence and contraction angles for all transitions. Generally transitions should not exceed 7 degrees on each side or a total of 15 degree included angle.

c. **Minimum Clearances.**—Stipulate the minimum duct length required between elbows and fan inlet and outlet connections. This can be done by showing the actual length required or specifying the equivalent duct diameters.

d. **Measuring Points.**—The specific location of critical measurement points away from fittings, transitions, obstructions and other equipment affecting airflow should be identified on the drawings to ensure that accurate readings can be taken.



SAMPLE CERTIFIED TEST, ADJUST, AND BALANCE REPORT

PREPARED FOR:

CUSTOMER NAME
MECHANICAL ENGINEER
ARCHITECT

PROJECT NAME
PROJECT STREET ADDRESS
PROJECT CITY STATE ZIP

REPORT ISSUE DATE 00/00/00

ARI PROJECT NO.
00000

PREPARED BY

TEST, ADJUST, AND BALANCE REPORT

THE DATA PRESENTED IN THIS REPORT IS AN EXACT RECORD OF SYSTEM PERFORMANCE AND WAS OBTAINED IN COMPLIANCE WITH NEBB PROCEDURAL STANDARDS. ANY VARIANCES FROM THE DESIGN REQUIREMENTS ARE NOTED THROUGHOUT THIS REPORT.

THE RESULTS SHOWN AND INFORMATION PRESENTED IN THIS REPORT IS CERTIFIED TO BE ACCURATE AND COMPLETE, TO THE EXTENT POSSIBLE BY EQUIPMENT AND PROCEDURES USED THROUGHOUT THIS SURVEY.

THE FINAL AIR DISTRIBUTION SYSTEM MEASUREMENTS ARE IN COMPLIANCE WITH NEBB "PROCEDURAL STANDARDS FOR TESTING, ADJUSTING AND BALANCING OF ENVIRONMENTAL SYSTEMS" AND THE PROJECT SPECIFICATIONS.

CERTIFIED BY: ______________________________
DATE: ______________

THE FINAL HYDRONIC DISTRIBUTION SYSTEM MEASUREMENTS ARE IN COMPLIANCE WITH NEBB "PROCEDURAL STANDARDS FOR TESTING, ADJUSTING AND BALANCING OF ENVIRONMENTAL SYSTEMS" AND THE PROJECT SPECIFICATIONS.

CERTIFIED BY: ______________________________
DATE: ______________

FAN TEST DATA SUMMARY

PROJECT: ______________________________

UNIT	DESIGN CFM	FINAL CFM	NOTES

PAGE: 00

AIR APPARATUS TEST REPORT

PROJECT: ______________ UNIT: ______________

AIR HANDLER UNIT DATA	
MANUFACTURER	
MODEL	
SERIAL	
FAN TYPE	
RATING	
OUTLET CFM	

MOTOR NAMEPLATE DATA			
MANUFACTURER			
HP	R.P.M.		
PHASE	FRAME		
S.F.	TYPE		
VOLTS	AMPS		
MOTOR IS THERMALLY PROTECTED			

STARTER DATA	
MANUFACTURER	
MODEL	
SIZE	
RANGE	
SET AT (AMPS)	
REQ. THERMAL	

DRIVE DATA	
MOTOR DRIVE	
MOTOR BORE	
FAN DRIVE	
FAN BORE	
NO. BELTS & SIZE	
CENTERLINE	
DRIVE CHANGE	

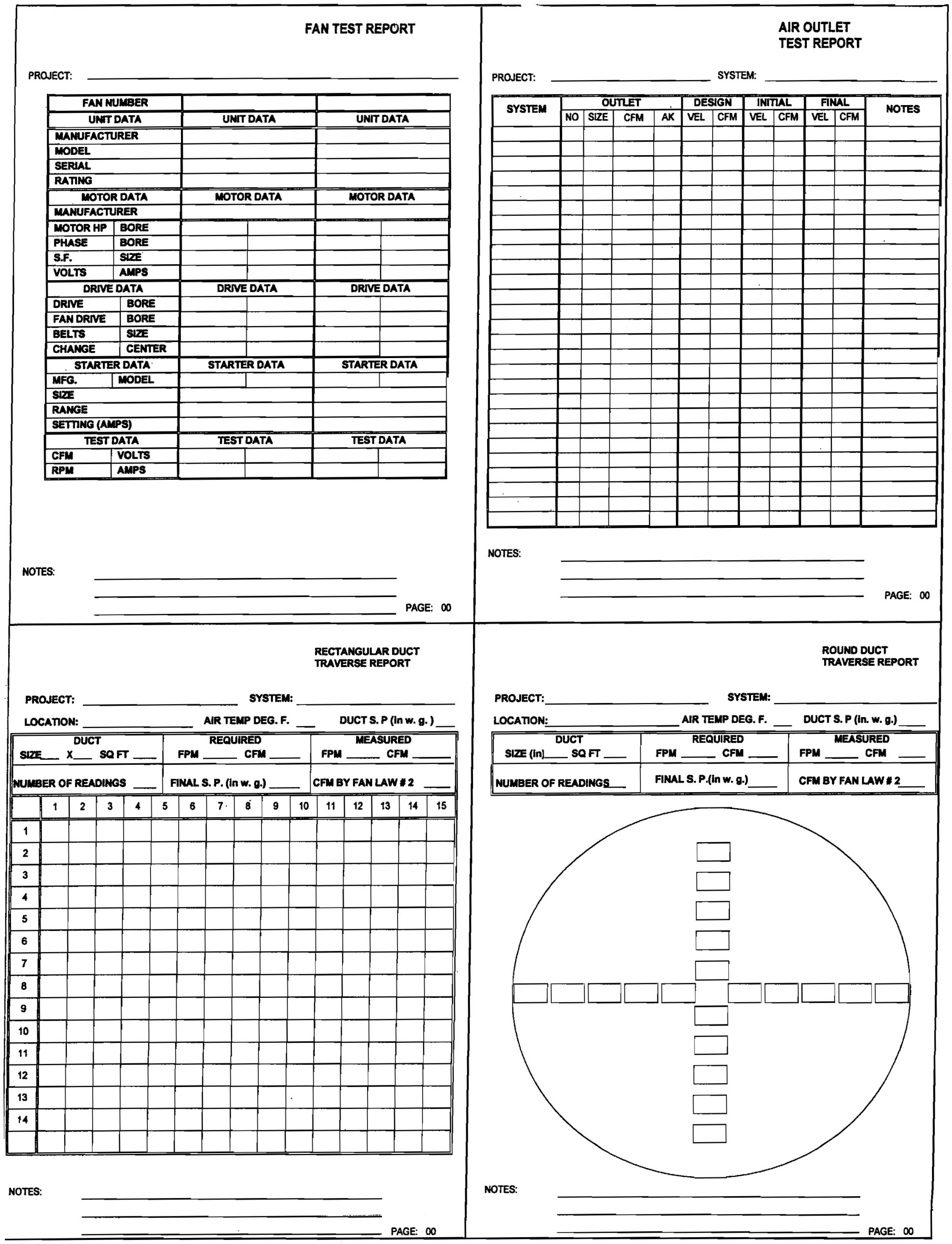
AIRFLOW DATA	
SUPPLY CFM	
OSA CFM	
RETURN CFM	

TEST DATA	INITIAL	FINAL
FAN R.P.M.		
VOLTAGE		
AMPERAGE		

NOTES: ______________________________

PAGE: 00

Figure 40.—Sample certified test, adjust, and balance report, page 1 of 4.



FAN TEST REPORT

PROJECT: ____

FAN NUMBER					
UNIT DATA		UNIT DATA		UNIT DATA	
MANUFACTURER					
MODEL					
SERIAL					
RATING					
MOTOR DATA		MOTOR DATA		MOTOR DATA	
MANUFACTURER					
MOTOR HP	BORE				
PHASE	BORE				
S.F.	SIZE				
VOLTS	AMPS				
DRIVE DATA		DRIVE DATA		DRIVE DATA	
DRIVE	BORE				
FAN DRIVE	BORE				
BELTS	SIZE				
CHANGE	CENTER				
STARTER DATA		STARTER DATA		STARTER DATA	
MFG.	MODEL				
SIZE					
RANGE					
SETTING (AMPS)					
TEST DATA		TEST DATA		TEST DATA	
CFM	VOLTS				
RPM	AMPS				

NOTES: ____

PAGE: 00

AIR OUTLET TEST REPORT

PROJECT: ____ SYSTEM: ____

SYSTEM	OUTLET				DESIGN		INITIAL		FINAL		NOTES
	NO	SIZE	CFM	AK	VEL	CFM	VEL	CFM	VEL	CFM	

NOTES: ____

PAGE: 00

RECTANGULAR DUCT TRAVERSE REPORT

PROJECT: ____ SYSTEM: ____

LOCATION: ____ AIR TEMP DEG. F. ___ DUCT S. P (in w. g.) ___

DUCT	REQUIRED	MEASURED
SIZE ___ X ___ SQ FT ___	FPM ___ CFM ___	FPM ___ CFM ___
NUMBER OF READINGS ___	FINAL S. P. (in w. g.) ___	CFM BY FAN LAW # 2 ___

	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15
1															
2															
3															
4															
5															
6															
7															
8															
9															
10															
11															
12															
13															
14															

NOTES: ____

PAGE: 00

ROUND DUCT TRAVERSE REPORT

PROJECT: ____ SYSTEM: ____

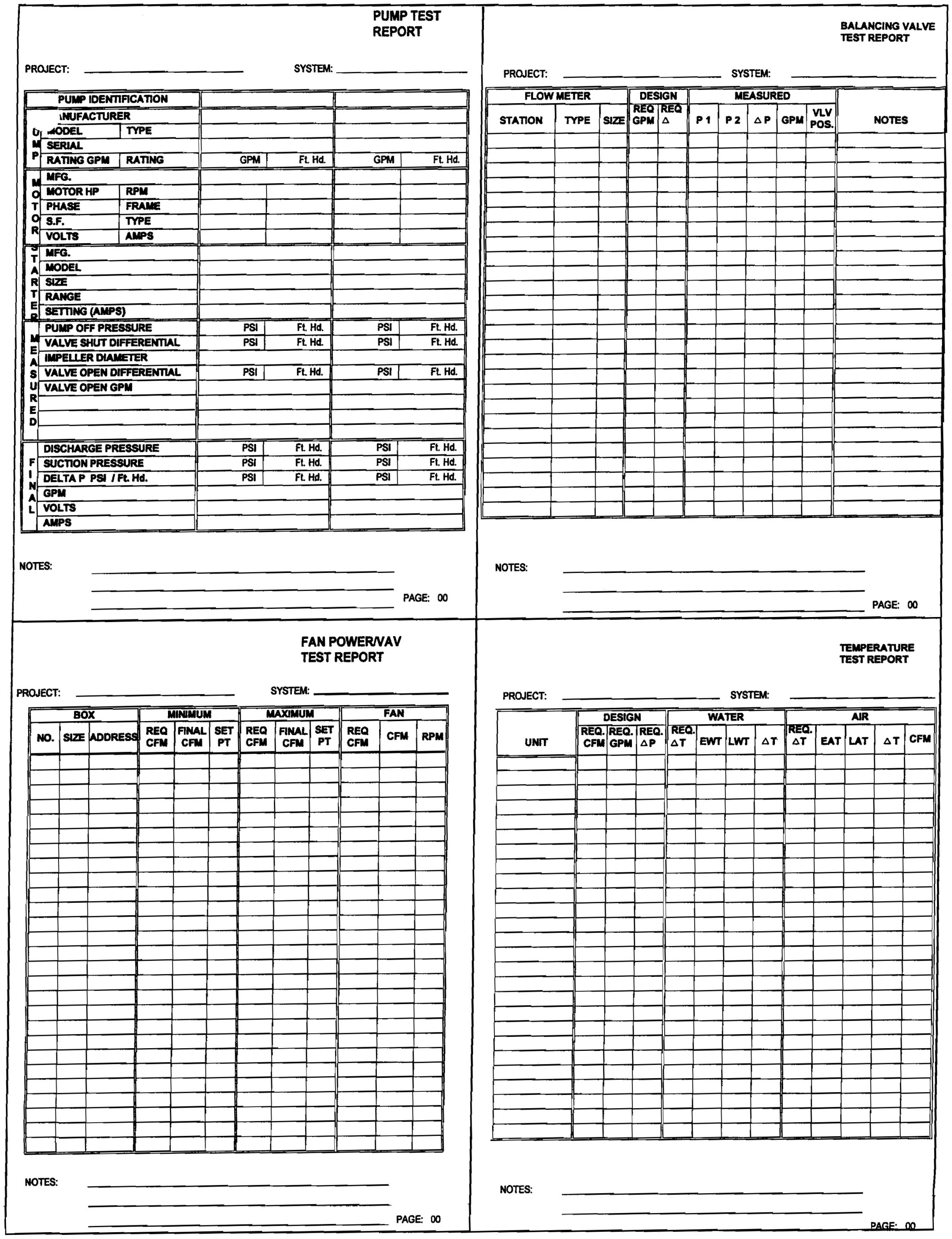
LOCATION: ____ AIR TEMP DEG. F. ___ DUCT S. P (in. w. g.) ___

DUCT	REQUIRED	MEASURED
SIZE (in) ___ SQ FT ___	FPM ___ CFM ___	FPM ___ CFM ___
NUMBER OF READINGS ___	FINAL S. P.(in w. g.) ___	CFM BY FAN LAW # 2 ___

NOTES: ____

PAGE: 00

Figure 40.—Sample certified test, adjust, and balance report, page 2 of 4.



PUMP TEST REPORT

PROJECT: ______________ SYSTEM: ______________

	PUMP IDENTIFICATION					
PUMP	MANUFACTURER					
	MODEL	TYPE				
	SERIAL					
	RATING GPM	RATING	GPM	Ft. Hd.	GPM	Ft. Hd.
MOTOR	MFG.					
	MOTOR HP	RPM				
	PHASE	FRAME				
	S.F.	TYPE				
	VOLTS	AMPS				
STARTER	MFG.					
	MODEL					
	SIZE					
	RANGE					
	SETTING (AMPS)					
MEASURED	PUMP OFF PRESSURE		PSI	Ft. Hd.	PSI	Ft. Hd.
	VALVE SHUT DIFFERENTIAL		PSI	Ft. Hd.	PSI	Ft. Hd.
	IMPELLER DIAMETER					
	VALVE OPEN DIFFERENTIAL		PSI	Ft. Hd.	PSI	Ft. Hd.
	VALVE OPEN GPM					
FINAL	DISCHARGE PRESSURE		PSI	Ft. Hd.	PSI	Ft. Hd.
	SUCTION PRESSURE		PSI	Ft. Hd.	PSI	Ft. Hd.
	DELTA P PSI / Ft. Hd.		PSI	Ft. Hd.	PSI	Ft. Hd.
	GPM					
	VOLTS					
	AMPS					

NOTES: ______________

PAGE: 00

BALANCING VALVE TEST REPORT

PROJECT: ______________ SYSTEM: ______________

FLOW METER			DESIGN		MEASURED					
STATION	TYPE	SIZE	REQ GPM	REQ Δ	P 1	P 2	Δ P	GPM	VLV POS.	NOTES

NOTES: ______________

PAGE: 00

FAN POWER/VAV TEST REPORT

PROJECT: ______________ SYSTEM: ______________

BOX			MINIMUM			MAXIMUM			FAN		
NO.	SIZE	ADDRESS	REQ CFM	FINAL CFM	SET PT	REQ CFM	FINAL CFM	SET PT	REQ CFM	CFM	RPM

NOTES: ______________

PAGE: 00

TEMPERATURE TEST REPORT

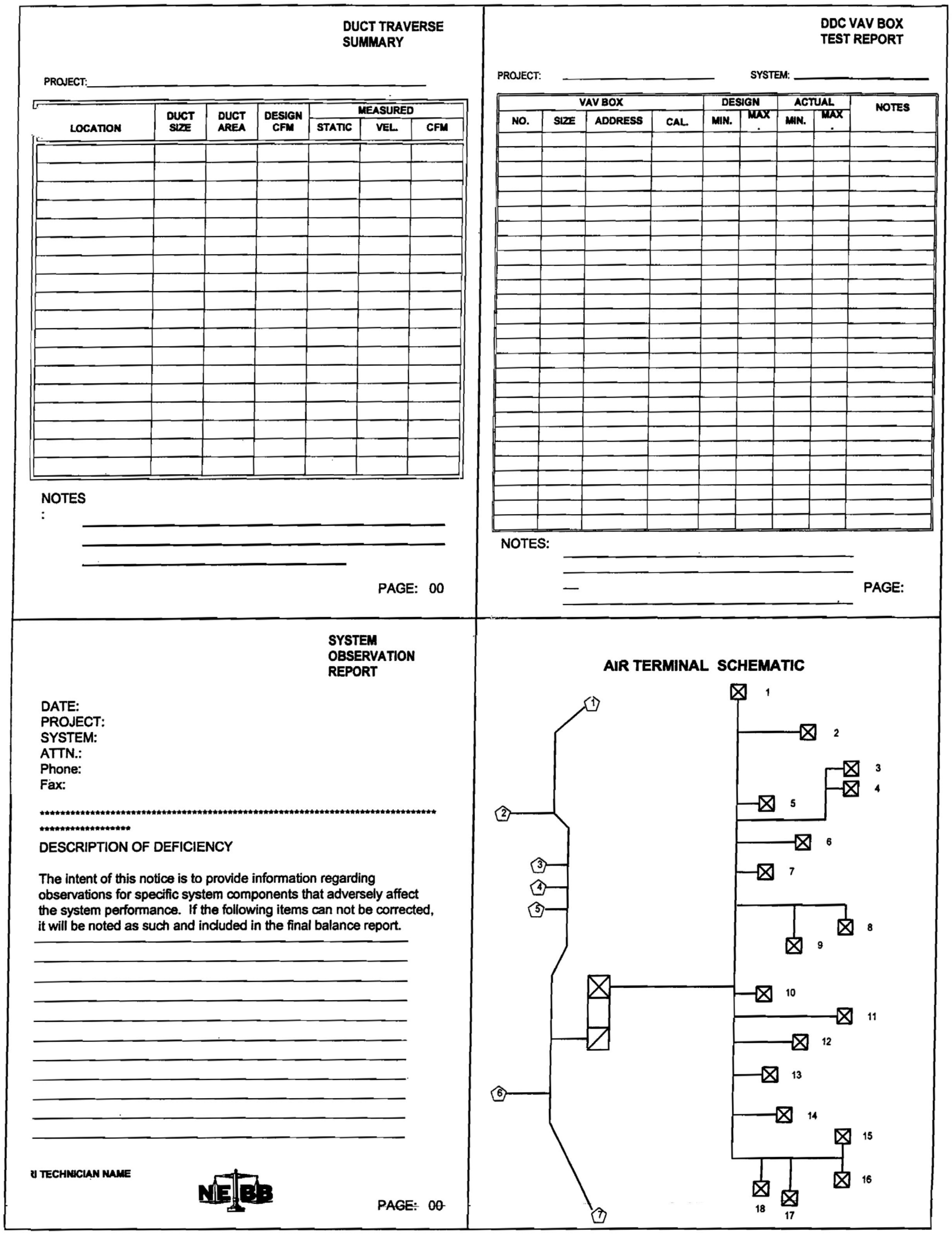
PROJECT: ______________ SYSTEM: ______________

	DESIGN			WATER				AIR				
UNIT	REQ. CFM	REQ. GPM	REQ. ΔP	REQ. ΔT	EWT	LWT	ΔT	REQ. ΔT	EAT	LAT	ΔT	CFM

NOTES: ______________

PAGE: 00

Figure 40.—Sample certified test, adjust, and balance report, page 3 of 4.



DUCT TRAVERSE SUMMARY

PROJECT:

LOCATION	DUCT SIZE	DUCT AREA	DESIGN CFM	MEASURED		
				STATIC	VEL.	CFM

NOTES
:

PAGE: 00

DDC VAV BOX TEST REPORT

PROJECT:
SYSTEM:

VAV BOX				DESIGN		ACTUAL		NOTES
NO.	SIZE	ADDRESS	CAL.	MIN.	MAX.	MIN.	MAX.	

NOTES:

PAGE:

SYSTEM OBSERVATION REPORT

DATE:
PROJECT:
SYSTEM:
ATTN.:
Phone:
Fax:

****

DESCRIPTION OF DEFICIENCY

The intent of this notice is to provide information regarding observations for specific system components that adversely affect the system performance. If the following items can not be corrected, it will be noted as such and included in the final balance report.

TECHNICIAN NAME

NEBB

PAGE: ~~00~~

AIR TERMINAL SCHEMATIC

Figure 40.—Sample certified test, adjust, and balance report, page 4 of 4.

4. **Leakage Testing.—**All duct systems experience some degree of air leakage. Leakage testing is very expensive and should be reserved for very critical systems where normal air leakage cannot be tolerated. This condition is rarely encountered on Reclamation facilities. Leakage testing is most likely to be conducted for duct systems in excess of 3-inch pressure where the additional expense can be justified. Leakage should be controlled by specifying the proper seal class based on the pressure class of ductwork. Careful inspection of the contractors work will also help reduce leakage. Detailed information on leakage testing can be found in the *SMACNA HVAC Air Duct Leakage Test Manual*.

5. **Sound and Vibration**.—Sound and vibration testing is rarely required in Reclamation facilities. Many of the conditions that cause excessive sound and vibration and the preventive measures were discussed elsewhere in this manual. When necessary, testing should be conducted by an *NEBB* certified firm and testing should be conducted in accordance with *NEBB Procedural Standards for Measuring Sound and Vibration*.

6. **Commissioning**.—The goal of commissioning is similar, but not the same as, the TAB. In many cases the commissioning procedures include the TAB procedures and both may be conducted simultaneously. *ASHRAE* and *SMACNA* have published manuals describing the commissioning process in great detail and also include guide specifications. *SMACNA* identifies three levels of commissioning which can be applied to an HVAC system. These levels are briefly described below. For detailed checklist and procedures, refer to *SMACNA HVAC Systems Commissioning Manual*.

    a. **Level 1: Basic Commissioning.—**Basic commissioning is intended to accomplish the following objectives:

        (1) Ensure the contractor provides a fully functional system in accordance with the specifications.

        (2) Enable planning and coordination of work.

        (3) Ensure that systems operation, including control sequences, are checked and that functional performance is achieved.

        (4) Identification and resolution of problems as they occur.

(5) Provide documentation that system installation and operation conforms to specifications.

(6) Demonstration of system performance and functionality.

(7) Provide demonstration and training on system operation and maintenance to designated personnel.

b. **Level 2: Comprehensive Commissioning.**—Comprehensive commissioning is required when the Reclamation mandates specific requirements or procedures that the contractor is obliged to perform. These procedures or conditions must be fully explained in the specifications requirements. Level 2 includes:

(1) Designer and owner review of the commissioning plan.

(2) Comprehensive pre-start, start-up, and functional performance checks and tests. For detailed checklist and procedures, refer to *SMACNA HVAC Systems Commissioning Manual*.

(3) Special attention to interfaces between the HVAC system, and other systems such fire detection and alarm systems.

c. **Level 3: Critical Systems**.—Level 3 commissioning is the most rigorous level and is reserved for systems where performance is extremely important. For these systems, verification of functional performance is not sufficient. Testing must also verify that the required environmental conditions (temperature, humidity, pressure, etc.) are achieved and maintained. Generally, the building owner dictates the need for Level 3 commissioning requirements according to the specific needs of the installation. However, in some cases the requirements are dictated by existing codes or regulations and the test procedures must be witnessed and certified by a code enforcement officer. Level 3 commissioning includes the following:

(1) ***Fire Safety***.—Systems providing fire alarm signaling, fire suppression, smoke control, or evacuation or responding to a fire condition such as shutdown or smoke purge (100 percent exhaust). Reclamation applications that may require level 3 commissioning are the stairwell pressure systems, which must minimize smoke entry during emergency egress and simultaneously prevent over pressurization that could make doors difficult to open.

(2) ***Life Safety.***—Systems in which failure could threaten the health or safety of personnel. Reclamation systems that fall in this category may include ventilating systems for laboratories. Confined space ventilating systems may also be included, especially those with a history of contamination by hazardous substances such as hydrogen sulfide, carbon monoxide, and methane.

(3) ***Automatic Backup.***—Critical systems that justify standby or redundant equipment designed for automatic activation in the event of a failure of the primary equipment. The means for detecting and responding to failure must be covered.

(4) ***Cost of a Failure.***—Systems in which a failure may result in significant cost due to equipment damage or shutdown.

References

ASHRAE (American Society of Heating, Refrigeration and Air Conditioning Engineers)

Handbook: Fundamentals (1997 – Revised every 4 years)

Handbook: HVAC Systems and Equipment (1992 – Revised every 4 years)

Handbook: HVAC Applications (1991 – Revised every 4 years)

Handbook: Refrigeration (1990 – Revised every 4 years)

ASHRAE Standard ANSI/ASHRAE 15-1992 – Safety Code for Mechanical Refrigeration

ASHRAE Standard 62-1989 – Ventilation for Acceptable Indoor Air Quality

ASHRAE Guideline 1-1996 – HVAC Commissioning Process

Design of Smoke Management Systems

A Practical Guide to Noise and Vibration Control (1991)

NFPA (National Fire Protection Association)

NFPA 12 – Carbon Dioxide Extinguishing Systems

NFPA 54 – National Fuel Gas Code

NFPA 55 – Compressed and Liquefied Gases in Portable Cylinders

NFPA 69 – Standard on Explosion Prevention

NFPA 70 – National Electric Code

NFPA 90A – Installation of Air Conditioning and Ventilating Systems

NFPA 90B – Installation of Warm Air Heating and Air Conditioning Systems

NFPA 91 – Installation of Exhaust Systems for Air Conveying of Materials

NFPA 92A – Smoke Control Systems

NFPA 101 – Life Safety Code

NFPA 204M – Smoke and Heat Venting

NFPA 820 - Standard for Fire Protection in Wastewater Treatment and Collection Facilities

NFPA 850 - Recommended Practice for Fire Protection for Electric Generating Plants and High Voltage Direct Current Converter Stations

NFPA 852 – Hydroelectric Generating Plants

SMACNA (Sheet Metal and Air Conditioning Contractors National Association, Inc.)

SMACNA Duct Construction Standards – Metal and Flexible (1995)

SMACNA Round Industrial Duct Construction Standards (1977)

SMACNA Thermoplastic Duct (PVC) Construction Manual (1994)

SMACNA Thermoset FRP Duct Construction Manual (1997)

SMACNA Fibrous Glass Duct Construction Standards (1992)

SMACNA HVAC System Duct Design (1990)

SMACNA Fire, Smoke, and Radiation Damper Installation Guide (1992)

SMACNA HVAC Systems – Testing, Adjusting and Balancing (1993)

SMACNA HVAC System Commissioning Manual (1994)

ICBO (International Conference of Building Officials)

UBC – Uniform Building Code (Revised every three years)

UMC – Uniform Mechanical Code (Revised every three years)

UFC – Uniform Fire Code (Revised every three years)

IAPMO (International Association of Plumbing and Mechanical Officials

UPC – Uniform Plumbing Code (Revised every three years)

Reclamation Publications

Engineering Monograph No. 44 – Ozone Abatement in Air-Cooled Hydroelectric Generators (1997)

Reclamation Safety and Health Standards

American Conference of Government Industrial Hygienists

Industrial Ventilation – A Manual of Recommended Practice (21st. edition, 1992 Revised every 2 to 4 years)

Jennings, Burgess H., Environmental Engineering: Analysis and Practice, International Textbook Company, 1970.

Wendes, Herb, HVAC Retrofits: Energy Savings Made Easy, Fairmont Press, 1994.

Ottaviano, Victor B. and John L., National Mechanical Estimator, Fairmont Press, 1996.

Schaffer, Mark E., A Practical Guide to Noise and Vibration Control for HVAC Systems, American Society of Heating

Dossat, Roy J., and Horan, Thomas J., Principles of Refrigeration, Prentice-Hall

Organizations Associated with Codes and Standards Commonly Used for HVAC Design Work

ABMA	American Boiler Manufacturers Association
ACCA	Air Conditioning Contractors of America
ACGIH	American Conference of Governmental Industrial Hygienists
ADC	Air Diffusion Council
AGA	American Gas Association
AGMA	American Gear Manufacturers Association
AHA	Association of Home Appliance Manufacturers
AISI	American Iron and Steel Institute
AISC	American Institute of Steel Construction
AMCA	Air Movement and Control Association
ANSI	American National Standards Institute
ARI	Air-Conditioning and Refrigeration Institute
ASA	Acoustical Society of America
ASHRAE	American Society of Heating, Refrigerating, and Air-Conditioning Engineers, Inc.
ASME Intn’l	The American Society of Mechanical Engineers
ASTM	American Society for Testing and Materials
AWS	American Welding Society, Inc.
AWWA	American Water Works Association
CAGI	Compressed Air and Gas Institute
CID	Commercial Item Description
CMAA	Crane Manufacturers Association of America
CTI	Cooling Tower Institute
FS	Federal Specifications
HEI	Heat Exchange Institute

HI	Hydraulic Institute
HYDI	Hydronics Institute
IAPMO	International Association of Plumbing and Mechanical Officials
ICBO	International Conference of Building Officials
ICC	International Code Council
IEEE	Institute of Electrical and Electronics Engineers
ISA	The International Society for Measurement and Control
ISO	International Standards Organization
MICA	Midwest Insulation Contractors Association
MSS	Manufacturers Standardization Society of the Valve and Fittings Industry, Inc.
NAIMA	North American Insulation Manufacturers Association
NEBB	National Environmental Balancing Bureau
NEMA	National Electrical Manufacturers Association
NFPA	National Fire Protection Association
OSHA	Occupational Health and Safety Act
SMACNA	Sheet Metal and Air Conditioning Contractors' National Association
TEMA	Tubular Exchanger Manufacturers Association, Inc.
UL	Underwriters Laboratories, Inc.
UBC	Uniform Building Code
UMC	Uniform Mechanical Code

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September 21, 2006

Glossary of Terms and Abbreviations

AC	air conditioning
AHU	air handling unit
ATD	air terminal device
BTU	British Thermal Unit (quantity of heat required to raise 1 pound of water by 1 ºF)
BTUH	British Thermal Units per Hour
ºC	degrees Celsius
cfm	cubic feet per minute
CV	constant volume
db	dry bulb
dbA	sound pressure level, using "A" weighting filter
DOC	Designer's Operating Criteria
DX	direct expansion (heat exchanger coils)
ºF	degrees Fahrenheit
FCU	fan coil unit
inch w.g.	inch of water gauge - common name for the static air pressure to raise an inch of water column. The word gauge after a pressure reading indicates that the pressure stated is actually the difference between the absolute, or total, pressure and the air pressure at the time of the reading.
LFL	lower flammable limit
NC	Noise Criteria
OA	outside air
PEL	personnel exposure limits
RA	return air
rpm	revolutions per minute
SOP	Standard Operating Procedures

TAB	Testing, Adjusting, and Balancing
TLV	threshold limit values
UPS	uninterrupted power supply
VAV	variable air volume
wb	wet bulb