ENGINEERING COOKBOOK

LOREN COOK COMPANY

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1941-2016

A Handbook
for the
Mechanical
Designer

Engineering Cookbook

A Handbook For The Mechanical Designer Third Edition

This handy pocket reference is a token of LOREN COOK COMPANY's appreciation to the many fine mechanical designers in our industry. It provides access to frequently needed information:

- Fan Basics
- System Design
- Duct Design
- Motors & Drives
- Heating & Refrigeration
- Formulas & Conversion Factors

Downloads

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Loren Cook Company

Springfield, Missouri, USA

Table of Contents

Fan Basics

Fan Types

Fans are described in three types and several sub-types.

Axial
- Propeller
- Tube Axial
- Vane Axial

Mixed Flow
- Low Pressure
- High Pressure
- Extended Pressure

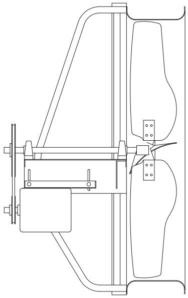
Centrifugal
- Airfoil
- Backward Inclined
- Radial
- Forward Curved

Axial Fans

An axial fan discharges air parallel to the axis of the impeller rotation.

Propeller Fans

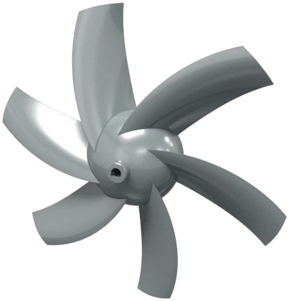
- Cost is generally low
- Blades attached to a relatively small hub
- Energy transfer is primarily in the form of velocity pressure
- Primarily for low pressure, high volume, non-ducted applications

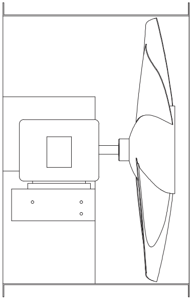


Cutaway View

Tube Axial Fans

- More useful static pressure range than propeller fans
- Axial impeller in a tubular housing
- Blade cross-section either single thickness or airfoil
- Primarily for low to medium pressure applications such as ducted HVAC applications where downstream air distribution is not critical

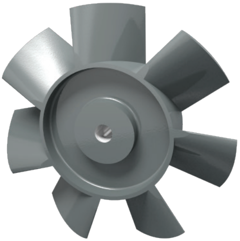


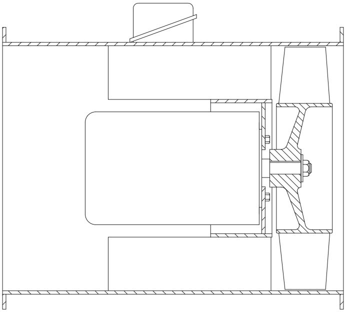


Cutaway View

Vane Axial Fans

- Axial impeller in a tubular housing with straightening vanes which improve pressure capabilities & efficiency
- Cost is generally high
- Generate medium-to-high pressure at good efficiencies
- Fixed or adjustable pitch blades
- Hub diameter is greater than 50% of impeller diameter
- Used for straight-through flow and compact installations
- Downstream air distribution is good
- Noise levels may be high
- Used for general HVAC in a wide range of pressure applications
- Industrial applications include: drying ovens, paint spray booths, and fume exhaust systems





Cutaway View

Mixed Flow Fans

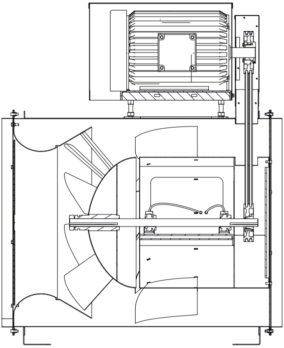
A mixed flow fan is a hybrid of an axial and a centrifugal fan, with a smaller backplate and angled blades. It discharges air parallel to the axis of the impeller rotation. As a general rule, it is more efficient, smaller and quieter when used in ducted, inline, supply and exhaust.

- Best selection for most inline airflow applications
- Higher pressure ability than axial
- Lower sound levels than equally sized centrifugal inline units
- Lower rpm than equal sized inline units, for the same flow and pressure
- Does not require inlet bell or outlet cone

Low Pressure

High Pressure

Extended Pressure

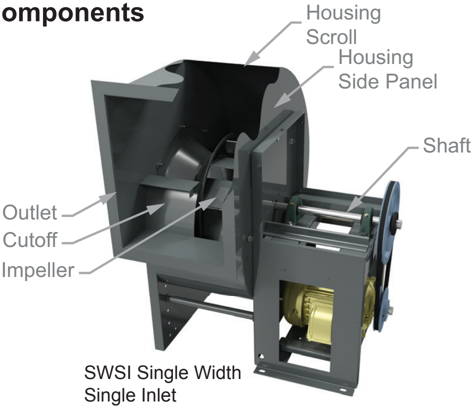


Cutaway View

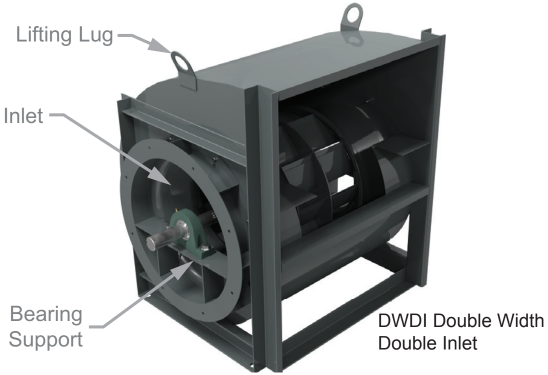
Centrifugal Fans

A centrifugal fan draws in air parallel to the axis of rotation and discharges air perpendicular to the axis of rotation. The air then follows the shape of the fan housing to exit. In general, centrifugal fans are preferred for ducted or higher-pressure systems.

Basic Components



SWSI Single Width Single Inlet



DWDI Double Width Double Inlet

Centrifugal fans are the ‘workhorse’ of the fan industry, operating in a wide variety of configurations, duties and environments. From light commercial, non-ducted applications to high pressure industrial and process ventilation, centrifugal fans can often be configured to meet the application demands. Examples are shown on these two pages.

Centrifugal Power
Roof Ventilator

Centrifugal
Kitchen Exhaust

Square Inline
Centrifugal

Fan Basics

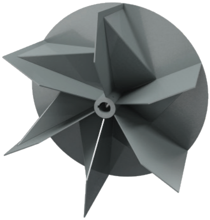
Airfoil (AF) Fans

- Highest efficiency of centrifugal impeller designs
- 10-12 airfoil blades inclined away from the direction of rotation
- Relatively deep blades provide for efficient expansion
- Primarily used in general HVAC systems
- Industrial use in clean air applications

Backward Inclined (BI) Fans

- Efficiency slightly less than airfoil
- 10-12 single thickness blades inclined away from the direction of rotation
- Relatively deep blades provide for efficient expansion within blade passages
- Primarily used in general HVAC systems
- Industrial applications where airfoil blade is not acceptable due to a corrosive and/or erosive airstream

Radial Fans



- Less efficient than airfoil or backward inclined fans
- High mechanical strength
- High speed at given duty relative to other designs
- 6-10 blades arrayed in a radial configuration
- Primarily used for material handling applications in industrial environments
- Often specially coated to resist corrosion or surface erosion
- Used in high pressure applications not commonly found in HVAC systems.

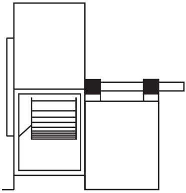
Forward Curved (FC) Fans

- Less efficient than airfoil or backward inclined fans
- Low cost to manufacture
- Lightweight construction
- 24-64 shallow blades with blade heel and tip curved in direction of rotation
- Smallest wheel diameter for given duty of all centrifugal fans types
- Most efficient at low speeds
- Primary uses are low pressure HVAC applications such as residential furnaces and packaged air conditioning equipment

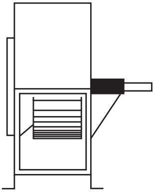
Fan Basics

Drive Arrangements for Centrifugal Fans

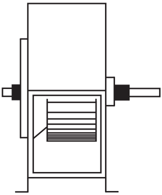
SWSI: Single Width, Single Inlet
DWDI: Double Width, Double Inlet



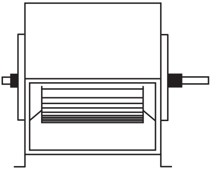
Arr. 1 SWSI: Belt drive or direct drive. Impeller overhung. Two bearings on pedestal.



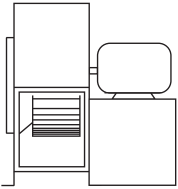
Arr. 2 SWSI: Belt drive or direct drive. Bearings on bracket supported by fan housing.



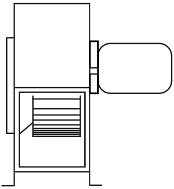
Arr. 3 SWSI: Belt drive or direct drive. One bearing on each side supported by fan housing.



Arr. 3 DWDI: Same configuration as Arrangement 3 SWSI

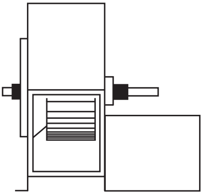


Arr. 4 SWSI: Direct drive. Impeller overhung on motor shaft. No bearings on fan. Motor mounted on pedestal.

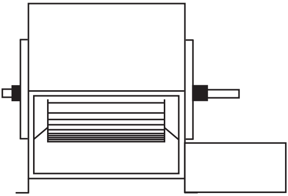


Arr. 5 SWSI: Direct drive. Impeller overhung on motor. No bearings on fan. Motor mounted to fan housing.

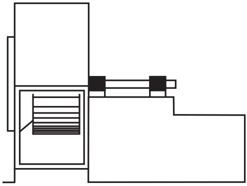
Rotation is viewed from drive side



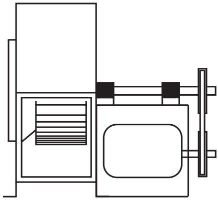
Arr. 7 SWSI: Direct drive, coupling connection. Arrangement 3 plus base for motor.



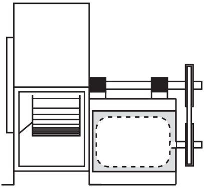
Arr. 7 DWDI: Direct drive, coupling connection. Arrangement 3 plus base for motor.



Arr. 8 SWSI: Direct drive, coupling connection. Arrangement 1 plus extended base for motor.



Arr. 9 SWSI: Belt drive. Impeller overhung, two bearings, with motor outside base.

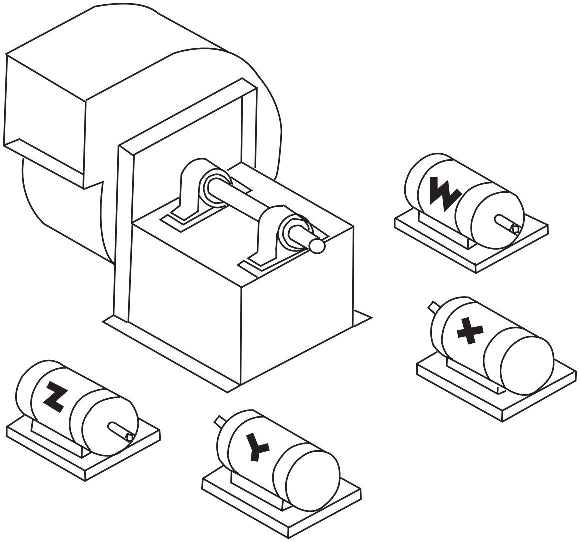


Arr. 10 SWSI: Belt drive. Impeller overhung, two bearings, with motor inside base.

Adapted from ANSI/AMCA Standard 99-10

Motor Positions and Rotation for Belt Drive Centrifugal Fans

To determine fan rotation and motor location, face the fan from the drive side and identify the rotation as either clockwise or counterclockwise and the motor position designated by the letters W, X, Y or Z as shown in the drawing below.



Adapted from ANSI/AMCA Standard 99-10

Inlet Boxes

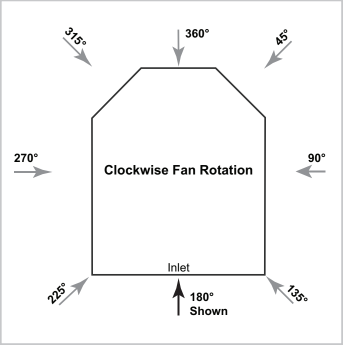
Position of inlet box and air entry to inlet box is determined from the drive side as defined below:

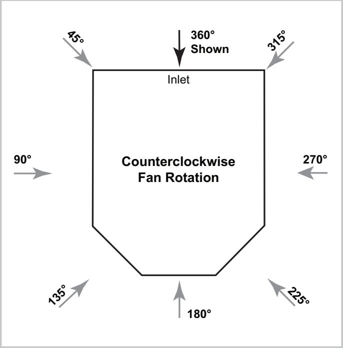
On single inlet fans, the drive side is the side opposite of the fan inlet.

On double inlet fans with a single driver, the side with the driver is considered the drive side.

Position of inlet box is determined in accordance with diagrams. Angle of air entry to box is referred to the top vertical axis of fan in degrees as measured in the direction of fan rotation. Angle of air entry to box may be any intermediate angle as required.

Positions 135° to 225° in some cases may interfere with floor structure.

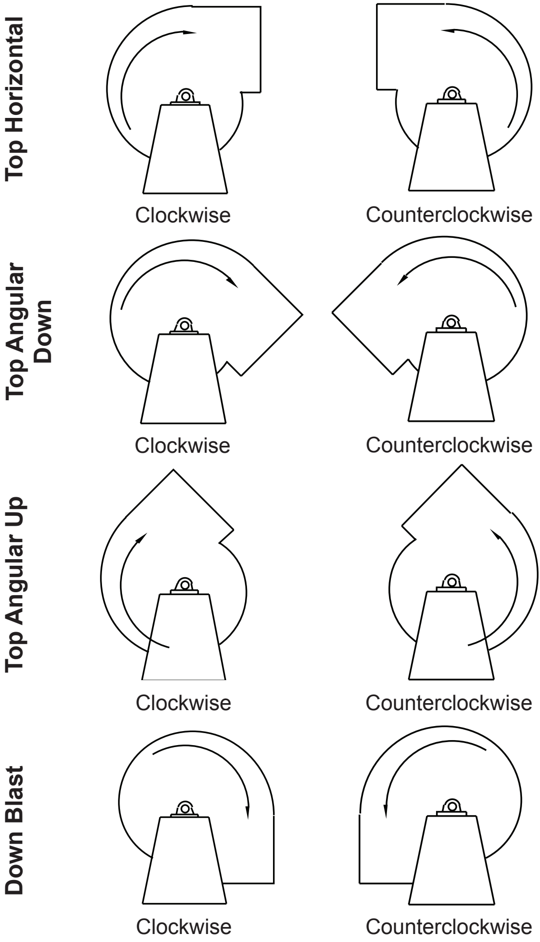




Adapted from AMCA Standard 99-2405

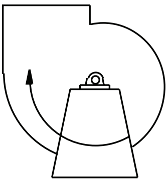
Fan Basics

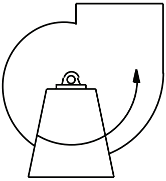
Rotation & Discharge Designations
Centrifugal Fans



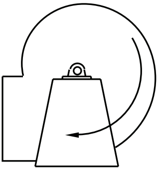
Rotation is viewed from drive side

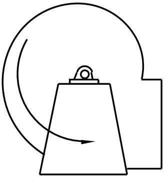
Up Blast


Clockwise

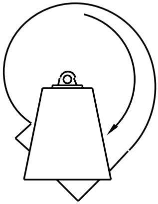

Counterclockwise

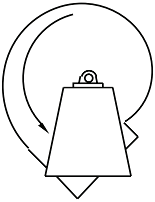
Bottom Horizontal


Clockwise

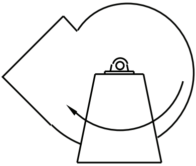

Counterclockwise

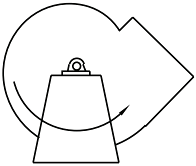
Bottom Angular Down


Clockwise


Counterclockwise

Bottom Angular Up


Clockwise


Counterclockwise

Adapted from ANSI/AMCA Standard 99-10

Spark Resistant Construction

Fan and damper applications may involve the handling of potentially explosive or flammable particles, fumes, or vapors. Such applications require careful consideration to insure the safe handling of such gas streams. AMCA Standard 99-0401-10 deals only with the fan and/or damper unit installed in that system. The standard contains guidelines to be used by the manufacturer and user to establish general methods of construction. The exact method of construction and choice of alloys is the responsibility of the manufacturer, however, the customer must accept both the type and design with full recognition of the potential hazard and the degree of protection required.

Type A

- All parts of the fan or damper in contact with the air or gas being handled and subject to impact by particles in the airstream shall be made of non-ferrous material. Ferrous shafts/axles and hardware exposed to the airstream shall be covered by non-ferrous materials.
- **Fans only**: Steps must also be taken to assure that the impeller, bearings, and shaft are adequately attached and/or restrained to prevent a lateral or axial shift in these components.
- **Dampers only:** Construction shall ensure that linkages, bearings and blades are adequately attached or restrained to prevent independent action. Ferrous containing bearings are acceptable if the bearings are located out of the airstream and shielded from particle impact.

Type B

- **Fans only:** The fan shall have a non-ferrous impeller and non-ferrous ring about the opening through which the shaft passes. Ferrous hubs, shafts, and hardware are allowed provided construction is such that a shift of impeller or shaft will not permit two ferrous parts of the fan to rub or strike. Steps must also be taken to assure that the impeller, bearings, and shaft are adequately attached and/or restrained to prevent a lateral or axial shift in these components.

- **Dampers only:** Construction shall ensure that linkages, bearings, and blades are adequately attached or restrained to prevent independent action. Damper blades shall be non-ferrous.

Type C

- **Fans only:** The fan shall be so constructed that a shift of the impeller or shaft will not permit two ferrous parts of the fan to rub or strike.
- **Dampers only:** Construction shall ensure that linkages, bearings, and blades are adequately attached or restrained to prevent independent action. Damper blades shall be non-ferrous.

Notes for AMCA Type A, B & C Construction:

1. No bearings, drive components or electrical devices shall be placed in the air or gas stream unless they are constructed or enclosed in such a manner that failure of that component cannot ignite the surrounding gas stream.
2. The user shall electrically ground all fan and/or damper parts.
3. Non-ferrous material shall be any material with less than 5% iron or any other material with demonstrated ability to be spark resistant.
4. The use of aluminum or aluminum alloys in the presence of steel which has been allowed to rust requires special consideration. Research by the U.S. Bureau of Mines and others has shown that aluminum impellers rubbing on rusty steel may cause high intensity sparking.
5. All structural components within the airstream, including non-metallic materials, must be suitable for conducting static charge safely to ground, thus preventing buildup of electrical potential. Dampers with non-metallic bearings must include means by manufacturer of transferring electrical charge from the blades to suitable ground.

This Standard in no way implies a guarantee of safety for any level of spark resistance. “Spark resistant construction also does not protect against ignition of explosive gases caused by catastrophic failure or from any airstream material that may be present in a system.”

Adapted from AMCA Standard 99-401-86

Basic Terms

- **Air Flow (Q)**: Amount of air moved per rate of time, typically measured in cubic feet of air per minute (CFM).
- **Static Pressure (Ps)**: Resistance against airflow by the system (ductwork, fittings, dampers, filters, etc.). Typically measured in inches of water gauge (in. wg.) For most applications Static Pressure along with CFM is used for fan selection.
- **Total Pressure (Pt)**: The amount of pressure exerted by airflow on anything directly in the airstream.
- **Velocity Pressure (Pv)**: Directly related to the velocity of the airflow at any given point in a system. Used to calculate the airflow at any point in a system. Cannot be measured directly and is calculated as the difference between Total Pressure and Static Pressure.

$$Pv = Pt - Ps$$

- **Velocity (V)**: Speed of air in the direction of flow. Measured in feet per minute (FPM).
- **Power (HP)**: Rate of doing work, typically measured in Horsepower. For rotating machinery power is the amount of torque applied to a shaft to maintain a given rotating speed (RPM).

$$HP = RPM \times torque\ (ft\text{-}lb) / 5252.$$

$$1\ HP = 33{,}000\ foot\text{-}lbs\ per\ minute.$$

- **Brake Horsepower (BHP)**: (as listed in a fan performance table) The amount of HP required at the fan shaft to move the specified volume of air against a given system resistance. It does not include drive losses.

Fan Selection Criteria

Before selecting a fan, the following information is needed.

- Airflow required (CFM)
- Static pressure or system resistance (in. wg.)
- Air density or altitude and temperature
- Type of service
  - Environment type
  - Vapors / materials to be exhausted
  - Operation temperature
- Space requirements (including service access)
- Fan type (see Fan Basics)
- Drive type (direct or belt)
- Allowable noise levels
- Number of fans
- System configuration
  - Supply
  - Exhaust
  - Inline
  - Recirculating
  - Reversible
- Rotation
- Motor position
- Fan Class
- Expected fan life in years

Fan Testing

Fans are tested and performance certified under ideal laboratory conditions. When fan performance is measured in field conditions, the difference between the ideal laboratory condition and the actual field installation must be considered. Consideration must also be given to fan inlet and discharge connections as they can dramatically affect fan performance in the field. If possible, readings must be taken in straight runs of ductwork in order to ensure validity. If this cannot be accomplished, motor amperage and fan RPM should be used along with performance curves to estimate fan performance. Refer to Fan Installation Guidelines for more information.

Fan Classes of Operation

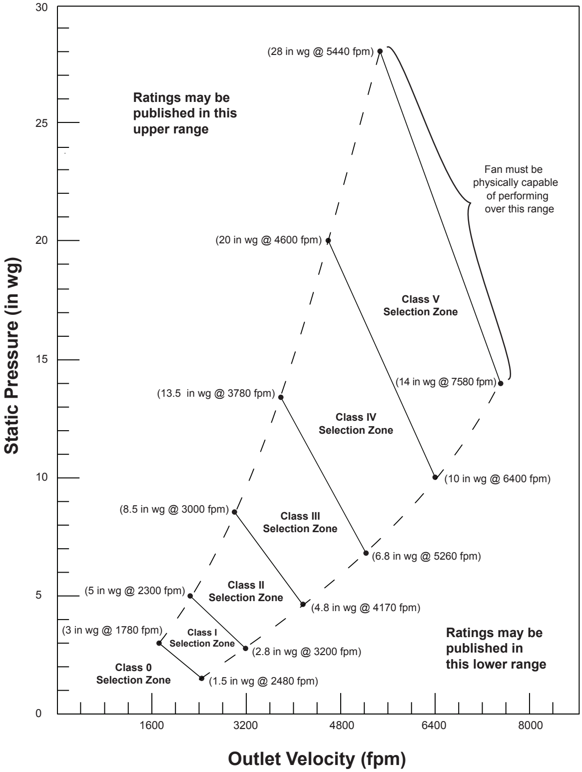
AMCA has defined classes of operation for various types of centrifugal fans and blowers. The class of operation is defined by a range of pressures and discharge velocities for a given configuration and type of blower. A fan must be physically capable of operating satisfactorily over the entire class range in order to be designated as meeting that fan class. Lower class numbers represent lower discharge velocities and static pressures (See AF and BI SWSI example on next page).

Although AMCA has defined the operating ranges of various classes, it does not prescribe how the fan manufacturer meets that requirement, leaving it up to the ingenuity of each to design the product as they see fit. Higher class operating ranges will often require heavier duty components resulting in a higher cost.

This is an important distinction. When specifying a fan class, the designer is stating that a fan be capable of operating over a specific range, and is not specifying a particular set of features or the physical construction of the fan.

With the advent of modern selection programs, the fan class is determined at the time of selection. For more information, refer to AMCA Standard 99.

Operating Limits - Centrifugal AF and BI Fans - SWSI



Performance Correction Factors for Altitude & Temperature

Unless otherwise noted, all published fan performance data is at standard operating conditions, 70°F at sea level.

When operating conditions vary significantly from standard, air density should be taken into consideration. When using COOK's Compute-A-Fan, these calculations will be made automatically based on input conditions.

Air Density Correction Factors for Altitude & Temperature

Altitude (feet)	Temperature (°F)							
	70	100	200	300	400	500	600	700
0	1.000	.946	.803	.697	.616	.552	.500	.457
1000	.964	.912	.774	.672	.594	.532	.482	.441
2000	.930	.880	.747	.648	.573	.513	.465	.425
3000	.896	.848	.720	.624	.552	.495	.448	.410
4000	.864	.818	.694	.604	.532	.477	.432	.395
5000	.832	.787	.668	.580	.513	.459	.416	.380
6000	.801	.758	.643	.558	.493	.442	.400	.366
7000	.772	.730	.620	.538	.476	.426	.386	.353
8000	.743	.703	.596	.518	.458	.410	.372	.340
9000	.714	.676	.573	.498	.440	.394	.352	.326
10000	.688	.651	.552	.480	.424	.380	.344	.315
15000	.564	.534	.453	.393	.347	.311	.282	.258
20000	.460	.435	.369	.321	.283	.254	.230	.210

Example: Fan pressure and horsepower vary directly with air density. Fan air volume, however, is not affected by air density. A cubic foot of air is always a cubic foot of air, it is just more or less dense depending primarily upon the conditions of altitude and temperature.

A fan selected from a catalog to operate at 10,000 CFM, 1" wg, and 3.5 BHP (standard air density), but operating at 5000 feet above sea level and 100°F would have a correction factor applied to the static pressure and horsepower of 0.787 as read from the table. The corrected static pressure (SP_c) and horsepower (BHP_c) would be:

$$SP_c = 0.787 \times 1 = 0.787\text{'' wg}$$

$$BHP_c = 0.787 \times 3.5 = 2.75\ BHP.$$

Fan Laws

CFM varies directly with RPM

$$CFM_1 / CFM_2 = RPM_1 / RPM_2$$

$$CFM_2 = (RPM_2 / RPM_1) \times CFM_1$$

SP varies with the square of the RPM

$$SP_1 / SP_2 = (RPM_1 / RPM_2)^2$$

$$SP_2 = (RPM_2 / RPM_1)^2 \times SP_1$$

HP varies with the cube of the RPM

$$HP_1 / HP_2 = (RPM_1 / RPM_2)^3$$

$$HP_2 = (RPM_2 / RPM_1)^3 \times HP_1$$

Example:

Fan operating at 1000 RPM, 3000 CFM, 0.5” wg, 0.5 BHP. Speed fan up 10% to 1100 RPM; what is the performance of the fan at the new speed?

$$RPM_1 = 1000; CFM_1 = 3000; SP_1 = 0.5; BHP_1 = 0.5; RPM_2 = 1100$$

Therefore:

$$CFM_2 = (1100/1000) \text{ x } 3000 = 3300 \text{ CFM (10\% increase in airflow)}$$

$$SP_2 = (1100/1000)^2 \text{ x } 0.5 = 0.605\text{” wg (21\% increase in static pressure)}$$

$$HP_2 = (1100/1000)^3 \text{ x } 0.5 = 0.6655 \text{ BHP (33\% increase in horsepower)}$$

So remember that a 10% increase in fan speed will result in a 33% increase in horsepower!

Fan Balancing

AMCA/ANSI Publication 204-05 has established recommended levels of balance and vibration for fan assemblies. Fans covered by this standard would include those for most commercial HVAC as well as most industrial process applications. The Standard does not address environments involving extreme forces or temperatures.

Balance and Vibration Categories for Fans

Application	Driver HP Limits	Fan Application Category
Residential	≤ 0.2	BV-1
Ceiling fan, attic fan & window AC	> 0.2	BV-2
HVAC & Agricultural	≤ 0.5	BV-2
Commercial systems, building vent & AC	> 0.5	BV-3
Industrial Process & Power Generation	≤ 400	BV-3
Baghouse, scrubber, mine, conveying, boiler, combustion air, pollution control & wind tunnels	> 400	BV-4
Transportation & Marine	> 20	BV-3
Locomotives, trucks & automotive	≤ 20	BV-4
Transit & Tunnels	≤ 100	BV-3
Subway emergency ventilation, tunnel fans & garage ventilation	> 100	BV-4
Tunnel jet fans	All	BV-4
Petrochemical Process	≤ 50	BV-3
Hazardous gases & process fans	> 50	BV-4
Computer Chip Manufacture Clean room	All	BV-5

The values associated with these BV categories are as follows:

Vibration Limits Associated with Balance and Vibration Levels

Fan Application Category	Rigid Mounted (in./s)	Flexible Mounted (in./s)
BV-1	0.5	0.6
BV-2	0.2	0.3
BV-3	0.15	0.2
BV-4	0.1	0.15
BV-5	0.08	0.1

Notes:
These values are for assembled fans balanced at the manufacturing location. Values shown are peak velocity values, filtered (measured at fan RPM).

Balance and vibration levels measured in the field will vary from those taken in the manufacturing plant. This is due primarily to differences in the mass and rigidity of the support system. There will also be situations where fans are shipped without motor or drives installed. In such cases, field or trim balancing will be necessary. For such cases, acceptable vibration levels are provided in the following table for various BV categories for three conditions: startup, alarm and shutdown. All values are unfiltered and are to be taken at the bearing housings.

Condition	Fan Application Category	Rigid Mounted (in./s)	Flexible Mounted (in./s)
Startup	BV-1	0.55	0.6
	BV-2	0.3	0.5
	BV-3	0.25	0.35
	BV-4	0.16	0.25
	BV-5	0.1	0.16
Alarm	BV-1	0.6	0.75
	BV-2	0.5	0.75
	BV-3	0.4	0.65
	BV-4	0.25	0.4
	BV-5	0.2	0.3
Shutdown	BV-1	*	*
	BV-2	*	*
	BV-3	0.5	0.7
	BV-4	0.4	0.6
	BV-5	0.3	0.4

* Startup values are for newly installed fans (values should be at or below these levels).

Notes:

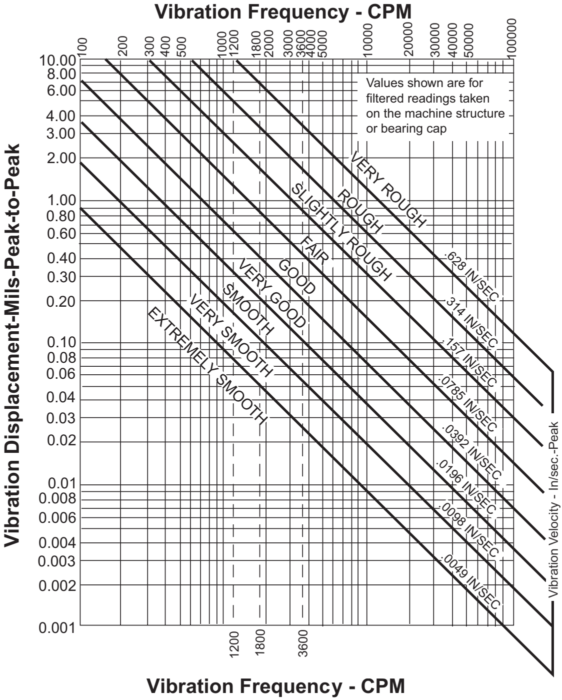
Over time, wear and tear will naturally increase vibration levels. Once they increase to "Alarm" values, corrective action should be taken. If "Shutdown" values are reached, fans should be shut down immediately to prevent severe damage or catastrophic failure, and action should be taken to determine the root cause of the vibration.

Vibration Severity

When evaluating vibration severity, the following factors must be taken into consideration:

- When using displacement measurements, only filtered displacement readings for a specific frequency should be applied to the chart. Unfiltered or overall velocity readings can be applied since the lines which divide the severity regions are, in fact, constant velocity lines.
- The chart applies only to measurements taken on the bearings or structure of the machine. The chart does not apply to measurements of shaft vibration.
- The chart applies primarily to machines which are rigidly mounted or bolted to a fairly rigid foundation. Machines mounted on resilient vibration isolators such as coil springs or rubber pads will generally have higher amplitudes of vibration than those rigidly mounted. A general rule is to allow twice as much vibration for a machine mounted on isolators. However, this rule should not be applied to high frequencies of vibration such as those characteristic of gears and defective rolling-element bearings, as the amplitudes measured at these frequencies are less dependent on the method of machine mounting.

Vibration Severity Chart



Vibration

System Natural Frequency

The natural frequency of a system is the frequency at which the system prefers to vibrate. It can be calculated by the following equation:

$$f_n = 188\ (1/d)^{1/2}\ (\text{cycles per minute})$$

The static deflection corresponding to this natural frequency can be calculated by the following equation:

$$d = (188/f_n)^2\ (\text{inches})$$

By adding vibration isolation, the transmission of vibration to the building can be minimized. A common rule of thumb for selection of vibration isolation is as follows

Equipment	Static Deflection of Vibration Isolation	
RPM	Critical Installation	Non-Critical Installation
1200+	1.0 in	0.5 in
600+	1.0 in	1.0 in
400+	2.0 in	1.0 in
300+	3.0 in	2.0 in

Notes:

- Critical installations are upper floor or roof mounted equipment. Non-critical installations are grade level or basement floor.
- Always use total weight of equipment when selecting isolation.
- Always consider weight distribution of equipment in selection.

Fan Troubleshooting Guide

Low Capacity or Pressure

- Incorrect direction of rotation. Make sure the fan impeller rotates in the proper direction.
- Poor fan inlet conditions. There should be a straight, clear duct at the fan inlet.
- Wheel to inlet overlap incorrect

Excessive Vibration

- Excessive belt tension
- Belts too loose, damaged, worn, or oily
- Damaged or unbalanced wheel
- Speed too high
- Incorrect direction of rotation. Make sure the fan impeller rotates in the proper direction.
- Bearings need lubrication or replacement
- Fan surge
- Loose mechanical components
- Poor inlet duct connections

Overheated Motor

- Motor improperly wired
- Incorrect direction of rotation. Make sure the fan impeller rotates in the proper direction.
- Cooling air diverted or blocked
- Incorrect fan RPM
- Incorrect voltage
- Overheated bearings
- Improper bearing lubrication
- Excessive belt tension.

System Design

General Ventilation Guidelines

- Locate intake and exhaust fans to make use of prevailing winds.
- Locate fans and intake ventilators for maximum sweeping effect over the working area.
- Size intake ventilator to keep intake losses below 1/8" in. wg.
- Avoid fans blowing opposite each other. When unavoidable, separate by at least 6 fan diameters.
- If air moving over motors contains hazardous chemicals or particles, use explosion-proof motors mounted in or out of the airstream, depending on job requirements.
- For hazardous atmosphere applications, use fans of non-sparking construction. See Spark Resistant Construction, AMCA Standard 99.

Process Ventilation Guidelines

- Collect fumes and heat as near the point of generation as possible.
- Make all duct runs as short and direct as possible.
- Keep duct velocity as low as practical considering capture for fumes or particles being collected.
- When turns are required in the duct system use long radius elbows to keep the resistance to a minimum (preferably 2 duct diameters or greater).
- After calculating duct resistance, select the fan having reserve capacity beyond the static pressure determined.
- Use same rationale regarding intake ventilators and motors as in General Ventilation Guidelines above.
- Install the exhaust fan at a location to eliminate any recirculation into other parts of the plant.
- When hoods are used, they should be sufficient to collect all contaminating fumes or particles created by the process.

Kitchen Ventilation Guidelines

Hoods and Ducts

- Duct velocity should be between 1500 and 4000 FPM.
- Hood velocities (not less than 50 FPM over face area between hood and cooking surface).
  - Wall Type: 80 CFM/ft²
  - Island Type: 125 CFM/ft²
  - Extend hood beyond cooking surface 0.4 × distance between hood and cooking surface.

Filters

- Select filter velocity between 100 - 400 FPM
- Determine number of filters required from manufacturer's data (usually 2 CFM exhaust for each sq. in. of filter area maximum)
- Install at 45 - 60° to horizontal, never horizontal
- Shield filters from direct radiant heat
- Filter mounting height:
  - No exposed cooking flame, 1-1/2' minimum to filter
  - Charcoal and similar fires, 4' minimum to filter
- Provide removable grease drip pan
- Establish a schedule for cleaning drip pan and filters and follow it diligently

Fans

- Use upblast discharge fan
- Select design CFM based on hood design and duct velocity
- Select SP based on design CFM and resistance of filters and duct system
- Adjust fan specification for expected exhaust air temperature

Ventilation Rates for Acceptable Indoor Air Quality

Space	Outdoor Air Required (CFM/person)	Occupancy (People/1000 ft²)
Auditoriums	5	150
Ballrooms/Discos	20	100
Bars	7.5	100
Beauty Shops	20	25
Classrooms	10	25
Conference Rooms	5	50
Correctional Facility Cells	5	25
Dormitory Sleeping Rooms	5	20
Gambling Casinos	7.5	120
Game Rooms	7.5	70
Libraries	5	20
Office Spaces	5	5
Pharmacies	5	10
Photo Studios	5	10
Restaurant Dining Areas	7.5	70
Retail Facilities	7.5	15
Sporting Spectator Areas	7.5	150
Supermarkets	7.5	8
Theaters	10	150

Adapted from table 6.1 ASHRAE Standard 62.1-2012 Ventilation for Acceptable Indoor Air Quality

Heat Removal Method

When the temperature of a space is higher than the ambient outdoor temperature, general ventilation may be utilized to provide "free cooling". Using the temperature differential (the desired indoor temperature minus the design outdoor dry bulb temperature) and the amount of heat removal required (BTU/Hr).

$$CFM = Heat\ Removal\ (BTU/Hr) / (1.10 \times \Delta T)$$

Heat Gain From Occupants of Conditioned Spaces

Typical Application	Heat Gain (BTU/Hr)		
	Total	Sensible	Latent
Theater (seated)	350	245	105
Offices, Hotels, Apartments	400	245	155
Retail and Department Stores	450	250	200
Banks & Drug Stores	500	250	250
Restaurant	550	275	275
Factory - light (bench work)	750	275	475
Factory - heavy work	1450	580	870
Gym (participants)	1800	710	1090
Dance Hall	850	305	545
Bowling Alley (participants)*	1450	580	870

Notes:
Tabulated values are based on 75°F room dry-bulb temperature.
Total heat gain is based on normal percentage of men, women and children for the application listed.
*Figure one person actually bowling per alley and all others as sitting (400 BTU/Hr) or standing (550 BTU/Hr).

Adapted from Chapter 18.4 ASHRAE "Fundamentals" Handbook, 2013

Rate of Heat Gain from Appliances and Equipment

Commercial Hooded Cooking Appliances in Air-Conditioned Area

Item	Sensible BTU/Hr	
	Electric	Gas
Broiler, underfired 3 ft	10,800	9,000
Fryer: open deep fat, 1 vat	1,000	1,100
Fryer: pressure	500	800
Griddle double sided 3 ft (clamshell down)*	1,400	1,800
Griddle double sided 3 ft (clamshell up)*	3,600	4,900
Griddle flat 3 ft	4,500	3,700
Induction cooktop*	2,700	-
Induction wok*	0	-
Range wok*	5,200	5,200
Range top: top off , oven on*	1,000	2,000
Range top: 3 elements on, oven off	6,300	7,100
Range: 6 elements on, oven off	13,900	11,500
Range: 6 elements on, oven on	14,500	13,600
Oven: combination: combination mode*	800	400
Oven: combination: convection mode	1,400	1,000
Oven: convection full size	1,500	1,000
Oven: conveyor (pizza)	-	7,800
Rotisserie*	4,500	-
Pasta cooker*	0	0
Rice cooker*	-	300
Salamander*	7,000	5,300
Steam kettle large (60 gal) simmer lid down*	100	0
Steam kettle small (40 gal) simmer lid down*	300	-
Steam kettle small (10 gal) simmer lid down *	-	300
Tilting skillet/braising pan	0	400

* Swiercyna, R, P.A. Sobiski, and D.R. Fisher. 2009. Revised heat gain rates from typical commercial cooking appliances from RP-1362. ASHRAE Transactions 115(2).

Adapted from ASHRAE “Fundamentals” Handbook, 2013

Commercial Unhooded Electric Cooking Appliances in Air-Conditioned Area

Item	Sensible BTU/Hr	
	Electric	Convective
Coffee brewing urn	200	300
Espresso machine	400	800
Hotdog roller	900	1,500
Hotplate: single burner, high speed	900	2,100
Microwave oven: commercial (heavy duty)	0	0
Oven: countertop conveyorized bake/ finishing	2,200	10,400
Panini	1,200	2,000
Popcorn popper	100	100
Refrigerator, reach-in	300	900
Refrigerated prep table	600	300
Steamer (bun)	600	100
Toaster: 4 slice, pop up (large) cooking	200	1,400
Toaster: contact (vertical)	2,700	2,600
Toaster: conveyor (large)	3,000	7,300
Toaster: conveyor (small)	800	400
Waffle iron	0	400
Dishwasher (conveyor type, chemical sanitizing)	0	4,450
Dishwasher (conveyor type, hot-water sanitizing) standby	0	4,750
Dishwasher (door type, chemical sanitizing) washing	0	1,980
Dishwasher (door type, hot-water sanitizing) washing	0	1,980
Dishwasher (under-counter type, chemical sanitizing) standby	0	2,280
Dishwasher (under-counter type, hot-water sanitizing) standby	800	1,040
Booster heater	500	0

Adapted from ASHRAE “Fundamentals” Handbook, 2013

Miscellaneous Appliances

Type	Item	Manufacturer's Rating		Recommended Rate of Heat Gain, BTU/Hr		
		Watts	BTU/ Hr	*Sensible	Latent	Total
Electrical	Hair dryer	1,580	5,400	2,300	400	2,700
	Hair dryer	705	2,400	1,870	330	2,200
	Neon sign,	-	-	30	-	30
	per linear ft of tube	-	-	60	-	60
	Sterilizer, instrument	1,100	3,750	650	1,200	1,850
Gas-Burning	Lab burners Bunsen	-	3,000	1,680	420	2,100
	Fishtail	-	5,000	2,800	700	3,500
	Meeker	-	6,000	3,360	840	4,200
	Gas Light, per burner	-	2,000	1,800	200	2,000
	Cigar lighter	-	2,500	900	100	1,000

* Use sensible heat gain for ventilation calculation

Adapted from ASHRAE "Fundamentals" Handbook, 2013

Office Equipment

Type of Use	Description			Load Factor W/ft²
	Density	ft² / Workstation	Accessories	
100% laptop	Light	167	1 printer per 10, speakers, misc.	0.25
	Medium	125		0.33
50% laptop 50% desktop	Light	167		0.40
	Medium	125		0.50
100% desktop	Light	167		0.60
	Medium	125		0.80
			2 monitors, 1 printer per 10, speakers, misc.	1.00
	Heavy	85	2 monitors, 1 printer per 8, speakers, misc.	1.50
			2 monitors, 1 printer per 8, speakers, misc. (no diversity)	2.00

Adapted from ASHRAE "Fundamentals" Handbook, 2013

Heat Gain From Electric Motors

			Location Relative to Airstream		
			Motor In Equipment In	Motor Out Equipment In	Motor Out Equipment Out
HP	Phase	Full Load Efficiency %	BTU/Hr	BTU/Hr	BTU/Hr
0.25	1	68.5	929	636	293
0.33	1	72.4	1,160	840	320
0.5	1	76.2	1,670	1,273	397
0.75	1	81.8	2,333	1,909	425
1	3	85.5	2,977	2,545	432
1.5	3	86.5	4,413	3,818	596
2	3	86.5	5,884	5,090	794
3	3	89.5	8,531	7,635	896
5	3	89.5	14,218	12,725	1,493
7.5	3	91.0	20,975	19,088	1,888
10	3	91.7	27,754	25,450	2,304
15	3	93.0	41,048	38,175	2,873
20	3	93.0	54,731	50,900	3,831
25	3	93.6	67,975	63,625	4,350
30	3	94.1	81,137	76,350	4,787
40	3	94.1	108,183	101,800	6,383
50	3	94.5	134,656	127,250	7,406
60	3	95.0	160,737	152,700	8,037
75	3	95.0	200,921	190,875	10,046
100	3	95.4	266,771	254,500	12,271
125	3	95.4	333,464	318,125	15,339
150	3	95.8	398,486	381,750	16,736
200	3	95.8	531,315	509,000	22,315
250	3	95.8	664,144	636,250	27,894

Calculated using information from 2009 ASHRAE Fundamentals, Chapter 18 and DOE minimum motor efficiencies for nominal 1800 RPM, ODP, General Purpose Motors

Air Filtration

ANSI/ASHRAE 52.2 is now the accepted standard used for evaluating filter performance. ASHRAE Standard 52.1 has been discontinued.

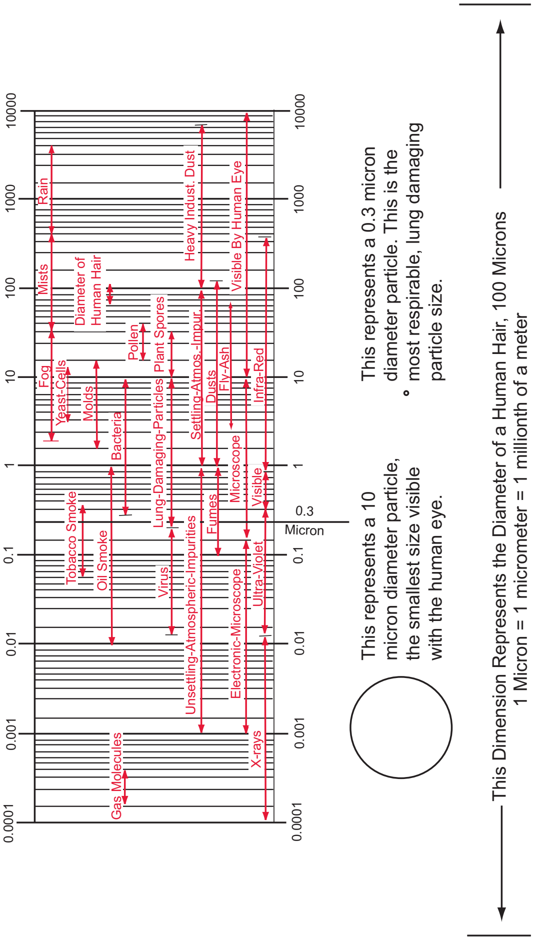
Filter performance is now reported as a Minimum Efficiency Reporting Value (MERV) at a specific test velocity. While this is a completely new test with no direct comparison to the Average Dust Spot Efficiency parameters of earlier Standards, two criteria of these standards have been adopted by 52.2; “Arrestance” for determining MERV 1-4 and “Dust Holding Capacity." Dust Holding Capacity is reported simply as the weight of synthetic loading dust captured by the filter for the duration of this test and should not be used to calculate the expected life of the device in use. (ASHRAE 52.1 information is shown for comparison purposes only)

52.2 tests are run intermittently on representative samples of filters from normal manufacturing runs, not on each individual filter elements (See HEPA filters, below).

The standard quantifies the ability of a filter to remove particulate, on average, in specific size ranges, from the airstream. This is referred to as Particulate Size Efficiency. Filters are rated for particle sizes ranging from 0.3 to 10 micrometers in three ranges; E1 (0.3 – 1 micrometer), E2 (1 – 3 micrometers) and E3 (3-10 micrometers).

See Filter Application Guidelines chart for more information.

Relative Size Chart of Common Air Contaminants



Filter Application Guidelines

Particle Size	Typical Controlled Contaminant	Typical Application	Typical Air Filter/ Cleaner Type
< 0.3 µm	Virus (unattached) Carbon dust Tobacco smoke All combustion	Electronics manufacturing Pharmaceutical manufacturing Carcinogenic materials	**HEPA / ULPA Filters**
0.3-1 µm	Bacteria Droplet nuclei (sneeze) Cooking oil Most smoke Insecticide dust Most face powder Most paint	Superior commercial buildings Hospital inpatient care General surgery	**Bag filters:** Non supported (flexible) microfine fiberglass or synthetic media, 12 to 36 inches deep. **Box filters:** Rigid style cartridge, 6 to 12 inches deep.
1-3 µm	Legionella Humidifier dust Lead dust Auto emission particles Nebulizer drops	Superior residential Better commercial buildings Hospital laboratories	**Pleated filters:** Extended surface with cotton or polyester media or both, 1 to 6 inches thick. **Box filters:** Rigid style cartridge, 6 to 12 inches deep.
3-10 µm	Mold spores Dust mite feces Cat and dog dander Hair spray Fabric protector Dusting aids	Better residential Commercial buildings Industrial workspaces	**Pleated filters:** Extended surface with cotton or polyester media or both, 1 to 6 inches thick **Cartridge filters:** Viscous cube or pocket filters **Throwaway:** Synthetic media panel filters
> 10 µm	Pollen Dust mites Cockroach feces Sanding dust Spray paint dust Textile fibers Carpet fibers	Minimum filtration Residential window air conditioners	**Throwaway:** Fiberglass or synthetic media panel, 1 inch thick **Washable:** Aluminum mesh, foam rubber panel **Electrostatic:** Self-charging (passive) woven polycarbonate panel

Minimum Efficiency Reporting Value (MERV) Parameters

ASHRAE Standard 52.2				ASHRAE Standard 52.1
MERV	Particle Size Removal Efficiency, %			Dust - Spot Efficiency
	0.3 to 1 μm	1 to 3 μm	3 to 10 μm	
16	> 95	> 95	> 95	-
15	85 - 95	> 90	> 90	> 95
14	75 - 85	> 90	> 90	90 - 95
13	< 75	> 90	> 90	80 - 90
12	-	> 80	> 90	70 - 75
11	-	65 - 80	> 85	60 - 65
10	-	50 - 65	> 85	50 - 55
9	-	< 50	> 85	40 - 45
8	-	-	> 70	30 - 35
7	-	-	50 - 70	25 - 30
6	-	-	35 - 50	< 20
5	-	-	20 - 35	< 20
1 - 4	-	-	< 20	< 20

Notes:
1. This table is adapted from ANSI/ASHRAE Standard 52.2-2007.15
2. HEPA filters (formerly delineated as MERV 17-20 filters) are not part of this Standard. HEPAs are each individually tested for penetration to absolute – not average – values by strictly controlled IEST test methods.
3. For residential applications, the ANSI/ASHRAE Standard 62.2-2007.16 requires a filter with a designated minimum efficiency of MERV 6 or better. Current LEED requirements specify conditions for MERV 8 and MERV 13 filters.

Sound

- **Sound Power (W)**: the amount of power a source converts to sound in watts.
- **Sound Power Level (L_W)**: a logarithmic comparison of sound power output by a source to a reference sound source, W_0 (10^{-12} watt).

$$L_W = 10 \log_{10} (W/W_0) \text{ dB}$$

- **Sound Pressure (P)**: pressure associated with sound output from a source. Sound pressure is what the human ear reacts to.
- **Sound Pressure Level (Lp)**: a logarithmic comparison of sound pressure output by a source to a reference sound source, P_0 (2×10^{-5} Pa).

$$Lp = 20 \log_{10} (P/P_0) \text{ dB}$$

- **Noise Criteria (NC):** a single numerical index commonly used to define design goals for the maximum allowable noise in a given space. NC levels consist of a family of curves that define the maximum allowable octave-band sound pressure level corresponding to a chosen NC design goal.

Even though sound power level and sound pressure level are both expressed in dB, there is no outright conversion between the two. A constant sound power output will result in significantly different sound pressures and sound pressure levels when the source is placed in different environments.

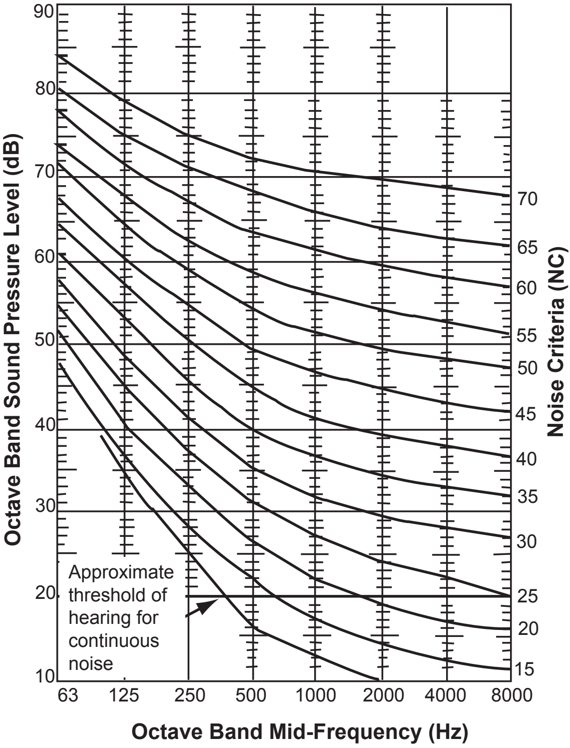
Rules of Thumb

- When specifying sound criteria for HVAC equipment, refer to sound power level, not sound pressure level.
- When comparing sound power levels, remember the lowest and highest octave bands are only accurate to about +/- 4 dB.
- Lower frequencies are the most difficult to attenuate.
- When trying to calculate the additive effect of two sound sources, use the approximation (logarithms cannot be added directly) on the following chart.

Noise Criteria (NC)

Graph sound pressure level for each octave band on NC curve. Highest curve intercepted is NC level of sound source.

NC Curves



Adding Sound Pressure Levels

Difference between sound pressure levels	dB to add to highest sound pressure level
0	3.0
1	2.5
2	2.1
3	1.8
4	1.5
5	1.2
6	1.0
7	0.8
8	0.6
9	0.5
10+	0

2 × sound pressure (single source) = +3 dB (sound pressure level)

2 × distance from sound source = -6 dB (sound pressure level)

+10 dB (sound pressure level)= 2 × original loudness perception

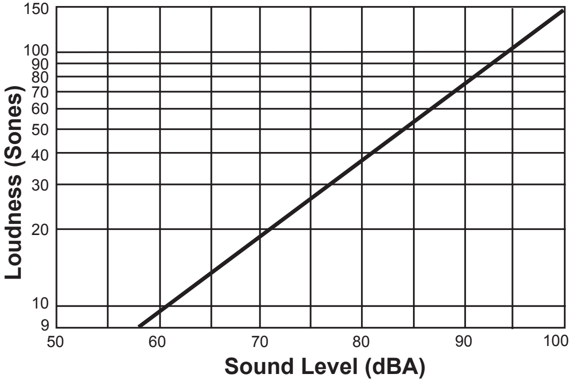
Sound Power and Sound Power Level

Sound Power (Watts)	Sound Power Level dB	Source
100,000	170	Jet engine with afterburner
10,000	160	Jet aircraft at takeoff
1,000	150	Turboprop at takeoff
100	140	Prop aircraft at takeoff
10	130	Loud rock band
1	120	Small aircraft engine
0.1	110	Blaring radio
0.01	100	Car at highway speed
0.001	90	Voice - shouting
0.0001	80	Garbage disposal unit
0.00001	70	Voice - conversational
0.000001	60	Electronic equipment cooling fan
0.0000001	50	Office air diffuser
0.00000001	40	Small electric clock
0.000000001	30	Voice - very soft whisper

Sound Pressure and Sound Pressure Level

Sound Pressure (Pascals)	Sound Pressure Level dB	Typical Environment
200.0	140	100 ft. from military aircraft at take-off
63.0	130	Pneumatic chipping and riveting (operator's position)
20.0	120	Passenger Jet takeoff at 100 ft.
6.3	110	Automatic punch press (operator's position)
2.0	100	Automatic lathe shop
0.63	90	Construction site - pneumatic drilling
0.2	80	Computer printout room
0.063	70	Loud radio (in average domestic room)
0.02	60	Restaurant
0.0063	50	Conversational speech at 3 ft.
0.002	40	Whispered conversation at 6 ft.
0.00063	30	Quiet bedroom at night
0.0002	20	Background in TV recording studios
0.00002	0	Normal threshold of hearing

Room Sones: dBA Correlation



From ASHRAE "Fundamentals" Handbook, 2013

Design Criteria for Room Loudness (Sones)

Values shown are room loudness in sones and are not fan sone ratings. For additional detail see AMCA publication 302-73 (R2008) Application of Sone Rating.

Auditoriums	
Concert and opera halls	1 to 3
Stage theaters	1.5 to 5
Movie theaters	2 to 6
Semi-outdoor amphitheaters	2 to 6
Lecture halls	2 to 6
Multi-purpose	1.5 to 5
Courtrooms	3 to 9
Auditorium lobbies	4 to 2
TV audience studios	2 to 6

Manufacturing Areas	
Heavy machinery	25 to 60
Foundries	20 to 60
Light machinery	12 to 36
Assembly lines	12 to 36
Machine shops	15 to 50
Plating shops	20 to 50
Punch press shops	50 to 60
Tool maintenance	7 to 21
Foreman's office	5 to 15
General storage	10 to 30

Churches and Schools	
Sanctuaries	1.7 to 5
Schools & classrooms	2.5 to 8
Recreation halls	4 to 12
Kitchens	6 to 18
Libraries	2 to 6
Laboratories	4 to 12
Corridors and halls	5 to 15

Indoor Sports Activities	
Gymnasiums	4 to 12
Coliseums	3 to 9
Swimming pools	7 to 21
Bowling alleys	4 to 12
Gambling casinos	4 to 12

Offices	
Executive	2 to 6
Supervisor	3 to 9
General open offices	4 to 12
Tabulation/ computation	6 to 18
Drafting	4 to 12
Professional offices	3 to 9
Conference rooms	1.7 to 5
Board of Directors	1 to 3
Halls and corridors	5 to 15

Continued on next page

Hospitals and Clinics	
Private rooms	1.7 to 5
Wards	2.5 to 8
Laboratories	4.0 to 12
Operating rooms	2.5 to 8
Lobbies & waiting rooms	4.0 to 12
Halls and corridors	4.0 to 12

Hotels	
Lobbies	4 to 12
Banquet rooms	8 to 24
Ball rooms	3 to 9
Individual rooms/ suites	2 to 6
Kitchens and laundries	7 to 21
Halls and corridors	4 to 12
Garages	6 to 18

Restaurants	
Restaurants	4 to 12
Cafeterias	6 to 8
Cocktail lounges	5 to 15
Social clubs	3 to 9
Night clubs	4 to 12
Banquet room	8 to 24

Miscellaneous	
Reception rooms	3 to 9
Washrooms and toilets	5 to 15
Studios for sound reproduction	1 to 3
Other studios	4 to 12

Transportation (rail, bus & plane)	
Waiting rooms	5 to 15
Ticket sales office	4 to 12
Control rooms & towers	6 to 12
Lounges	5 to 15
Retail shops	6 to 18

Retail Stores	
Clothing stores	4 to 12
Supermarkets	7 to 21
Department stores (main floor)	6 to 18
Department stores (upper floor)	4 to 12
Small retail stores	6 to 18

Residences	
Two & three family units	3 to 9
Apartment houses	3 to 9
Private homes (urban)	3 to 9
Private homes (rural & suburban)	1.3 to 4

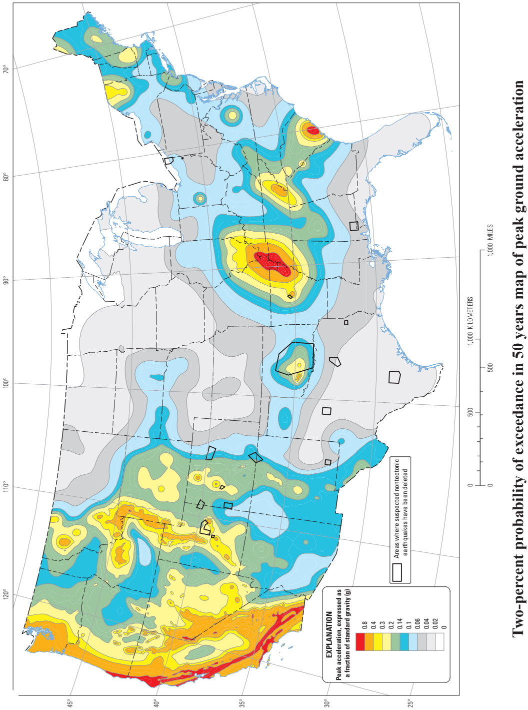
Public Buildings	
Museums	3 to 9
Planetariums	2 to 6
Post offices	4 to 12
Courthouses	4 to 12
Public libraries	2 to 6
Banks (general)	4 to 12
Lobbies and corridors	4 to 12

Seismic & Wind Forces

Seismic Forces

Designers are responsible for ensuring that equipment they specify does not become a hazard under the effects of seismic activity. Specific requirements are detailed in local codes and may include such concerns as proper fastening and restraint, mounting and isolation. The accompanying map indicates areas which have the highest probability of seismic activity in the continental United States.

Seismic Zones

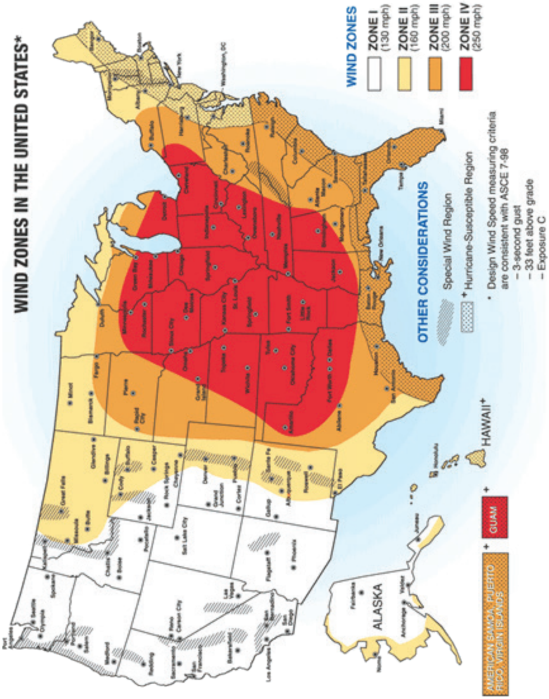


Seismic Action Zones defined by USGS 2008

Wind Forces

High winds can apply significant forces to equipment mounted on building exteriors. In some cases, where buildings or systems provide critical services, the ventilation equipment may be required to operate continuously under such conditions. Local codes detail requirements which may include equipment duty/suitability, tie downs, guy wires or anchoring. The accompanying Wind Zone map indicates maximum wind speeds for the United States.

Wind Zones



Duct Design

Maximum Duct Velocities (FPM)

Application	Controlling Factor				
	Noise Generation	Duct Friction			
	Main Ducts	Main Ducts		Branch Ducts	
		Supply	Return	Supply	Return
Residences, private	600	1000	800	600	600
Apartments, hotel rooms, hospital rooms	1000	1200	1300	1200	1000
Private offices, libraries	1200	2000	1500	1600	1200
Theaters, auditoriums	800	1300	1100	1000	800
General offices, upscale restaurants, upscale stores, banks	1500	2000	1500	1600	1200
General stores, cafeterias	1800	2000	1500	1600	1200
Industrial	2500	3000	1800	2200	1500

Carrier Air System Design Manual

Duct Loss Estimating

Engineers have standards by which they design HVAC systems. Some standards ensure code compliance and some are for the sake of design efficiency. For ducted HVAC systems, these standards include such things as duct velocities, maximum flex duct length, type of supply and return devices, duct materials, etc.

This section is intended to help estimate duct system pressure loss when preliminary equipment selection is being made. It can be useful for cost estimating, preliminary proposals and other situations where a complete system design is not practical. It is NOT intended to replace comprehensive system design or analysis by qualified professionals.

Method

- Layout single line duct diagram. (No need to size duct, but duct velocity and the **Design Flow Loss** (DFL - pressure loss/100' of duct) are necessary.
- Select most likely critical path (duct path with greatest loss).
- Measure **Total Duct Length** (TDL) including supply and return paths along critical path.
- Tabulate the type and quantity of duct fittings along critical path.
- Utilize the Fitting Loss charts in this section to determine the **Total Equivalent Length** (TEL) of duct for all fittings. Tables are calculated for 1350 fpm and 0.1" w.g./100 feet of duct.
- Use Fitting Loss Conversion Factors Chart to convert equivalent length for other system design conditions other than 1350 fpm and 0.1" w.g./100 feet of duct.
- Calculate estimated duct system losses as follows:

$$\frac{(TDL + TEL)}{100} \times DFL$$

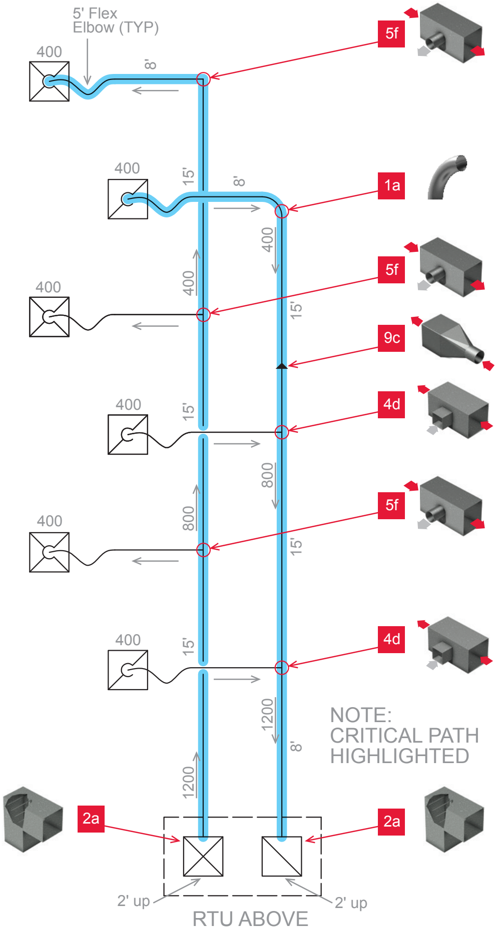
- Add losses from other components

Examples of common external static pressure losses (Substitute your standard design values):

- Flex duct (5' smooth radius elbow to inlet/outlet) - 120' equivalent duct length
- Throwaway Filter - 0.1"
- Supply Outlet - 0.1"
- Return Inlet - 0.1"

Duct Design

Sample System Layout



Duct Loss Estimation Example

3 Ton Packaged Rooftop Unit (RTU) - 1200 CFM

Duct velocity - 1000 FPM @ 0.1"/100'

Calculate Total Duct Length (TDLsupply and TDLreturn) along Critical Path

TDLsupply = 2' + 15' + 15' + 15' + 8' = 55'

TDLreturn = 2' + 8' + 15' + 15' + 8' = 48'

TDLtotal = 55' + 48' = 103'

Fitting Losses

Supply Path Fittings

Type 2a (H/W = 1) - 45'

Type 5f (where Qb/Qt = 0.3) - 19'

Type 5f (where Qb/Qt = 0.5) - 10'

Type 5f (where Qb/Qt = 1.0) - 0'

Flex Drop - 120'

TELsupply = 45' + 19' + 10' + 0' + 120' = 194'

Return Path Fittings

Flex Elbow - 120'

Type 1a (where R/D=1) - 25'

Type 9c (where A1/A2 = 2 and transition angle = 90) - 36'

Type 4d (where Qb/Qt) = 0.5) - 60'

Type 4d (whereQb/Qt) = 0.3) - 43'

Type 2a - 45'

TELreturn = 120' + 25' + 36' + 60' + 43' + 45' = 329'

TELtotal = 194' + 329' = 523'

As the duct design velocity is 1000 fpm in lieu of 1350 fpm, using 'Fitting Loss Conversion Factors' table on page 66, correct for design criteria. For 1000 fpm @ 0.1", read 0.55.

Therefore, 523' x 0.55 = 288' (TEL corrected for system design criteria)

TDL + TEL = 103' + 288' = 391'

Therefore: $\frac{(TDL + TEL)}{100}$ x DFL = 391/100 x 0.1 = 0.391" or ~ 0.4" wg.

Add in estimated losses of supply diffuser, return diffuser and filter:

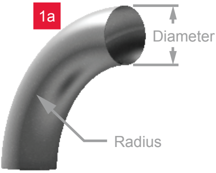
0.4 + 0.1 + 0.1 + 0.1 = 0.7" wg.

So the estimated external system pressure for the RTU is 0.7" wg.

Your Standard Design Values

Fitting Loss Charts - Equivalent Length Of Straight Duct

Round Elbows

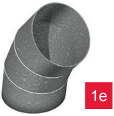


90º Smooth

90º 5-Piece

90º 3-Piece

90° Mitered



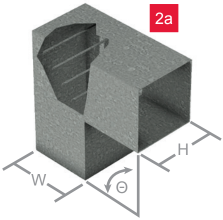
45° 3-Piece

45° 2-Piece

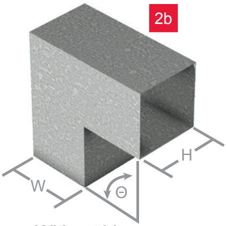
Radius / Diameter	90º				45°	
	1a	1b	1c	1d	1e	1f
0.75	37'	52'	61'		15'	20'
1.0	25'	37'	48'	136'	15'	20'
1.5+	17'	27'	39'		15'	20'

Duct Design

Rectangular Mitered Elbows



With Vanes



Without Vanes

2a With Vanes

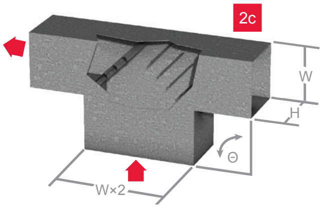
H/W=0.25	H/W=1	H/W=4
53'	45'	72'

Angles Other Than 90°	
Angle	Factor
30°	0.45
45°	0.6
60°	0.78

2b Without Vanes

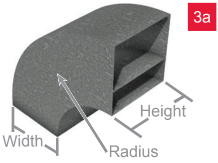
H/W=0.25	H/W=1	H/W=4
148'	136'	105'

2c Tee (Double Elbow Equivalent)



Flow can be diverging or converging, with or without vanes. Use table above corresponding to similar elbows.

Rectangular Radius Elbows



With Vanes

Without Vanes

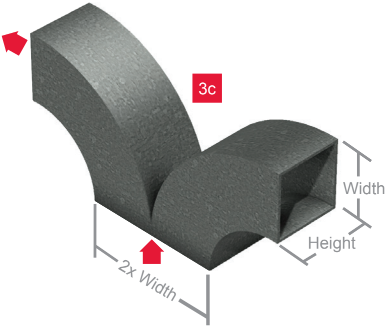
3a With Vanes

Radius / Width	H/W=0.25	H/W=1	H/W=4
0.25	9'	5'	6'
0.5	3'	2'	1'

3b Without Vanes

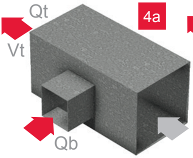
Radius / Width	H/W=0.25	H/W=1	H/W=4
0.5	170'	136'	125'
1.0	31'	24'	22'
2.0	23'	17'	16'

3c Wye (Double Elbow Equivalent)

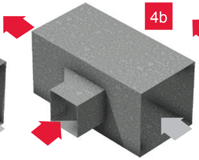


Flow can be diverging or converging, with or without vanes. Use table above corresponding to similar elbows.

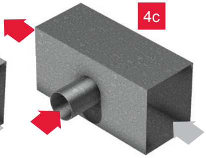
Converging Tee Rectangular Trunk



Rectangular Branch



Rectangular Branch With 45° Entry



Round Branch

Use the values below when evaluating the branch path. Refer to red airflow arrows for clarification.

4a Rectangular Branch

Velocity trunk (Vt)	Volume Rate of Flow (Branch) / Volume Rate of Flow (Trunk) (Qb/Qt)								
	0.1	0.2	0.3	0.4	0.5	0.6	0.7	0.8	0.9
<1200 FPM	0'	3'	10'	37'	117'	125'	244'	333'	475'
>1200 FPM	1'	8'	26'	76'	133'	189'	303'	382'	447'

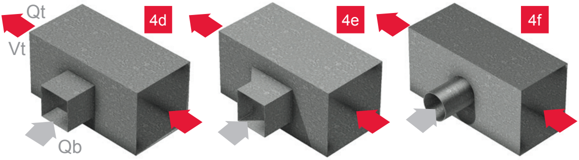
4b Rectangular Branch With 45° Entry

Velocity Trunk (Vt)	Volume Rate of Flow (Branch) / Volume Rate of Flow (Trunk) (Qb/Qt)								
	0.1	0.2	0.3	0.4	0.5	0.6	0.7	0.8	0.9
<1200 FPM	0'	8'	16'	32'	62'	117'	170'	219'	284'
>1200 FPM	2'	10'	20'	39'	86'	130'	208'	228'	330'

4c Round Branch

Velocity trunk (Vt)	Volume Rate of Flow (Branch) / Volume Rate of Flow (Trunk) (Qb/Qt)								
	0.1	0.2	0.3	0.4	0.5	0.6	0.7	0.8	0.9
<1200 FPM	0'	7'	15'	26'	89'	148'	219'	352'	554'
>1200 FPM	2'	12'	26'	68'	144'	234'	312'	420'	560'

Converging Tee Rectangular Trunk



Rectangular Branch | Rectangular Branch With 45° Entry | Round Branch

Use the values below when evaluating the trunk path.
Refer to red airflow arrows for clarification.

4d Rectangular Branch

4e Rectangular Branch With 45° Entry

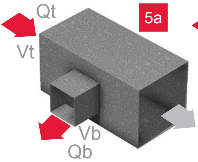
4f Round Branch

Qb/Qt	Volume Rate of Flow (Branch) / Volume Rate of Flow (Trunk) (Qb/Qt)								
	0.1	0.2	0.3	0.4	0.5	0.6	0.7	0.8	0.9
EL	18'	31'	43'	52'	60'	65'	67'	68'	67'

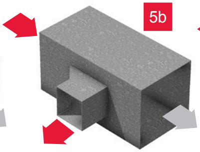
Note: EL = Equivalent Length of duct in feet.

Duct Design

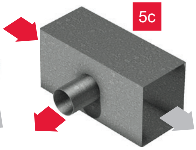
Diverging Tee Rectangular Trunk



Rectangular Branch



Rectangular Branch
With 45° Entry



Round Branch

Use the values below when evaluating the branch path.
Refer to red airflow arrows for clarification.

5a Rectangular Branch

Vb/Vt	Qb/Qt				
	0.1	0.2	0.3	0.4	0.5
0.2	117'				
0.4	118'	115'			
0.6	126'	117'	119'		
0.8	132'	137'	133'	127'	
1.0	157'	159'	148'	155'	144'

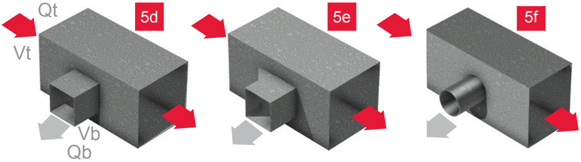
5b Rectangular Branch With 45° Entry

Vb/Vt	Qb/Qt				
	0.1	0.2	0.3	0.4	0.5
0.2	103'				
0.4	92'	90'			
0.6	87'	82'	80'		
0.8	89'	83'	78'	75'	
1.0	89'	111'	97'	90'	84'

5c Round Branch

Vb/Vt	Qb/Qt				
	0.1	0.2	0.3	0.4	0.5
0.2	114'				
0.4	115'	122'			
0.6	130'	125'	123'		
0.8	134'	149'	127'	128'	
1.0	148'	157'	136'	140'	143'

Diverging Tee Rectangular Trunk



Rectangular Branch | Rectangular Branch With 45° Entry | Round Branch

Use the values below when evaluating the branch path. Refer to red airflow arrows for clarification.

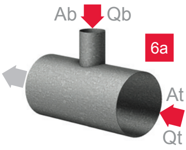
5d Rectangular Branch
5e Rectangular Branch With 45° Entry
5f Round Branch

Vb/Vt	Equivalent Length (Trunk Path)							
	0.1	0.2	0.3	0.4	0.5	0.6	0.8	1
EL	32'	25'	19'	15'	10'	7'	2'	0'

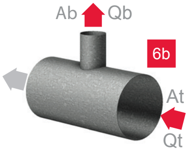
Note: EL = Equivalent Length of duct in feet.

Duct Design

Tee, Round, 90 Degree



Converging



Diverging

Use the values below when evaluating the branch path.
Refer to red airflow arrows for clarification.

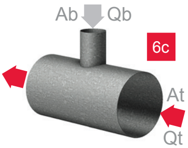
6a Converging

Qb/Qt	Area of Branch / Area of Trunk (Ab/At)						
	0.1	0.2	0.3	0.4	0.6	0.8	1
0.2	432'	82'	19'	17'	15'	13'	12'
0.4		489'	239'	107'	61'	45'	36'
0.6			534'	182'	105'	78'	65'
0.8				307'	170'	125'	98'
1				454'	239'	159'	125'

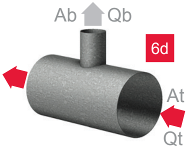
6b Diverging

Ab/At	Volume Rate of Flow for Branch (Qb) / Volume Rate of Flow for Trunk (Qt)								
	0.1	0.2	0.3	0.4	0.5	0.6	0.7	0.8	0.9
0.8	89'	70'	56'	45'	39'	35'	36'	40'	45'
0.6	84'	64'	50'	42'	40'	41'	49'	61'	77'
0.4	75'	53'	45'	49'	61'	78'	108'	148'	193'
0.2	64'	64'	114'	205'					

Tee, Round, 90 Degree



Converging



Diverging

Use the values below when evaluating the trunk path.
Refer to red airflow arrows for clarification.

6c Converging

Qb/Qt	Volume Rate of Flow (Branch) / Volume Rate of Flow (Trunk) (Qb/Qt)							
	0.1	0.2	0.3	0.4	0.5	0.6	0.8	0.9
EL	18'	31'	43'	52'	60'	65'	68'	67'

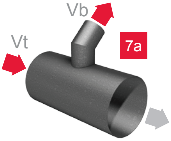
Note: EL = Equivalent Length of duct in feet.

6d Diverging

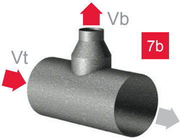
Vb/Vt	Volume Rate of Flow (Branch) / Volume Rate of Flow (Trunk) (Qb/Qt)							
	0.1	0.2	0.3	0.4	0.5	0.6	0.8	1
EL	32'	25'	19'	15'	10'	7'	2'	0'

Note: EL = Equivalent Length of duct in feet.

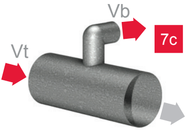
Diverging Tee Round Trunk



With 45° Round Branch Elbow



With Conical Round Branch



With 90° Round Branch Elbow

Use the values below when evaluating the branch path. Refer to red airflow arrows for clarification.

7a With 45° Round Branch Elbow

Vb/Vt	0.2	0.4	0.6	0.8	1
EL	108'	102'	98'	92'	90'

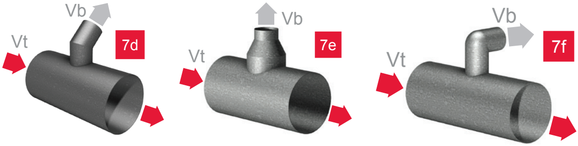
7b With Conical Round Branch

Vb/Vt	0.2	0.4	0.6	0.8	1
EL	97'	84'	70'	59'	48'

7c With 90° Round Branch Elbow

Vb/Vt	0.2	0.4	0.6	0.8	1
EL	117'	123'	134'	151'	177'

Diverging Tee Round Trunk



Rolled 45° Round Branch Elbow

Conical Round Branch

90° Round Branch Elbow

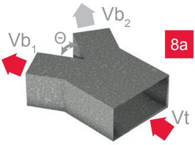
Use the values below when evaluating the trunk path.
Refer to red airflow arrows for clarification.

7d **Rolled 45° Round Branch Elbow**
7e **Conical Round Branch**
7f **90° Round Branch Elbow**

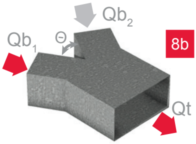
Vb/Vt	0.1	0.2	0.3	0.4	0.5	0.6	0.8	1.0
EL	32'	25'	19'	15'	10'	7'	2'	0'

Note: EL = Equivalent Length of duct in feet.

Wye Rectangular Or Round (Pair Of Pants)



Diverging



Converging

The values below are used to evaluate either branch.

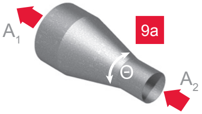
8a Diverging

Vb1 / Vt or Vb2 / Vt							
Angle	0.1	0.2	0.3	0.4	0.5	0.6	0.8
15°	92'	70'	58'	43'	32'	23'	12'
30°	95'	78'	64'	50'	39'	32'	22'
45°	99'	84'	72'	61'	51'	43'	33'
60°	102'	93'	90'	75'	67'	60'	49'

8b Converging

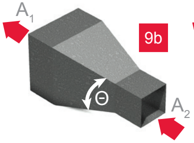
Volume Rate of Flow Qb1 / Qt or Qb2 / Qt							
Angle	0.1	0.2	0.3	0.4	0.5	0.6	0.8
15°	0'	1'	2'	4'	11'	47'	97'
30°	2'	4'	8'	16'	32'	78'	125'
45°	4'	8'	10'	23'	64'	105'	182'

Transition Expanding Flow

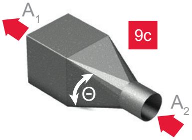


9a Round, Conical

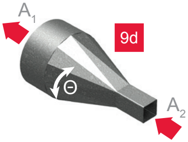
A_1 / A_2	Transition Angle							
	16°	20°	30°	45°	60°	90°	120°	180° (Abrupt)
2	16'	22'	36'	37'	37'	36'	35'	34'
4	26'	34'	52'	69'	77'	73'	72'	70'
6	31'	37'	55'	75'	87'	84'	83'	82'
10	33'	43'	67'	86'	91'	94'	95'	94'
16+	35'	43'	68'	95'	100'	100'	100'	100'



Rectangular to Rectangular



Round to Rectangular



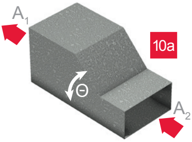
Rectangular to Round

9b Rectangular to Rectangular
9c Round to Rectangular
9d Rectangular to Round

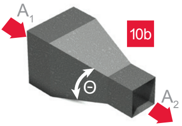
A_1 / A_2	Transition Angle							
	16°	20°	30°	45°	60°	90°	120°	180° (Abrupt)
2	20'	25'	28'	33'	35'	36'	37'	34'
4	41'	49'	57'	64'	69'	72'	72'	72'
6	48'	53'	66'	77'	82'	86'	86'	85'
10	48'	56'	67'	80'	91'	99'	97'	98'

Duct Design

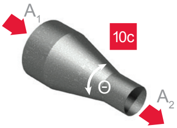
Transition Contracting Flow



Expanding Flow,
Rectangular,
Straight Sides



Contracting Flow,
Rectangular



Contracting Flow,
Round

10a Expanding Flow, Rectangular, Straight Sides

A_1 / A_2	Transition Angle						
	14°	20°	30°	45°	60°	90°	180° (Abrupt)
2	10'	14'	23'	39'	42'	43'	40'
4	18'	28'	48'	68'	77'	80'	75'
6	22'	34'	55'	74'	86'	94'	91'

10b 10c Contracting Flow, Rectangular and Round

A_1 / A_2	Transition Angle						
	10°	15-40°	50-60°	90°	120°	150°	180° (Abrupt)
2	6'	6'	7'	14'	20'	27'	30'
4	6'	5'	8'	19'	31'	40'	47'
6	6'	5'	8'	20'	32'	41'	48'
10	6'	6'	9'	22'	33'	42'	49'

Fitting Loss Conversion Factors

Design Duct Velocity (FPM)	Design Rate of Friction Loss ("/100' of duct)	
	0.08	0.1
900	0.56	0.44
1000	0.69	0.55
1100	0.83	0.66
1200	0.99	0.79
1300	1.16	0.93
1400	1.34	1.08
1500	1.54	1.23
1600	1.76	1.40
1700	1.98	1.59
1800	2.22	1.78
1900	2.48	1.98
2000	2.74	2.19

Look up the appropriate factor for the system design conditions and multiply by the Total Equivalent Length of fittings. See Duct Loss Estimation Example for clarification.

Typical Design Velocities for HVAC Components

Component	Velocity (FPM)
Intake Louvers (7000 CFM and UP)	400
Exhaust Louvers (5000 CFM and UP)	500
Panel Filters	
Viscous Impingement	200 to 800
Dry-Type, Pleated Media	
Low Efficiency	350
Medium Efficiency	500
High Efficiency	500
HEPA	250
Renewable Media Filters	
Moving-Curtain Viscous Impingement	500
Moving-Curtain Dry-Media	200
Electronic Air Cleaners	
Ionizing-Plate-Type	300 to 500
Charged-Media Non-ionizing	250
Charged-Media Ionizing	150 to 350
Steam and Hot Water Coils	500 to 1000 200 min 1500 max
Electric Coils	
Open Wire	Refer to Mfg. Data
Finned Tubular	
Dehumidifying Coils	400 to 600
Air Washers	
Spray-Type Air Washers	Refer to Mfg. Data
Cell-Type Air Washers	
High-Velocity, Spray-Type Air Washers	1200 to 1800

Adapted from ASHRAE “Fundamentals" Handbook 2013

Velocity & Pressure Relationships

Velocity (FPM)	Velocity Pressure (in. w.g.)	Velocity (FPM)	Velocity Pressure (in. w.g.)
300	0.0056	3500	0.7637
400	0.0097	3600	0.8079
500	0.0155	3700	0.8534
600	0.0224	3800	0.9002
700	0.0305	3900	0.9482
800	0.0399	4000	0.9975
900	0.0504	4100	1.0480
1000	0.0623	4200	1.0997
1100	0.0754	4300	1.1527
1200	0.0897	4400	1.2069
1300	0.1053	4500	1.2624
1400	0.1221	4600	1.3191
1500	0.1402	4700	1.3771
1600	0.1596	4800	1.4364
1700	0.1801	4900	1.4968
1800	0.2019	5000	1.5586
1900	0.2250	5100	1.6215
2000	0.2493	5200	1.6857
2100	0.2749	5300	1.7512
2200	0.3017	5400	1.8179
2300	0.3297	5500	1.8859
2400	0.3591	5600	1.9551
2500	0.3896	5700	2.0256
2600	0.4214	5800	2.0972
2700	0.4544	5900	2.1701

For calculation of velocity pressures at velocities other than those listed above: $P_v = (V/4005)^2$

For calculation of velocities when velocity pressures are known:

$$V = 1096\sqrt{P_V/\rho}$$

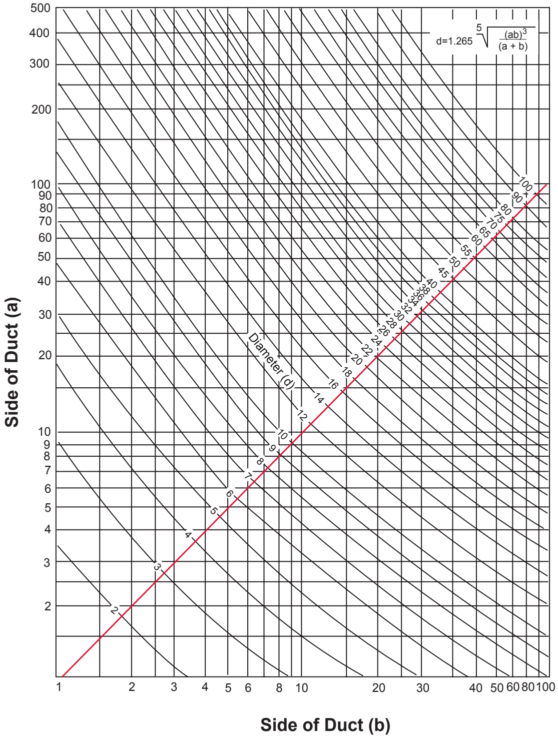
Where: V = *velocity (fpm)*

P_V = *velocity pressure (in. w.g.)*

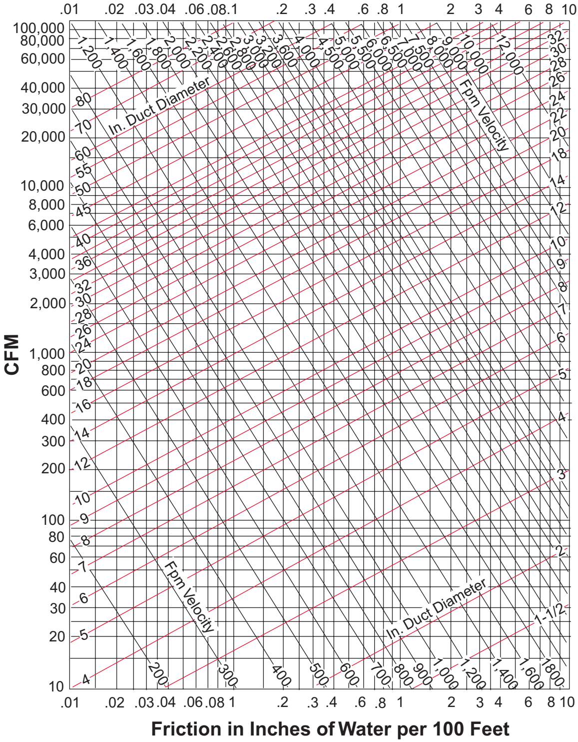
ρ = *air density (lbs/ft²)*

at standard air density: $V=4005\sqrt{P_V}$

Rectangular Equivalent of Round Ducts



Duct Resistance



Friction of Air in Straight Duct

U.S. Sheet Metal Gauges

Gauge No.	Steel		Galvanized		Stainless Steel		Aluminum	
	Thick (in)	Lb/ft²	Thick (in)	Lb/ft²	Thick (in)	Lb/ft²	Thick (in)	Lb/ft²
26	.0179	.750	.0217	.906	.0188	.7875	.020	.282
24	.0239	1.00	.0276	1.156	.0250	1.050	.025	.353
22	.0299	1.25	.0336	1.406	.0312	1.313	.032	.452
20	.0359	1.50	.0396	1.656	.0375	1.575	.040	.564
18	.0478	2.00	.0516	2.156	.050	2.100	.050	.706
16	.0598	2.50	.0635	2.656	.062	2.625	.064	.889
14	.0747	3.125	.0785	3.281	.078	3.281	.080	1.13
12	.1046	4.375	.1084	4.531	.109	4.594	.100	1.41
10	.1345	5.625	.1382	5.781	.141	5.906	.125	1.76
8	.1644	6.875	.1681	7.031	.172	7.218	.160	2.26
7	.1793	7.50	—	—	.188	7.752	.190	2.68

Recommended Duct Metal Gauges

Rectangular Duct		Round Duct		
Greatest Dimension	Galvanized Steel	Diameter	Galvanized Steel	Aluminum
to 30 in	24	to 8 in	24	22
31- 60	22	9 - 24	22	20
61- 90	20	25 - 48	20	18
91- up	18	49 - 72	18	16

Note: Galvanized stated in U.S. gauge

Duct Pressure and Leakage Classes

SMACNA has defined duct leakage classes by the amount of leakage in CFM allowable per 100 square feet of duct surface area at a given pressure.

Once test pressure and leakage class are defined, allowable leakage is calculated as follows:

$$Lmax = Cl \times P^{0.65}$$

- **Lmax:** maximum permitted leakage in CFM / 100 sq. ft. of duct surface area
- **Cl:** duct leakage class in CFM / 100 sq. ft. at 1" w.g.
- **P:** test pressure (duct pressure class rating in.h w.g.

Duct seal classes are also established to correspond to static pressure classes, indicating what duct elements must be sealed.

Duct Leakage and Seal Classes

Duct Class	1/2", 1", 2" w.g.		3" w.g.	4", 6", 10" w.g.
Seal Class	None	C	B	A
Sealant Application	N/A	Transverse Joints Only	Transverse Joints and Seams	Joints, Seams & All Wall Penetrations
Leakage Class (CFM/100 sq ft @ 1" 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

Note: Any VAV system duct of 1/2" and 1" w.g. construction classes upstream of the VAV box shall meet Seal Class C

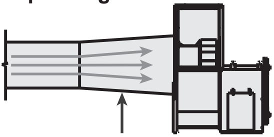
Adapted from SMACNA HVAC Duct Systems Design

Duct Design

Typical Inlet Conditions

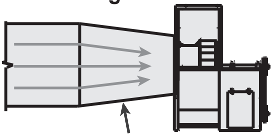
Correct Installation Examples

Expanding



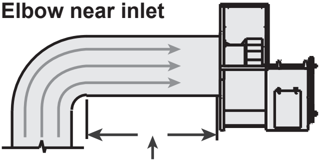
Limit slope to 7º diverging

Contracting



Limit slope to 15º converging

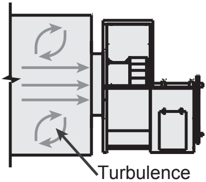
Elbow near inlet



Minimum 2.5 inlet diameters

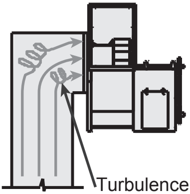
Incorrect Installation Examples

Abrupt contraction near inlet



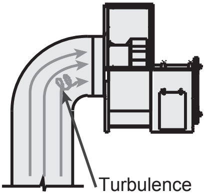
Turbulence

Elbow near inlet



Turbulence

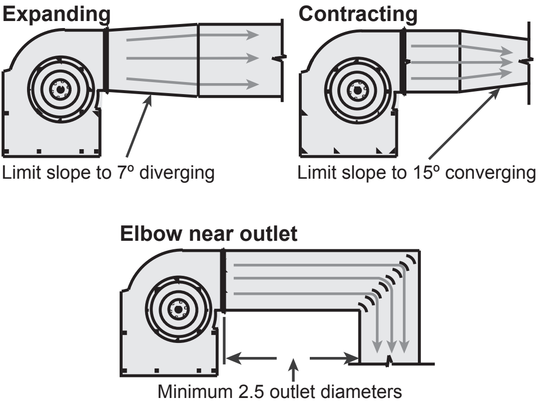
Elbow near inlet



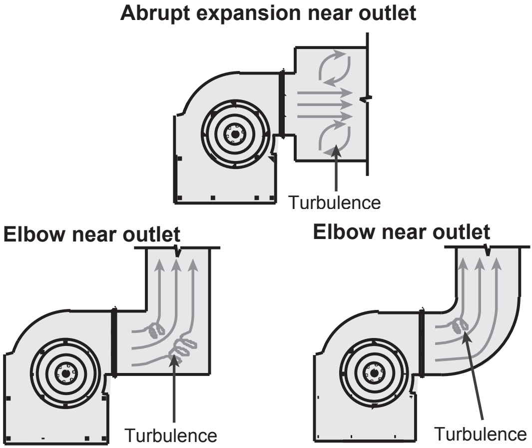
Turbulence

Typical Outlet Conditions

Correct Installation Examples



Incorrect Installation Examples



Adapted from ANSI/AMCA Standard 99-10

Motors & Drives

Terms & Formulas

Alternating Current: Electric current that alternates or reverses at a defined frequency, typically 60 cycles per second (hertz) in the U.S.

Breakdown Torque: The maximum torque a motor will develop with rated voltage and frequency applied without an abrupt drop in speed.

Efficiency: A rating of how much input power an electric motor converts to actual work at the rotating shaft expressed in percent.

$$\%\ efficiency = (power\ out\ /\ power\ in) \times 100$$

Locked Rotor Torque: The minimum torque that a motor will develop at rest for all angular positions of the rotor with rated voltage and frequency applied.

Rated Load Torque: The torque necessary to produce rated horsepower at rated load speed.

Single Phase AC: Typical household type electric power consisting of a single alternating current at 110-115 volts.

Slip: The difference between synchronous speed and actual motor speed. Usually expressed in percent slip.

$$\%\ slip = \frac{(synchronous\ speed - actual\ speed)}{synchronous\ speed} \times 100$$

Synchronous speed: The speed of the rotating magnetic field in an electric motor.

$$\text{Synchronous Speed} = (60 \times 2f)\ /\ p$$

Where: f = frequency of the power supply

p = number of poles in the motor

Three Phase AC: Typical industrial electric power consisting of 3 alternating currents of equal frequency differing in phase of 120 degrees from each other. Available in voltages ranging from 200 to 575 volts for typical industrial applications.

Torque: A measure of rotational force defined in foot-pounds or Newton-meters.

$$Torque\ lb\text{-}ft. = (HP \times 5252\ lb\text{-}ft.)\ /\ RPM$$

Motor Efficiency & EISA

Motor efficiency is a measure of the ability of a motor to convert electrical energy into mechanical work. Higher efficiency motors will consume less energy than their standard efficiency counterparts. Over 60% of energy usage in the United States is used to power motors.

The Energy Independence and Security Act (EISA) of 2007 establishes minimum energy efficiency levels for general purpose, three-phase AC motors from 1 to 500 HP manufactured for sale in the U.S. Effective December 19, 2010, EISA expands upon EPAct (Energy Policy Act, effective October 24, 1997) which covered T-frame, single speed, three-phase industrial motors from 1 to 200HP manufactured for sale in the US.

Under EISA motors are divided into two categories: Subtype I and Subtype II

- **Subtype I** motors include all integral horsepower 3 phase motors of less than 600 volts except those configurations covered by subtype II. Efficiencies are covered by NEMA MG-1 Table 12-12. Configurations include:
  - ODP and TEFC motors
  - Nominal 3600, 1800 and 1200 RPM motors
  - Frame sizes 143T – 445T (56 frame motors are not included)
- **Subtype II** motors are essentially specific exceptions to the subtype I definition. They are covered by NEMA MG-1 Table 12-11. The configurations are as follows:
  - Design C motors
  - U-frame motors
  - Close-coupled pump motors
  - Footless motors
  - Vertical solid shaft normal thrust motor (as tested in a horizontal configuration)
  - 8 pole motors (nominal 900 RPM)
  - Polyphase motors rated less than 600V other than 230 or 460V
  - 201 – 500 HP motors not previously covered by EPAct (Energy Policy Act of 1992)

Premium Efficient AC Motors Full-Load Efficiencies

Open Motors (ODP)

HP	2 POLE Efficiency		4 POLE Efficiency		6 POLE Efficiency		8 POLE Efficiency	
	Nom	Min	Nom	Min	Nom	Min	Nom	Min
1	77.0	74.0	85.5	82.5	82.5	80.0	75.5	72.0
1.5	84.0	81.5	86.5	84.0	86.5	84.0	77.0	74.0
2	85.5	82.5	86.5	84.0	87.5	85.5	86.5	84.0
3	85.5	82.5	89.5	84.0	88.5	86.5	87.5	85.5
5	86.5	84.0	89.5	84.0	89.5	87.5	88.5	86.5
7.5	88.5	86.5	91.0	89.5	90.2	88.5	89.5	87.5
10	89.5	87.5	91.7	90.2	91.7	90.2	90.2	88.5
15	90.2	88.5	93.0	91.7	91.7	90.2	90.2	88.5
20	91.0	89.5	93.0	91.7	92.4	91.0	91.0	89.5
25	91.7	90.2	93.6	92.4	93.0	91.7	91.0	89.5
30	91.7	90.2	94.1	93.0	93.6	92.4	91.7	90.2
40	92.4	91.0	94.1	93.0	94.1	93.0	91.7	90.2
50	93.0	91.7	94.5	93.6	94.1	93.0	92.4	91.0
60	93.6	92.4	95.0	94.1	94.5	93.6	93.0	91.7
75	93.6	92.4	95.0	94.1	94.5	93.6	94.1	93.0
100	93.6	92.4	95.4	94.5	95.0	94.1	94.1	93.0
125	94.1	93.0	95.4	94.5	95.0	94.1	94.1	93.0
150	94.1	93.0	95.8	95.0	95.4	94.5	94.1	93.0
200	95.0	94.1	95.8	95.0	95.4	94.5	94.1	93.0
250	95.0	94.1	95.8	95.0	95.4	94.5	95.0	94.1
300	95.4	94.5	95.8	95.0	95.4	94.5	—	—
350	95.4	94.5	95.8	95.0	95.4	94.5	—	—
400	95.8	95.0	95.8	—	—	95.0	—	—
450	95.8	95.0	96.2	—	—	95.4	—	—
500	95.8	95.0	96.2	—	—	95.4	—	—

[MG 1 Table 12-12]

Premium Efficient AC Motors Full-Load Efficiencies

Enclosed Motors (TEFC, TENV)

HP	2 POLE		4 POLE		6 POLE		8 POLE	
	Efficiency		Efficiency		Efficiency		Efficiency	
	Nom	Min	Nom	Min	Nom	Min	Nom	Min
1	77.0	74.0	85.5	82.5	82.5	80.0	75.5	72.0
1.5	84.0	81.5	86.5	84.0	87.5	85.5	78.5	75.5
2	85.5	82.5	86.5	84.0	88.5	86.5	84.0	81.5
3	86.5	84.0	89.5	87.5	89.5	87.5	85.5	82.5
5	88.5	86.5	89.5	87.5	89.5	87.5	86.5	84.0
7.5	89.5	87.5	91.7	90.2	91.0	89.5	86.5	84.0
10	90.2	88.5	91.7	90.2	91.0	89.5	89.5	87.5
15	91.0	89.5	92.4	91.0	91.7	90.2	89.5	87.5
20	91.0	89.5	93.0	91.7	91.7	90.2	90.2	88.5
25	91.7	90.2	93.6	92.4	93.0	91.7	90.2	88.5
30	91.7	90.2	93.6	92.4	93.0	91.7	91.7	90.2
40	92.4	91.0	94.1	93.0	94.1	93.0	91.7	90.2
50	93.0	91.7	94.5	93.6	94.1	93.0	92.4	91.0
60	93.6	92.4	95.0	94.1	94.5	93.6	92.4	91.0
75	93.6	92.4	95.4	94.5	94.5	93.6	93.6	92.4
100	94.1	93.0	95.4	94.5	95.0	94.1	93.6	92.4
125	95.0	94.1	95.4	94.5	95.0	94.1	94.1	93.0
150	95.0	94.1	95.8	95.0	95.8	95.0	94.1	93.0
200	95.4	94.5	96.2	95.4	95.8	95.0	94.5	93.6
250	95.8	95.0	96.2	95.4	95.8	95.0	95.0	94.1
300	95.8	95.0	96.2	95.4	95.8	95.0	—	—
350	95.8	95.0	96.2	95.4	95.8	95.0	—	—
400	95.8	95.0	96.2	95.4	—	—	—	—
450	95.8	95.0	96.2	95.4	—	—	—	—
500	95.8	95.0	96.2	95.4	—	—	—	—

[MG 1 Table 12-12]

Alternating Current Motor Characteristics

Single Phase AC Motors

- Fan applications requiring less than 1hp
- Four types (see chart below)
- Multi-speed motors are single-speed motors modified to operate at two or more speeds with internal or external modifications

Single Phase AC Motors (60Hz)

Motor Type	HP Range	Average Efficiency	Average Slip	Poles/ RPM	Use
Shaded Pole	1/6 to 1/4 HP	30%	14%	4/1550 6/1050	Small direct drive fans (low start torque)
Permanent Split Capacitor	Up to 1/3 HP	50%	10%	4/1625 6/1075	Small direct drive fans (low start torque)
Split-phase	Up to 1/2 HP	65%	4%	2/3450 4/1725 6/1140 8/850	Small belt drive fans (good start torque)
Capacitor Start	1/2 to 3/4 HP	65%	4%	2/3450 4/1725 6/1140 8/850	Small belt drive fans (good start torque)

Three Phase AC Motors

Most common motor for fan applications is 3-phase squirrel cage induction motor. Three-phase current produces a rotating magnetic field in stator which, in turn, induces a magnetic field in rotor. Attraction/repulsion of these fields on a rotating basis causes rotor to turn.

Operational characteristics

- Constant speed
- Simple construction
- Relatively high starting torque

Typical Speeds Three Phase AC Motors

Number of Poles	60 Hz Synchronous Speed (RPM)	50 Hz Synchronous Speed (RPM)
2	3600	3000
4	1800	1500
6	1200	1000
8	900	750

Notes:

- Actual motor speed is somewhat less than synchronous speed due to slip
- Motors of 5% or less slip are referred to as 'normal slip' motors
- Normal slip motor speeds change very little with load variations.
- Motor manufacturers generally specify actual speed on the nameplate rather than synchronous speed

NEMA Torque Designs For Three Phase Motors

NEMA Torque Designs	Starting Current	Locked Rotor	Breakdown Torque	Slip %
Type B	medium	medium torque	high	max. 5%
Type C	medium	high torque	medium	max. 5%
Type D	medium	extra-high torque	low	5% or more

NEMA Designs	**Application**
Type B	Normal starting torque for fans, blowers, rotary pumps, compressors, conveyors, machine tools. Constant load speed.
Type C	High inertia start requirements such as large centrifugal blowers, flywheels, and crusher drums. As well as loaded starts such as piston pumps, compressors, and conveyors. Constant load speed.
Type D	Very high inertia and loaded starts such as punch presses, shears, and forming machine tools. As well as cranes, hoists, elevators, and oil well pumping jacks. Considerable variation in speed under load.

AC Motor Full Load Amp Draw

Single Phase Motors

HP	Amp Draw		
	115V	200V	230V
1/6	4.4	2.5	2.2
1/4	5.8	3.3	2.9
1/3	7.2	4.1	3.6
1/2	9.8	5.6	4.9
3/4	13.8	7.9	6.9
1	16	9.2	8
1-1/2	20	11.5	10
2	24	13.8	12
3	34	19.6	17
5	56	32.2	28
7-1/2	80	46	40
10	100	57.5	50

Note: For motors running at usual speeds and motors with normal torque characteristics.

Based on Table 430-248 of the National Electric Code®, 2011

Three Phase Motors

AC Induction Type-Squirrel Cage and Wound Rotor Motors (For conductor sizing only)

HP	Amp Draw				
	115V	208V	230V	460V	575V
1/2	4.4	2.4	2.2	1.1	0.9
3/4	6.4	3.5	3.2	1.6	1.3
1	8.4	4.6	4.2	2.1	1.7
1-1/2	12	6.6	6	3	2.4
2	13.6	7.5	6.8	3.4	2.7
3		10. 6	9.6	4.8	3.9
5		16.7	15.2	7.6	6.1
7-1/2		24.2	22	11	9
10		30.8	28	14	11
15		46.2	42	21	17
20		59.4	54	27	22
25		74.8	68	34	27
30		88	80	40	32
40		114	104	52	41
50		143	130	65	52
60		169	154	77	62
75		211	192	96	77
100		273	248	124	99
125		343	312	156	125
150		396	360	180	144
200		528	480	240	192
Over 200 HP approx. amps per HP		2.65	2.4	1.2	0.96

Note: Branch-circuit conductors supplying a single motor shall have an ampacity not less than 125 percent of the motor full-load current rating.

Based on Table 430-250 of the National Electrical Code®, 2011.

Motor Service Factors

Motor Service Factor is a factor applied to the rated HP indicating the load the motor can safely handle. Example: A 10 HP motor with a 1.15 service factor can handle 11.5 HP of load.

In general, motors should be selected within the rated horsepower (not using the service factor). This allows the motor to better withstand adverse conditions such as higher than normal ambient conditions, voltage fluctuations, and occasional overload.

Motor Insulation Classes

Electrical motor insulation classes are rated by their resistance to thermal degradation.

Four common NEMA insulation classes are A, B, F and H with temperature ratings as follows:

- Class A: 105°C (221°F)
- Class B: 130°C (266°F)
- Class F: 155°C (311°F)
- Class H: 180°C (356°F)

Insulation class rating of any motor must be able to withstand at least the maximum ambient temperature plus the temperature rise that occurs as a result of continuous full load operation.

Motor HP Rerate for Altitude and Temp

Altitude Above Sea Level (feet)	Ambient Temperature (°F)						
	86	95	104	113	112	131	140
3300	1.06	1.03	1	0.96	0.92	0.87	0.82
5000	1.03	1	0.97	0.93	0.89	0.84	0.8
6600	1	0.97	0.94	0.9	0.86	0.82	0.77
8300	0.95	0.93	0.9	0.86	0.83	0.78	0.74
9900	0.91	0.89	0.86	0.83	0.79	0.75	0.71
11500	0.87	0.84	0.82	0.79	0.75	0.71	0.67
13125	0.82	0.79	0.77	0.74	0.71	0.67	0.63

Example:

BHP at Standard Air	25
Site Ambient Temp	95°F
Site Altitude	9900
Reduction Factor	0.89
Rerated HP @ Alt and Temp	= 25/0.89 = 28.1
Motor HP Selection	30 HP

Locked Rotor kVA/HP

Locked rotor kVA per horsepower is a rating commonly specified on motor nameplates. The rating is shown as a code letter on the nameplate which represents various kVA/HP ratings.

Code Letter	kVA/HP	Code Letter	kVA/HP	Code Letter	kVA/HP
A	0 - 3.15	J	7.1 - 8.0	T	18.0 - 20.0
B	3.15 - 3.55	K	8.0 - 9.0	U	20.0 - 22.4
C	3.55 - 4.0	L	9.0 - 10.0	V	22.4 and up

A letter at the beginning of the alphabet indicates a low starting current, and a letter at the end of the alphabet indicates a high starting current. Starting current can be calculated using the following formula:

Starting current is calculated as follows:

$$\text{Starting current} = \frac{(1000 \times HP \times kVA/HP)}{(1.73 \times \text{Voltage})}$$

General Effect of Voltage and Frequency Variations on Induction Motor Characteristics

Characteristic	Voltage	
	110%	90%
Starting torque	Up 21%	Down 19%
Maximum torque	Up 21%	Down 19%
Percent slip	Down 15-20%	Up 20-30%
Efficiency - full load	Down 0-3%	Down 0-2%
3/4 Load	0 - Down slightly	Little change
1/2 Load	Down 0-5%	Up 0-1%
Power factor - full load	Down 5-15%	Up 1-7%
3/4 Load	Down 5-15%	Up 2-7%
1/2 Load	Down 10-20%	Up 3-10%
Full load current	Up 0-5%	Up 5-10%
Starting current	Up 10%	Down 10%
Full load - temperature rise	Up 10%	Down 10-15%
Maximum overload capacity	Up 21%	Down 19%
Magnetic noise	Up slightly	Down slightly

Characteristic	Frequency	
	105%	95%
Starting torque	Down 10%	Up 11%
Maximum torque	Down 10%	Up 11%
Percent slip	Up 10-15%	Down 5-10%
Efficiency - full load	Up slightly	Down slightly
3/4 load	Up slightly	Down slightly
1/2 load	Up slightly	Down slightly
Power factor - full load	Up slightly	Down slightly
3/4 load	Up slightly	Down slightly
1/2 load	Up slightly	Down slightly
Full load current	Down slightly	Up slightly
Starting current	Down 5%	Up 5%
Full load - temperature rise	Down slightly	Up slightly
Maximum overload capacity	Down slightly	Up slightly
Magnetic noise	Down slightly	Up slightly

AC Motor Troubleshooting Guide

Problem	Like Causes	What to Do
Motor fails to start upon initial	Motor is miswired.	Verify motor is wired correctly.
	Motor damaged and rotor is striking stator.	May be able to reassemble; otherwise, motor should be replaced.
	Fan guard bent and contacting fan.	Replace fan guard.
Motor has been running, then fails to start	Fuse or circuit breaker tripped.	Replace fuse or reset the breaker.
	Motor overload or load jammed.	Inspect to see that the load is free. Verify amp draw of motor versus nameplate rating.
	Capacitor (on single phase motor) may have failed.	First discharge capacitor. Check capacitor. If capacitor is OK, needle will jump to zero ohms, and drift back to high. Steady zero ohms indicates a short circuit; steady high ohms indicates an open circuit.
Motor runs but dies down	Voltage drop	If voltage is less than 10% of the motor's rating contact power company or check if some other equipment is taking power away from the motor.
	Load increased.	Verify the load has not changed. Verify equipment hasn't got tighter. If fan application verify the air flow hasn't changed.
Motor takes too long to accelerate	Defective capacitor.	Test capacitor.
	Faulty stationary switch.	Inspect switch contacts and connections. Verify that switch reeds have some spring in them.
	Bad bearings.	Noisy or rough feeling bearings should be replaced.
	Voltage too low.	Make sure that the voltage is within 10% of the motor's nameplate rating. If not, contact power company or check if some other equipment is taking power away from the motor.
Motor runs in the wrong direction	Incorrect wiring.	Rewire motor according to wiring schematic provided.

Continued on next page

AC Motor Troubleshooting Guide (cont.)

Problem	Like Causes	What to Do
Motor overload protector continually trips	Load too high.	Verify that the load is not jammed. If motor is a replacement, verify that the rating is the same as the old motor. If previous motor was a special design, a stock motor may not be able to duplicate the performance. Remove the load from the motor and inspect the amp draw of the motor unloaded. It should be less than the full load rating stamped on the nameplate.
	Ambient temperature too high.	Verify that the motor is getting enough air for proper cooling. Most motors are designed to run in an ambient temperature of less than 40°C. (Note: A properly operating motor may be hot to the touch.)
	Protector may be defective.	Replace the motor's protector with a new one of the same rating.
	Winding shorted or grounded.	Inspect stator for defects, or loose or cut wires that may cause it to go to ground.
Motor vibrates	Motor misaligned to load.	Realign load.
	Load out of balance. (Direct drive application.)	Remove motor from load and inspect motor by itself. Verify that motor shaft is not bent. Rule of thumb is .001" runout per every inch of shaft length.
	Motor bearings defective.	Test motor by itself. If bearings are bad, you will hear noise or feel roughness. Replace bearings. Add oil if a sleeve of bearing. Add grease if bearings have grease fittings.
	Motor may have too much endplay.	With the motor disconnected from power turned shaft. It should move but with some resistance. If the shaft moves in and out too freely, this may indicate a preload problem and the bearings may need additional shimming.

Continued on next page

Problem	Like Causes	What to Do
Bearings continuously fail	Load to motor may be excessive or unbalanced.	Besides checking load, also inspect drive belt tension to ensure it's not too tight may be too high. An unbalanced load will also cause the bearings to fail.
	High ambient temperature.	If the motor is used in a high ambient, a different type of bearing grease may be required. You may need to consult the factory or a bearing distributor.
Start capacitors continuously fail	The motor is not coming up to speed quickly enough.	Motor may not be sized properly. Verify how long the motor takes to come up to speed, Most single phase capacitor start motors should come up to speed within three seconds. Otherwise the capacitors may fail.
	The motor is being cycled too frequently.	Verify duty cycle. Capacitor manufacturers recommend no more than 20, three-second starts per hour. Install capacitor with higher voltage rating, or add bleed resistor to the capacitor.
	Voltage to motor is too low.	Verify that voltage to the motor is within 10% of the nameplate value. If the motor is rated 208-230V, the deviation must be calculated from 230V.
	Starting switch may be defective, preventing the motor from coming out of start winding.	Replace switch.
Run capacitor fail	Ambient temperature too high.	Verify that ambient does not exceed motor's nameplate value.
	Possible power surge to motor, caused by lightning strike or other high transient voltage.	If a common problem, install surge protector.

Electronically Commutated Motors

Electronically commutated motors (also known as 'EC' motors) are rapidly replacing traditional AC motors for HVAC equipment in the range of 1HP and below. They offer significant improvements in efficiency and controllability over shaded pole and permanent split capacitor motors.

EC motors have a rotor with permanently magnetized poles instead of the wound electromagnetic rotors in traditional motors.

This results in an increase in efficiency, described below, but it requires a special controller known as a 'commutator' to make the EC motor rotate. The commutator switches the polarity of the stator poles in a timed sequence, propelling the rotor forward.

The following information breaks down and compares inefficiencies or 'losses' within AC and EC In descending order of magnitude.

AC Induction Motor Losses

- Stator losses – Heat created by current flowing through stator windings (Typically 30% of total losses)
- Rotor losses – Heat created by current flowing through rotor windings, as well as motor slip which increases the current through rotor windings resulting in additional heat (typically 20% of total losses)
- Iron core losses – These losses are due to hysteresis and eddy currents which create 'resistance' in the core (Typically 18-20% of total losses)
- Friction Losses – Friction in bearings, windage, etc. (Typically 10-15% of total losses)
- Stray Load Losses – Other miscellaneous losses that cannot be accounted for above.

EC Motor Losses

- Stator losses – Heat created by current flowing through stator windings
- Rotor losses – None
- Electronic losses – Inherent low level losses associated with driving electronics with electricity
- Other losses – Friction in bearings, windage, etc.
- By replacing motor windings in the rotor with permanent

magnets, approximately 20% of the total losses due to current flow and slip are eliminated. These are replaced with very minor inefficiencies in powering the electronic controls.

Reduced Speeds

AC Induction motors slip more as their speed is reduced, increasing rotor losses. EC motors are 'no slip' motors and so the efficiency gains associated with EC motors are greater as motors are slowed down.

Other Advantages of EC Motors

- EC motors are often specified or selected due to their controllability. Integrated electronics allow the motor to be controlled accurately and efficiently using a wide variety of inputs such as temperature or pressure.
- Due to increased efficiency, EC motors run cooler than AC counterparts and are often physically smaller
- Finally they are generally quieter than traditional AC induction motors.

Summary

Advantages to Electronically Commutated (EC) motors are offset by an increase in cost. However, when the cost of a VFD is added or when compared to a belt drive application, the cost disparity is reduced.

Finally, savings in reduced energy consumption can help reduce life cycle cost and, under certain conditions, even provide an attractive payback.

Allowable Ampacities of Not More Than Three Insulated Conductors

Copper Conductor

AWG kcmil	Temperature Rating		
	60°C (140°F)	75°C (167°F)	90°C (194°F)
18**	—	—	14
16**	—	—	18
14**	15*	20*	25*
12**	20*	25*	30*
10**	30	35*	40*
8	40	50	55
6	55	65	75
4	70	85	95
3	85	100	115
2	95	115	130
1	110	130	145
1/0	125	150	170
2/0	145	175	195
3/0	165	200	225
4/0	195	230	260
250	215	255	290
300	240	285	320
350	260	310	350
400	280	335	380
500	320	380	430
600	350	420	475
700	385	460	520
750	400	475	535
800	410	490	555
900	435	520	585
1000	455	545	615
1250	495	590	665
1500	525	625	705
1750	545	650	735
2000	555	665	750

Rated 0-2000 Volts, 60° to 90°C (140° to 194°F), in Raceway or Cable or Earth (directly buried). Based on ambient air temperature of 30°C (86°F). Refer to 310.15(B)(2) for ampacity correction factors.

Aluminum or Copper-Clad Conductor

AWG kcmil	Temperature Rating		
	60°C (140°F)	75°C (167°F)	90°C (194°F)
12	20*	20*	25*
10	25	30*	35*
8	30	40	45
6	40	50	60
4	55	65	75
3	65	75	85
2	75	90	100
1	85	100	115
1/0	100	120	135
2/0	115	135	150
3/0	130	155	175
4/0	150	180	205
250	170	205	230
300	190	230	255
350	210	250	280
400	225	270	305
500	260	310	350
600	285	340	385
700	310	375	420
750	320	385	435
800	330	395	450
900	355	425	480
1000	375	445	500
1250	405	485	545
1500	435	520	585
1750	455	545	615
2000	470	560	630

Adapted from NFPA 70-1993, National Electrical Code®, Copyright 1992

Notes:

- 60°C (140°F): Types TW, UF
- 75°C (167°F): Types RHW, THHW, THW, THWN, XHHW & USE. ZW (Copper only)
- 90°C (194°F): Types TBS, SA, SIS, THHN, THHW, THW-2, RHH, RHW-2, USE-2, XHH, XHHW, XHHW-2 & ZW-2. FEPB & MI (Copper only)
- Refer to NEC 240.4(D) for conductor overcurrent protection limits.

Belts

Most fan drive systems are based on the standard "V" drive belt which is relatively efficient and readily available. The use of a belt drive allows fan RPM to be easily selected through a combination of AC motor RPM and drive pulley (sheave) ratios.

In general, select a pulley (sheave) combination that will result in the correct drive ratio with the smallest sheave pitch diameters. Depending upon belt cross section, there may be some minimum pitch diameter considerations. Multiple belts and sheave grooves may be required to meet horsepower requirements.

$$\textit{drive ratio} = \frac{\textit{motor RPM}}{\textit{desired fan RPM}}$$

V-belt Length Formula

Once a sheave combination is selected, calculate approximate belt length. Calculate the approximate V-belt length using the following formula:

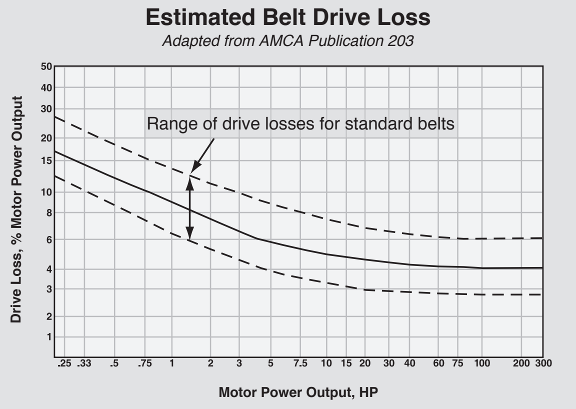
L = pitch length of belt
C = center distance of pulley (sheave)
D = diameter of large pulley (sheave)
d = diameter of small pulley (sheave)

$$L = 2C + 1.57\,(D+d) + \frac{(D-d)^2}{4C}$$

Belt Drive Guidelines

- Drives should always be installed with provision for center distance adjustment.
- If possible, centers should not exceed 3 times the sum of the sheave diameters nor be less than the diameter of the large sheave.
- If possible, the arc of contact of the belt on the smaller sheave should not be less than 120°.
- Be sure that shafts are parallel and sheaves are in proper alignment. Check at startup and after first eight hours of operation.
- Do not drive sheaves on or off shafts. Be sure shaft and keyway are smooth and that bore and key are of correct size.

Estimated Belt Drive Loss



Notes:

- 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 is determined to be 13.3 HP.
- From the chart above, the drive loss = 5.1%

  Drive loss = 0.051 × 13.3 = 0.7 HP
- Fan power input = *13.3 - 0.7 HP = 12.6 HP*

Adapted from AMCA Publication 203-90 (R2008)

Bearing Life

It has been shown that physically identical bearings being utilized under the same conditions may fail at different times. Nevertheless, the customer or user must have some way of estimating the expected life of the bearing.

The American Bearing Manufacturers Association (ABMA) and International Organization for Standardization (ISO) have established ISO 281:2007 as the standard for calculation of bearing life and ratings.

Basic Life (L10) is defined as the amount of time that 90% of relatively large sampling of physically identical bearings can be expected to reach before requiring replacement. This is generally the preferred specification when referring to required bearing life.

Median or Average Life (L50) is simply five times the calculated Basic Life. Although this is sometimes referred to in specifications, it is not the preferred method of quantifying bearing life.

Example:

A manufacturer might specify that the bearings supplied for a particular fan should have an L10 life of 40,000 hours at the maximum cataloged operating speed. This can be interpreted to mean 90% of the bearings applied to this product can be expected to last 40,000 hours or longer, and 10% or less of the bearings could fail within 40,000 hours of operation.

Bearing Life Equivalency Examples:

- L10 - 40,000 = L50 - 200,000
- L10 - 80,000 = L50 - 400,000
- L10 - 100,000 = L50 - 500,000
- L10 - 200,000 = L50 -1,000,000

Notes

Moisture & Air Relationships

If humidity control is of primary concern determine the necessary air volume (CFM) to reduce the latent load (humidity)

ASHRAE has adopted pounds of moisture per pound of dry air as standard nomenclature. Relations of other units are expressed below at various dew point temperatures.

Equivalent Dew Point °F	Lb H_2O/ Lb Dry Air	Parts Per Million	Grains/ Lb Dry Air*	Percent Moisture %**
-100	0.000001	1	0.0007	—
-90	0.000002	2	0.0016	—
-80	0.000005	5	0.0035	—
-70	0.00001	10	0.073	0.06
-60	0.00002	21	0.148	0.13
-50	0.00004	42	0.291	0.26
-40	0.00008	79	0.555	0.5
-30	0.00015	146	1.02	0.9
-20	0.00026	263	1.84	1.7
-10	0.00046	461	3.22	2.9
0	0.0008	787	5.51	5.0
10	0.0013	1,315	9.20	8.3
20	0.0022	2,152	15.1	13.6
30	0.0032	3,154	24.2	21.8
40	0.0052	5,213	36.5	33.0
50	0.0077	7,658	53.6	48.4
60	0.0111	11,080	77.6	70.2
70	0.0158	15,820	110.7	100.0
80	0.0223	22,330	156.3	—
90	0.0312	31,180	218.3	—
100	0.0432	43,190	302.3	—

* 7000 grains = 1 lb

** Compared to 70°F saturated

Normally the sensible heat factor determines the CFM required to accept a load. In some industrial applications, the latent heat factor may control the air circulation rate.

$$CFM = \frac{latent\ heat \times (BTU/hr)}{(W_2 - W_1) \times 4840}$$

where $(W_2 - W_1)$ = humidity ratio difference in lbs of H_2O/lb of dry air (determined from column 2)

Adapted from "Numbers," by Bill Hollady & Cy Otterholm 1985.

Properties of Saturated Steam

Temperature °F	Pressure PSIA	Specific Volume	Specific Enthalpy	
		Saturated Vapor Ft³/lbm	Saturated Liquid BTU/lbm	Saturated Vapor BTU/lbm
32	0.08859	3304.7	-0.0179	1075.5
40	0.12163	2445.8	8.027	1079.0
60	0.25611	1207.6	28.060	1087.7
80	0.50683	633.3	48.037	1096.4
100	0.94924	350.4	67.999	1105.1
120	1.6927	203.26	87.97	1113.6
140	2.8892	123.00	107.95	1122.0
160	4.7414	77.29	127.96	1130.2
180	7.5110	50.22	148.00	1138.2
200	11.526	33.639	168.09	1146.0
212	14.696	26.799	180.17	1150.5
220	17.186	23.148	188.23	1153.4
240	24.968	16.321	208.45	1160.6
260	35.427	11.762	228.76	1167.4
280	49.200	8.644	249.17	1173.8
300	67.005	6.4658	269.7	1179.7
320	89.643	4.9138	290.4	1185.2
340	117.992	3.7878	311.3	1190.1
360	153.010	2.9573	332.3	1194.4
380	195.729	2.3353	353.6	1198.0
400	247.259	1.8630	375.1	1201.0
420	308.780	1.4997	396.9	1203.1
440	381.54	1.21687	419.0	1204.4
460	466.87	0.99424	441.5	1204.8
480	566.15	0.81717	464.5	1204.1
500	680.86	0.67492	487.9	1202.2
520	812.53	0.55957	512.0	1199.0
540	962.79	0.46513	536.8	1194.3
560	1133.38	0.38714	562.4	1187.7
580	1326.17	0.32216	589.1	1179.0
600	1543.2	0.26747	617.1	1167.7
620	1786.9	0.22081	646.9	1153.2
640	2059.9	0.18021	679.1	1133.7
660	2365.7	0.14431	714.9	1107.0
680	2708.6	0.11117	758.5	1068.5
700	3094.3	0.07519	822.4	995.2

Based on “1967 ASME Steam Tables”

Fuel Comparisons

This provides equivalent BTU data for various fuels.

Natural gas	1,000,000 BTU = 10 Therms or 1,000,000 BTU = (1000 Cu. Ft.)
Propane gas	1,000,000 BTU = 46 Lb. or 1,000,000 BTU = 10.88 Gallon
No. 2 fuel oil	1,000,000 BTU = 7.14 Gallon
Electrical resistance	1,000,000 BTU = 293 KW (Kilowatts)
Municipal steam	1,000,000 BTU = 1000 Lbs. Condensate
Sewage gas	1,000,000 BTU = 1538 Cu.Ft. to 2380 Cu.Ft.
LP/air gas	1,000,000 BTU = 46 Lb. Propane or 1,000,000 BTU = 10.88 Gallon Propane or 1,000,000 BTU = 690 Cu.Ft. Gas/Air Mix

From Chemical Rubber Publishing Co., Handbook of Chemistry and Physics

Fuel Gas Characteristics

Natural gas	925 to 1125 BTU/Cu.Ft.	.6 to .66 Specific Gravity
Propane gas	2550 BTU/Cu.Ft.	1.52 Specific Gravity
*Sewage gas	420 to 650 BTU/Cu.Ft.	.55 to .85 Specific Gravity
*Coal gas	400 to 500 BTU/Cu.Ft.	.5 to .6 Specific Gravity
*LP/air mix	1425 BTU/Cu.Ft.	1.29 Specific Gravity

* Before attempting to operate units on these fuels, contact manufacturer.

Annual Fuel Use

Annual fuel use may be determined for a building by using one of the following formulas:

Electric resistance heating

$H/(\Delta T \times 3413 \times E) \times D \times 24 \times C_D = KWH/Year$

Natural gas heating

$H/(\Delta T \times 100{,}000 \times E) \times D \times 24 \times C_D = therms/year$

Propane gas heating

$H/(\Delta T \times 21739 \times E) \times D \times 24 \times C_D = pounds/year$

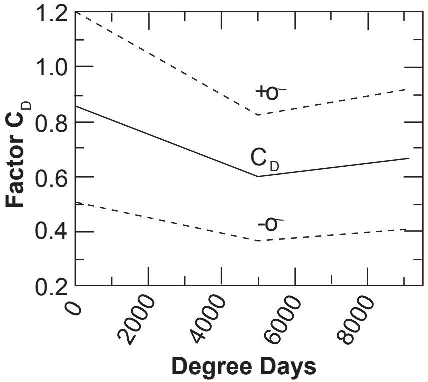
$H/(\Delta T \times 91911 \times E) \times D \times 24 \times C_D = gallons/year$

Oil heating

$H/(\Delta T \times 140{,}000 \times E) \times D \times 24 \times C_D = gallons/year$

Where:

ΔT = indoor design minus outdoor design temperature
H = building heat loss
D = annual degree days
E = seasonal efficiency (see above)
C_D = correlation factor C_D vs. degree-days



Water Flow & Piping

Pressure drop in piping varies approximate to the square of the flow:

$$\frac{h_2}{h_1} = \left(\frac{Q_2}{Q_1}\right)^2$$

The velocity of water flowing in a pipe is

$$V = \frac{gpm \times 0.41}{d^2}$$

Quiet Water Flows

Nom size	1/2	3/4	1	1-1/4	1-1/2	2	2-1/2	3	4
GPM	1.5	4	8	14	22	44	75	120	240

Six fps is a reasonable upper limit for water velocity in pipes. The relationship between pressure drop and flow rate can also be expressed:

$$h_2 = h_1 \times \left(\frac{Q_2}{Q_1}\right)^2 \text{ or } Q_2 = Q_1 \times \sqrt{\frac{h_2}{h_1}}$$

Example: If design values were 200 gpm and 40 ft head and actual flow were changed to 100 gpm, the new head would be:

$$h_2 = 40\left(\frac{100}{200}\right)^2 = 10ft$$

$$Pump\ hp = \frac{gpm \times ft\ head \times sp\ gr}{3960 \times \%\ efficiency}$$

Friction Loss for Water Flow

Feet loss / 100 ft - schedule 40 pipe

Average value for new pipe. Used pipe add 50%.

US GPM	1/2 in		3/4 in		1 in		1-1/4 in	
	v	h_F	v	h_F	v	h_F	v	h_F
2.0	2.11	5.5						
2.5	2.64	8.2						
3.0	3.17	11.2						
3.5	3.70	15.3						
4	4.22	19.7	2.41	4.8				
5	5.28	29.7	3.01	7.3				
6			3.61	10.2	2.23	3.1		
8			4.81	17.3	2.97	5.2		
10			6.02	26.4	3.71	7.9		
12					4.45	11.1	2.57	2.9
14					5.20	14.0	3.00	3.8
16					5.94	19.0	3.43	4.8

US GPM	1-1/2 in		2 in		2-1/2 in		1-1/4 in	
	v	h_F	v	h_F	v	h_F	v	h_F
18	2.84	2.8					3.86	6.0
20	3.15	3.4					4.29	7.3
22	3.47	4.1					4.72	8.7
24	3.78	4.8					5.15	10.3
26	4.10	5.5					5.58	11.9
28	4.41	6.3					6.01	13.7
30	4.73	7.2					6.44	15.6
35	5.51	9.6					7.51	20.9
40	6.30	12.4	3.82	3.6				
45	7.04	15.5	4.30	4.4				
50			4.78	5.4				
60			5.74	7.6	4.02	3.1		
70			6.69	10.2	4.69	4.2	3 in	
80			7.65	13.1	5.36	5.4	v	h_F
100					6.70	8.2		
120					8.04	11.5	5.21	3.9
140					9.38	15.5	6.08	5.2
160							6.94	6.7
180							7.81	8.4
200							8.68	10.2

v = feet per second. h_F= feet of head

Adapted from “Numbers”, Bill Holladay and Cy Otterholm, 1985

Equivalent Length of Pipe for Valves & Fittings

Screwed fittings, turbulent flow only, equivalent length in feet.

Fittings	Pipe Size							
	1/2"	3/4"	1"	1-1/4"	1-1/2"	2"	2-1/2"	3"
Standard 90° Ell	3.6	4.4	5.2	6.6	7.4	8.5	9.3	11
Long radius 90° Ell	2.2	2.3	2.7	3.2	3.4	3.6	3.6	4.0
Standard 45° Ell	0.71	0.92	1.3	1.7	2.1	2.7	3.2	3.9
Tee Line flow	1.7	2.4	3.2	4.6	5.6	7.7	9.3	12
Tee Br Flow	4.2	5.3	6.6	8.7	9.9	12	13	17
180° Return Bend	3.6	4.4	5.2	6.6	7.4	8.5	9.3	11
Globe Valve	22	24	29	37	42	54	62	79
Gate Valve	0.56	0.67	0.84	1.1	1.2	1.5	1.7	1.9
Angle Valve	15	15	17	18	18	18	18	18
Swing Check	8.0	8.8	11	13	15	19	22	27
Union or Coupling	0.21	0.24	0.29	0.36	0.39	0.45	0.47	0.53
Bellmouth Inlet	0.10	0.13	0.18	0.26	0.31	0.43	0.52	0.67
Sq mouth Inlet	0.96	1.3	1.8	2.6	3.1	4.3	5.2	6.7
Reentrant pipe	1.9	2.6	3.6	5.1	6.2	8.5	10	13
Sudden Enlargement	*Feet of liquid loss* = $\frac{(V_1 - V_2)^2}{2_g}$							

V_1 & V_2 = entering and leaving velocities in feet per second

$g = 32.17$ ft/sec²

Adapted from "Numbers", Bill Holladay and Cy Otterholm, 1985

Standard Pipe Dimensions

Schedule 40 (Steel)

Nominal Size	Diameter Outside (in)	Diameter Inside (in)	Area I.D. (sq. in.)	Volume gal/lin ft.	Weight lb/ lin ft
1/8	0.405	0.269	0.057	0.0030	0.244
1/4	0.540	0.364	0.104	0.0054	0.424
3/8	0.675	0.493	0.191	0.0099	0.567
1/2	0.840	0.622	0.304	0.0158	0.850
3/4	1.050	0.824	0.533	0.0277	1.13
1	1.315	1.049	0.864	0.0449	1.68
1-1/4	1.660	1.380	1.496	0.0777	2.27
1-1/2	1.900	1.610	2.036	0.1058	2.72
2	2.375	2.067	3.356	0.1743	3.65
2-1/2	2.875	2.469	4.788	0.2487	5.79
3	3.500	3.068	7.393	0.3840	7.57
4	4.500	4.026	12.730	0.6613	10.79
5	5.563	5.047	20.006	1.039	14.62
6	6.625	6.065	28.890	1.501	18.00

Copper Tube Dimensions

Type L

Nominal Size	Diameter Outside (in)	Diameter Inside (in)	Area I.D. (sq. in.)	Volume gal/lin ft.	Weight lb/lin ft
1/4	0.375	0.315	0.078	0.00404	0.126
3/8	0.500	0.430	0.145	0.00753	0.198
1/2	0.625	0.545	0.233	0.0121	0.285
5/8	0.750	0.666	0.348	0.0181	0.362
3/4	0.875	0.785	0.484	0.0250	0.455
1	1.125	1.025	0.825	0.0442	0.655
1-1/4	1.375	1.265	1.26	0.0655	0.884
1-1/2	1.625	1.505	1.78	0.0925	1.14
2	2.125	1.985	3.10	0.161	1.75
2-1/2	2.625	2.465	4.77	0.247	2.48
3	3.125	2.945	6.81	0.354	3.33
4	4.125	3.905	12.0	0.623	5.38

Adapted from “Numbers”, Bill Holladay and Cy Otterholm, 1985

Pump Impeller Types

- **Single Suction:** fluid enters impeller on one side only.
- **Double Suction:** fluid enters both sides of impeller.
- **Closed Impeller:** has a shroud which encloses the pump vanes, increasing efficiency. Used for fluid systems free of large particles which could clog impeller.
- **Semi-Open Impeller:** has no inlet shroud. Used for systems where moderate sized particles are suspended in pumped fluid.
- **Open Impeller:** has no shroud. Used for systems which have large particles suspended in pumped fluid, such as sewage or sludge systems.

Pump Bodies

- **Horizontal Split Case:** Split down centerline of pump horizontal axis. Disassembled by removing top half of pump body. Pump impeller mounted between bearings at center of shaft. Requires two seals. Usually double suction pump. Suction and discharge are in straight-line configuration.
- **Vertical Split Case:** Single-piece body casting attached to cover plate at the back of pump by capscrews. Pump shaft passes through seal and bearing in coverplate. Impeller is mounted on end of shaft. Suction is at right angle to discharge.

Pump Mounting Methods

- **Base Mount-Long Coupled:** Pump is coupled to base-mount motor. Motor can be removed without removing the pump from piping system. Typically standard motors are used.
- **Base Mount-Close Coupled:** Pump impeller is mounted on base mount motor shaft. No separate mounting is necessary for pump. Usually special motor necessary for replacement. More compact than long-coupled pump.
- **Line Mount:** Mounted to and supported by system piping. Usually resilient mount motor. Very compact. Usually for low flow requirements.

Pump Terms, Abbreviations & Conversion Factors

Term	Abbreviation	Multiply	By	To Obtain
Length	l	ft	0.3048	m
Area	A	ft^2	0.0929	m^2
Velocity	v	ft/s	0.3048	m/s
Volume	V	ft^3	0.0283	m^3
Flow rate	Q_V	gpm gpm	0.2272 0.0631	m^3/h L/s
Pressure	P	psi	6890	Pa
		psi	6.89	kPa
		psi	14.5	bar
Head (total)	H	ft	0.3048	m
NPSH	H	ft	0.3048	m
Output power (pump)	P_O	water HP (WHP)	0.7457	kW
Shaft power	P_S	BHP	0.7457	kW
Input power (driver)	P_i	kW	1.0	kW
Efficiency, %				
Pump	η_p	—	—	—
Equipment	η_e	—	—	—
Electric motor	η_m	—	—	—
Utilization	η_u	—	—	—
System Efficiency Index (decimal)	SEI	—	—	—
Speed	n	RPM RPM	0.1047 0.0167	rad/s rps
Density	r	lb/ft^3	16.0	kg/m^3
Temperature	°	°F-32	5/9	°C

Adapted from ASHRAE “Pocket Handbook”, Seventh Edition

Affinity Laws for Pumps

Impeller Diameter	Speed	Density	To Correct for	Multiply by
Constant	Variable	Constant	Flow	$\left(\frac{\text{new speed}}{\text{old speed}}\right)$
			Head	$\left(\frac{\text{new speed}}{\text{old speed}}\right)^2$
			BHP (or kW)	$\left(\frac{\text{new speed}}{\text{old speed}}\right)^3$
Variable	Constant	Constant	Flow	$\left(\frac{\text{new diameter}}{\text{old diameter}}\right)$
			Head	$\left(\frac{\text{new diameter}}{\text{old diameter}}\right)^2$
			BHP (or kW)	$\left(\frac{\text{new diameter}}{\text{old diameter}}\right)^3$
Constant	Constant	Variable	BHP (or kW)	$\frac{\text{new SG}}{\text{old SG}}$

Adapted from ASHRAE Pocket Guide, 8th Edition

Common Pump Formulas

Formula for	I-P Units
Head (ft)	H=psi × 2.31/SG
Output power (HP)	P_O = Qv × H × SG/3960
Shaft power (HP)	P_s = Qv × H × SG / 39.6 × E_p
Input power (kw)	P_i = P_s × 74.6/E_m
Utilization Q_D= design flow Q_A= actual flow H_D= design head H_A= actual head	hm = 100 (Q_DH_D/Q_AH_A)

SG = specific gravity

Notes

Pumping System Troubleshooting Guide

Noise

Possible Cause	Recommended Action
Shaft misalignment	Check and realign
Worn coupling	Replace and realign
Worn pump/motor bearings	• Replace, check manufacturer's lubrication recommendations • Check and realign shafts
Improper foundation or installation	• Check foundation bolting or proper grouting • Check possible shifting because of piping expansion/contraction • Realign shafts
Pipe vibration and/or strain caused by pipe expansion/contraction	Inspect, alter or add hangers and expansion provision to eliminate strain on pump(s)
Water velocity	• Check actual pump performance against specified and reduce impeller diameter as required • Check for excessive throttling by balance valves or control valves.
Pump operating close to or beyond end point of performance curve	Check actual pump performance against specified and reduce impeller diameter as required
Entrained air or low suction pressure	• Check expansion tank connection to system relative to pump suction • If pumping from cooling tower sump or reservoir, check line size • Check actual ability of pump against installation requirements • Check for vortex entraining air into suction line

Adapted from ASHRAE "Pocket Handbook", 1987

Inadequate or No Circulation

Possible Cause	Recommended Action
Pump running backward (3 phase)	Reverse any two-motor leads
Broken pump coupling	Replace and realign
Improper motor speed	Check motor nameplate wiring and voltage
Pump (or impeller diameter) too small	Check pump selection (impeller diameter) against specified system requirements
Clogged strainer(s)	Inspect and clean screen
System not completely filled	• Check setting of PRV fill valve • Vent terminal units and piping high points
Balance valves or isolating valves improperly set	• Check settings and adjust as required
Air-bound system	• Vent piping and terminal units • Check location of expansion tank connection line relative to pump suction • Review provision for air elimination
Air entrainment	Check pump suction inlet conditions to determine if air is being entrained from suction tanks or sumps
Low available NPSH	• Check NPSH required by pump • Inspect strainers and check pipe sizing and water temperature

Adapted from ASHRAE “Pocket Handbook”, 1987

Cooling Tower Ratings

Temperature °F			Capacity Factor
Hot Water	Cold Water	Wet Bulb	
90	80	70	0.85
92	82	70	1.00
95	85	70	1.24
90	80	72	0.74
92	82	72	0.88
95	85	72	1.12
95	85	74	1.00
95	85	76	0.88
95	85	78	0.75
95	85	80	0.62

Notes:
- Hot water - cold water = range
- Cold water - wet bulb = approach
- The Capacity Factor is a multiplier by which the capacity at any common assumed condition may be found if the rating at some other point is known.
- Factors are based on a Heat Rejection Ratio of 1.25 (15,000 BTU/hr × ton) and gpm/ton flow rate.

Example: at 95-85-80, the capacity is 0.62/0.85 or 0.73 that of the rating at 90-80-70.
- Capacity is reduced as the flow rate per ton is increased.
- If the refrigerant temperature is below 40°F, the heat rejection will be greater than 15,000 BTU/hr × ton.
- Evaporation will cause increasing deposit of solids and fouling of condenser tube unless water is bled off. A bleed of 1% of the circulation rate will result in a concentration of twice the original solids (two concentrations), and 0.5% bleed will result in three concentrations.

Adapted from “Numbers”, Bill Holladay and Cy Otterholm, 1985

Horsepower per Ton

Refrig. Temp. °F	40	20	0	-20	-40
Practical Avg.	0.87	1.20	1.70	2.40	3.20

Notes:
At 100°F Condensing Temperature
Vapor enters Compressor at 65°F

Adapted from “Numbers”, Bill Holladay and Cy Otterholm, 1985

Fouling Factors

Recommended minimum fouling allowances (f) * for water flowing at 3 ft/sec ** or higher:

Liquid:	Fouling Factor
Distilled water	0.0005
Water, closed system	0.0005
Water, open system	0.0010
Inhibited cooling tower	0.0015
Engine jacket	0.0015
Treated boiler feed (212°F)	0.0015
Hard well water	0.0030
Untreated cooling tower	0.0033
Steam:	
Dry, clean and oil free	0.0003
Wet, clean and oil free	0.0005
Exhaust from turbine	0.0010
Brines:	Ferrous Tubes
Methylene chloride	none
Inhibited salts	0.0010
Non-inhibited salts	0.0020
Inhibited glycols	0.0020
Vapors and Gases:	
Refrigerant vapors	none
Solvent vapors	0.0008
Air, (clean) centrifugal compressor	0.0015
Air, reciprocating compressor	0.0030
Other Liquids:	
Organic solvents (clean)	0.0001
Vegetable oils	0.0040
Quenching oils (filtered)	0.0050
Fuel oils	0.0060
Sea water	0.0005

**Insert factor in* $U = \dfrac{1}{\dfrac{1}{h_1} + f_1 + f_2 + \dfrac{1}{h_2}}$

where f_1 and f_2 are the surface fouling factors.
** Lower velocities require higher f values.

Typical Heat Transfer Coefficients

Controlling Fluid and Apparatus	Type of Exchanger	U Free Convection	U Forced Convection
Air: flat plates	Gas to gas *	0.6 -2	2-6
Air: bare pipes	Steam to air *	1-2	2-10
Air: fin coil	Air to water *	1-3	2-10
Air: HW radiator	Water to air *	1-3	2-10
Oil: preheater	Liquid to liquid	5-10	20-50
Air: aftercooler	Comp air to water **	5-10	20-50
Oil: preheater	Steam to liquid	10-30	25-60
Brine: flooded chiller	Brine to, R22		30-90
Brine: flooded chiller	Brine to NH_3		45-100
Brine: double pipe	Brine to NH_3		50-125
Water: double pipe	Water to NH_3		50-150
Water: Baudelot cooler	Water to R22		60-150
Brine: DX chiller	Brine to R22, NH_3		60-140
Brine: DX chiller	E glycol to R22		100-170
Water: DX Baudelot	Water to R22,R502		100-200
Water: DX shell & tube	Water to R22, NH_3		130-190
Water: shell & int finned tube	Water to R22		160-250
Water: shell & tube	Water to water		150-300
Water: shell & tube	Condensing vapor to water		150-800

Liquid velocities 3 ft/sec or higher
* At atmospheric pressure
** At 100 psig
Values shown are for commercially clean equipment.

U factor = BTU/hr - ft² x °F

Evaporative Condenser Ratings

An evaporative condenser rated at a condensing temperature of 100°F and a wet bulb temperature of 70°F will have rating factors under other conditions, as follows.

Cond. Temp., °F	Entering Air Wet Bulb Temperature °F					
	55°	60°	65°	70°	75°	78°
90	0.96	0.86	0.75	0.63	0.50	0.41
95	1.13	1.03	0.91	0.80	0.67	0.59
100	1.32	1.22	1.11	1.00	0.87	0.79
105	1.51	1.41	1.31	1.20	1.08	1.00
110	1.71	1.62	1.52	1.41	1.29	1.22
115	1.93	1.85	1.75	1.65	1.54	1.47
120	2.20	2.11	2.02	1.93	1.81	1.75

Adapted from “Numbers”, Bill Holladay and Cy Otterholm, 1985

Compressor Capacity vs. Refrigerant

Temperature at 100°F Condensing

Refrig. Temp. °F	Heat Rejection Ratio For Sealed Compressors.	Capacity % Based on Temperature			
		50°F	40°F	20°F	0°F
50	1.26	100			
40	1.28	83	100		
30	1.31	69	83		
20	1.35	56	67	100	
10	1.39	45	54	80	
0	1.45	36	43	64	100
-10	1.53	28	34	50	78
-20	1.64	22	26	39	61
-30	1.77	15	18	27	42
-40	1.92	10	12	18	28

Adapted from “Numbers”, Bill Holladay and Cy Otterholm, 1985

The capacity of a typical compressor is reduced as the evaporating temperature is reduced because of increased specific volume (cu ft/lb) of the refrigerant and lower compressor volumetric efficiency. The average 1 HP compressor will have a capacity of nearly 12,000 BTU/hr, 1 ton, at 40°F refrigerant temperature, 100°F condensing temperature. A 10° rise/fall in condensing temperature will reduce/increase capacity about 6%.

Refrigerant Line Capacities R-134a

Tons for 100 ft. - Type L Copper, Suction Lines, Δt = 2°F

Size O.D.	Saturated Suction Temperature °F Δp					Discharge Lines Δt 1°F	Liquid Lines
	0 1.00	10 1.19	20 1.41	30 1.66	40 1.93	0°F	Δt 1°F
1/2	0.14	0.18	0.23	0.29	0.35	0.54	2.79
5/8	0.27	0.34	0.43	0.54	0.66	1.01	5.27
7/8	0.71	0.91	1.14	1.42	1.75	2.67	14.00
1-1/8	1.45	1.84	2.32	2.88	3.54	5.40	28.40
1-3/8	2.53	3.22	4.04	5.02	6.17	9.42	50.00
1-5/8	4.02	5.10	6.39	7.94	9.77	14.90	78.60
2-1/8	8.34	10.60	13.30	16.50	20.20	30.80	163.00
2-5/8	14.80	18.80	23.50	29.10	35.80	54.40	290.00
3-1/8	23.70	30.00	37.50	46.40	57.10	86.70	462.00
3-5/8	35.10	44.60	55.80	69.10	84.80	129.00	688.00
4-1/8	49.60	62.90	78.70	97.40	119.43	181.00	971.00
5-1/8	88.90	113.00	141.00	174.00	213.00		
6-1/8	143.00	181.00	226.00	280.00	342.00		

Tables are based on 105°F condensing temperature.

Refrigerant temperature has little effect on discharge line size. Steel pipe has about the same capacity as Type L. copper 1/8" larger.

Refrigerant Line Capacities R-410a

Tons for 100 ft. - Type L Copper, Suction Lines, Δt = 2°F

Size O.D.	Saturated Suction Temperature, °F Δp					Discharge Lines Δt 1°F	Liquid Lines
	-40 1.27	-20 1.85	0 2.57	20 3.46	40 4.5	0°F	Δt 1°F
1/2	.26	.41	.62	.91	1.27	1.78	6.7
3/4	.59	.93	1.41	2.04	2.86	4.02	15.1
1	1.15	1.83	2.75	4	5.59	7.86	29.5
1 1/4	2.48	3.92	5.9	8.58	12	16.83	63.3
1 1/2	3.79	5.98	9.01	13.06	18.27	26.64	96.6
2	8.8	13.89	20.91	30.32	42.43	59.54	224.2
2 1/2	14.02	22.13	33.29	48.23	67.48	94.70	356.5
3	24.81	39.1	58.81	85.22	119.26	167.36	630
4	50.56	79.68	119.77	173.76	242.63	340.47	1284.6
5	91.27	143.84	216.23	312.97	437.56	614.79	2313.7
6	147.57	232.61	349.71	506.16	707.69	993.09	3741.9

Adapted from Chapter 1.2 ASHRAE Refrigeration Handbook 2014

Refrigerant Line Capacities R-502

Tons for 100 ft. - Type L Copper, Suction Lines, Δt = 2°F

Size O.D.	Saturated Suction Temperature °F Δp					Discharge Lines Δt 1°F	Liquid Lines
	-40 0.92	-20 1.33	0 1.84	20 2.45	40 3.18	0°F	Δt 1°F
1/2	0.08	0.14	0.22	0.33	0.49	0.63	2.4
5/8	0.16	0.27	0.42	0.63	0.91	1.2	4.5
7/8	0.43	0.70	1.1	1.7	2.4	3.1	11.8
1-1/8	0.87	1.4	2.2	3.4	4.8	6.3	24.1
1-3/8	1.5	2.5	3.9	5.8	8.4	10.9	42.0
1-5/8	2.4	4.0	6.2	9.2	13.3	17.2	66.4
2-1/8	5.0	8.2	12.8	19.1	27.5	35.6	138
2-5/8	8.8	14.5	22.6	33.7	48.4	62.8	244
3-1/8	14.1	23.2	36.0	53.7	77.0	99.8	389
3-5/8	21.0	34.4	53.5	79.7	114	148	579
4-1/8	29.7	48.5	75.4	112	161	208	817
5-1/8	53.2	86.7	135	200	287	371	
6-1/8	85.6	140	216	321	461	596	

Adapted from ASHRAE Refrigeration Handbook 2014

Refrigerant Line Capacities R-717

Tons for 100 ft.

Saturated Suction Temperature, °F Δp							Liquid Lines	
IPS	Sch	-40 0.31	-20 0.49	0 0.73	20 1.06	40 1.46	Δp = 3	Δp = 2
3/4	80				2.6	3.8		
1			2.1	3.4	5.2	7.6	13.9	106
1-1/4	40	3.2	5.6	8.9	13.6	19.9	36.5	229*
1-1/2		4.9	8.4	13.4	20.5	29.9	54.8	349*
2		9.5	16.2	26.0	39.6	57.8	106	811
2-1/2		15.3	25.9	41.5	63.2	92.1	168	1293
3		27.1	46.1	73.5	112	163	298	2288
4		55.7	94.2	150	229	333	600	4662
5		101	170	271	412	601	1095	
6		164	276	439	668	972	1771	

Adapted from ASHRAE Refrigeration Handbook 2014

Miscellaneous Formulas

Power for AC Circuits

$$\text{Efficiency} = \frac{746 \times \text{output horsepower}}{\text{Input watts}}$$

$$\text{Three-phase kilowatts} = \frac{\text{volts} \times \text{amperes} \times \text{power factor} \times 1.732}{1000}$$

$$\text{Three-phase volt-amperes} = \text{volts} \times \text{amperes} \times 1.732$$

$$\text{Three-phase amperes} = \frac{746 \times \text{horsepower}}{\text{volts x amperes} \times \text{power factor} \times 1.732}$$

$$\text{Three-phase efficiency} = \frac{746 \times \text{horsepower}}{\text{volts} \times \text{amperes} \times 1.732}$$

$$\text{Single-phase kilowatts} = \frac{\text{volts} \times \text{amperes} \times \text{power factor}}{1000}$$

$$\text{Single-phase amperes} = \frac{746 \times \text{horsepower}}{\text{volts} \times \text{efficiency} \times \text{power factor}}$$

$$\text{Single-phase efficiency} = \frac{746 \times \text{horsepower}}{\text{volts} \times \text{amperes} \times \text{power factor}}$$

$$\text{Single-phase power factor} = \frac{\text{input watts}}{\text{volts} \times \text{amperes}}$$

$$\text{Horsepower (3 Ph)} = \frac{\text{volts} \times \text{amperes} \times 1.732 \times \text{efficiency} \times \text{power factor}}{746}$$

$$\text{Horsepower (1 Ph)} = \frac{\text{volts} \times \text{amperes} \times \text{efficiency} \times \text{power factor}}{746}$$

Power for DC Circuits

$\text{Watts} = \text{volts} \times \text{amperes}\ (W = EI)$

$\text{Amperes} = \text{watts} / \text{volts}\ (I = W/E)$

$$\text{Horsepower} = \frac{\text{volts} \times \text{amperes} \times \text{efficiency}}{746}$$

Speed for AC Machinery

$$\text{Synchronous RPM} = \frac{\text{hertz x 120}}{\text{poles}}$$

$$\text{Percent slip} = \frac{\text{synchronous RPM - full-load RPM}}{\text{synchronous RPM}} \text{ x 100}$$

Ohms Law

Ohms =volts/amperes (R = E/I)
Amperes =volts/ohms (I = E/R)
Volts = amperes × ohms (E = IR)

Motor Application

$$\text{Torque (lb.-ft.)} = \frac{\text{horsepower} \times 5250}{\text{RPM}}$$

$$\text{Horsepower} = \frac{\text{torque} \times \text{RPM}}{5250}$$

Motor Ramp Up Time (Seconds)

$$\text{Seconds} = \frac{WK^2 \times \text{speed change}}{308 \times \text{average acceleration torque}}$$

$$\text{Average acceleration torque} = \frac{[(FLT + BDT) / 2] + BDT + LRT}{3}$$

WK^2 = Inertia of rotor + inertia of load (lb.-ft.^{2})
FLT = Full Load Torque
LRT = Lock Rotor Torque
BDT = Breakdown Torque

$$\text{Load } WK^2 \text{ at motor shaft} = \frac{WK^2 \text{ (load)} \times \text{load RPM}^2}{\text{Motor RPM}^2}$$

$$\text{Shaft stress (PSI)} = \frac{HP \times 321000}{\text{RPM} \times \text{shaft diameter}^3}$$

Change in Resistance Due to Change in Temperature

$$R_c = R_H \frac{K + T_c}{K + T_H}$$

$$R_H = R_c \frac{K + T_H}{K + T_c}$$

K = 234.5 - Copper
= 236 - Aluminum
= 180 - Iron
= 218 - Steel
R_c = Cold resistance (OHMS)
R_H = Hot resistance (OHMS)
T_c = Cold temp (°C)
T_H = Hot temp (°C)

Vibration

$D = 0.318\ V/f$	D = displacement (inches peak-peak)
$V = \pi(f)\ (D)$	V = velocity (inches per second peak)
$A = 0.051(f)^2\ (D)$	A = acceleration (g's peak)
$A = 0.016(f)\ (V)$	f = frequency (cycles per second)

Volume of Liquid in Tank

$Gallons = 5.875 \times D^2 \times H$
D = tank diameter (ft.)
H = height of liquid (ft.)

Pumps

$$BPH = \frac{GPM \times PSI \times specific\ gravity}{1713 \times efficiency\ of\ pump}$$

$$BPH = \frac{GPM \times FT \times specific\ gravity}{3960 \times efficiency\ of\ pump}$$

Fans & Blowers

$$Tip\ speed = \frac{D\ (in) \times RPM \times \pi}{720}$$

$$BPH = \frac{CFM \times PSF}{33000 \times efficiency\ of\ fan}$$

$$BPH = \frac{CFM \times PIW}{6344 \times efficiency\ of\ fan}$$

$$BPH = \frac{CFM \times PSI}{229 \times efficiency\ of\ fan}$$

1 ft. of water = 0.433 PSI

1 PSI = 2.309 ft. of water

specific gravity of water = 1.0

Where:

BHP = brake horsepower
GPM= gallons per minute
FT = feet
PSI = pounds per square inch
PSIG = pounds per square inch gauge
PSF = pounds per square foot
PIW = inches of water gauge

Temperature

$$°F = °C\left(\frac{9}{5}\right)+32$$

$$°C = (°F - 32)\,\frac{5}{9}$$

Conversion Factors

Multiply Length		By		To Obtain
Centimeters	×	0.3937	=	Inches
Fathoms	×	6.0	=	Feet
Feet	×	12.0	=	Inches
Feet	×	0.3048	=	Meters
Inches	×	2.54	=	Centimeters
Kilometers	×	0.6214	=	Miles
Meters	×	3.281	=	Feet
Meters	×	39.37	=	Inches
Meters	×	1.094	=	Yards
Miles	×	5280.0	=	Feet
Miles	×	1.609	=	Kilometers
Rods	×	5.5	=	Yards
Yards	×	0.9144	=	Meters
Multiply Area		**By**		**To Obtain**
Acres	×	4047.0	=	Square meters
Acres	×	0.4047	=	Hectares
Acres	×	43560.0	=	Square feet
Acres	×	4840.0	=	Square yards
Circular mils	×	7.854×10^{-7}	=	Square inches
Circular mils	×	0.7854	=	Square mils
Hectares	×	2.471	=	Acres
Hectares	×	1.076×10^{5}	=	Square feet
Square centimeters	×	0.155	=	Square inches
Square feet	×	144.0	=	Square inches
Square feet	×	0.0929	=	Square meters
Square inches	×	6.452	=	Square centimeters
Square meters	×	1.196	=	Square yards
Square meters	×	2.471×10^{-4}	=	Acres
Square miles	×	640.0	=	Acres
Square mils	×	1.273	=	Circular mils
Square yards	×	0.8361	=	Square meters
Multiply Volume		**By**		**To Obtain**
Cubic feet	×	0.0283	=	Cubic meters
Cubic feet	×	7.481	=	Gallons
Cubic inches	×	0.5541	=	Ounces (fluid)
Cubic meters	×	35.31	=	Cubic feet
Cubic meters	×	1.308	=	Cubic yards
Cubic yards	×	0.7646	=	Cubic meters
Gallons	×	0.1337	=	Cubic feet

Continued on next page

Multiply		By		To Obtain
Gallons	×	3.785	=	Liters
Liters	×	0.2642	=	Gallons
Liters	×	1.057	=	Quarts (liquid)
Ounces (fluid)	×	1.805	=	Cubic inches
Quarts (fluid)	×	0.9463	=	Liters
Multiply Force & Weight		By		To Obtain
Grams	×	0.0353	=	Ounces
Kilograms	×	2.205	=	Pounds
Newtons	×	0.2248	=	Pounds (force)
Ounces	×	28.35	=	Grams
Pounds	×	453.6	=	Grams
Pounds (force)	×	4.448	=	Newton
Tons (short)	×	907.2	=	Kilograms
Tons (short)	×	2000.0	=	Pounds
Multiply Torque		By		To Obtain
Gram-centimeters	×	0.0139	=	Ounce-inches
Newton-meters	×	0.7376	=	Pound-feet
Newton-meters	×	8.851	=	Pound-inches
Ounce-inches	×	71.95	=	Gram-centimeters
Pound-feet	×	1.3558	=	Newton-meters
Pound-inches	×	0.113	=	Newton-meters
Multiply Energy or Work		By		To Obtain
BTU	×	778.2	=	Foot-pounds
BTU	×	252.0	=	Gram-calories
Multiply Power		By		To Obtain
BTU per hour	×	0.293	=	Watts
Horsepower	×	33000.0	=	Foot-pounds per minute
Horsepower	×	550.0	=	Foot-pounds per second
Multiply Power		By		To Obtain
Horsepower	×	746.0	=	Watts
Kilowatts	×	1.341	=	Horsepower
Multiply Plane Angle		By		To Obtain
Degrees	×	0.0175	=	Radians
Minutes	×	0.01667	=	Degrees
Minutes	×	2.9×10^{-4}	=	Radians
Quadrants	×	90.0	=	Degrees
Quadrants	×	1.5708	=	Radians
Radians	×	57.3	=	Degrees
Seconds	×	0.00028	=	Degrees

Continued on next page

Multiply		By		To Obtain
Atmosphere, standard	×	101.35	=	kPa
Bar	×	100	=	kPa
Barrel (42 US gallons petroleum)	×	159	=	L
BTU (International Table)	×	1.055	=	kJ
BTU/ft²	×	11.36	=	kJ/m²
BTU×ft/h×ft²×°F	×	1.731	=	W/(m×K)
BTU×in/h×ft²×°F (thermal conductivity, k)	×	0.1442	=	W/(m×K)
BTU/hr	×	0.2931	=	W
BTU/hr×ft²	×	3.155	=	W/m²
BTU/hr×ft²×°F (heat transfer coefficient, U)	×	5.678	=	W/(m²×K)
BTU/lb	×	2.326	=	kJ/kg
BTU/lb×°F (specific heat, c_p)	×	4.184	=	kJ/(kg×K)
Bushel	×	0.03524	=	m³
Calorie, gram	×	4.187	=	J
Calorie, kilogram (kilocalorie)	×	4.187	=	kJ
Centipoise, dynamic viscosity,m	×	1.00	=	mPa×s
Centistokes, kinematic viscosity, v	×	1.00	=	mm²/s
Dyne/cm²	×	0.100	=	Pa
EDR hot water (150 BTU/hr)	×	44.0	=	W
EDR steam (240 BTU/hr)	×	70.3	=	W
Fuel Cost Comparison at 100% eff.				
Cents per gallon (no. 2 fuel oil)	×	0.0677	=	$/GJ
Cents per gallon (no. 6 fuel oil)	×	0.0632	=	$/GJ
Cents per gallon (propane)	×	0.113	=	$/GJ
Cents per kwh	×	2.78	=	$/GJ
Cents per therm	×	0.0948	=	$/GJ

Continued on next page

Multiply		By		To Obtain
ft of water	×	2.99	=	kPa
ft of water per 100 ft of pipe	×	0.0981	=	= kPa/m
ft^2	×	0.09290	=	m^2
ft^2×h×°F/Btu (thermal resistance, R)	×	0.176	=	m^2×K/W
ft^2/s, kinematic viscosity, v	×	92 900	=	mm^2/s
ft^3	×	28.32	=	L
ft^3	×	0.02832	=	m^3
ft^3/h, cfh	×	7.866	=	mL/s
ft^3/min, CFM	×	0.4719	=	L/s
ft^3/s, cfs	×	28.32	=	L/s
footcandle	×	10.76	=	lx
ft×lb_f (torque or moment)	×	1.36	=	N×m
ft×lb_f (work)	×	1.36	=	J
ft×lb_f / lb (specific energy)	×	2.99	=	J/kg
ft×lb_f / min (power)	×	0.0226	=	W
gallon, US (*231 in^3)	×	3.7854	=	L
gph	×	1.05	=	mL/s
gpm	×	0.0631	=	L/s
gpm/ft^2	×	0.6791	=	L/(s×m^2)
gpm/ton refrigeration	×	0.0179	=	mL/J
grain (1/7000 lb)	×	0.0648	=	g
gr/gal	×	17.1	=	g/m^3
Horsepower (boiler)	×	9.81	=	kW
Horsepower (550 ft×lb_f/s)	×	0.746	=	kW
Inch	×	25.4	=	mm
in of mercury (60°F)	×	3.377	=	kPa
in of water (60°F)	×	248.8	=	Pa
in/100 ft (thermal expansion)	×	0.833	=	mm/m
in×lb_f (torque or moment)	×	113	=	mN×m
in^2	×	645	=	mm^2

Continued on next page

Multiply		By		To Obtain
in^3 (volume)	×	16.4	=	mL
in^3/min (SCIM)	×	0.273	=	mL/s
in^3 (section modulus)	×	16 400	=	mm^3
in^4 (section moment)	×	416 200	=	mm^4
km/h	×	0.278	=	m/s
kWh	×	*3.60	=	MJ
kW/1000 CFM	×	2.12	=	kJ/m^3
kilopond (kg force)	×	9.81	=	N
kip (1000 lb_f)	×	4.45	=	kN
kip/in^2 (ksi)	×	6.895	=	MPa
Knots	×	1.151	=	mph
Litre	×	*0.001	=	m^3
Micron (mm) of mercury (60°f)	×	133	=	mPa
Mile	×	1.61	=	km
Mile, nautical	×	1.85	=	km
ft/min, FPM	×	0.00508	=	m/s
ft/s, fps	×	0.3048	=	m/s
Mph	×	1.61	=	km/h
Mph	×	0.447	=	m/s
Mph	×	0.8684	=	knots
Millibar	×	*0.100	=	kPa
mm of mercury (60°F)	×	0.133	=	kPa
mm of water (60°F)	×	9.80	=	Pa
Ounce (mass, avoirdupois)	×	28.35	=	g
Ounce (force of thrust)	×	0.278	=	N
Ounce (liquid, U.S.)	×	29.6	=	mL
Ounce (avoirdupois) per gallon	×	7.49	=	kg/m^3
Perm (permeance)	×	57.45	=	ng/(s×m^2×Pa)
Perm inch (permeability)	×	1.46	=	ng/(s×m×Pa)
Pint (liquid, U.S.)	×	473	=	mL
lb (mass)	×	0.4536	=	kg

Continued on next page

Multiply		By		To Obtain
lb (mass)	×	453.6	=	g
lb_f(force or thrust)	×	4.45	=	N
lb/ft (uniform load)	×	1.49	=	kg/m
lb_m/(ft×h) (dynamic viscosity, m)	×	0.413	=	mPa×s
lb_m/(ft×s) (dynamic viscosity, m)	×	1490	=	mPa×s
$lb_f s/ft^2$ (dynamic viscosity, m)	×	47 880	=	mPa×s
lb/min	×	0.00756	=	kg/s
lb/h	×	0.126	=	g/s
lb/h (steam at 212°F)	×	0.284	=	kW
lb_f/ft^2	×	47.9	=	Pa
lb/ft^2	×	4.88	=	kg/m^2
lb/ft^3 (density, p)	×	16.0	=	kg/m^3
lb/gallon	×	120	=	kg/m^3
ppm (by mass)	×	*1.00	=	mg/kg
psi	×	6.895	=	kPa
Quart (liquid, US)	×	0.946	=	L
Square (100 ft^2)	×	9.29	=	m^2
Tablespoon (approx.)	×	15	=	mL
Teaspoon (approx.)	×	5	=	mL
Therm (100,000 Btu)	×	105.5	=	MJ
Ton, short (2000 lb)	×	0.907	=	mg; t (tonne)
Ton, refrigeration (12,000 BTU/hr)	×	3.517	=	kW
Torr (1 mm Hg at 0°C)	×	133	=	Pa
Watt per square foot	×	10.8	=	W/m^2
yd	×	0.9144	=	m
yd^2	×	0.836	=	m^2
yd^3	×	0.7646	=	m^3

Pounds are U.S.
Gallons & Quarts are U.S.

* Conversion factor is exact.

Area & Circumference of Circles

Diameter (inches)	Area (sq. in.)	Area (sq. ft.)	Circumference (feet)
1	0.7854	0.0054	0.2618
2	3.142	0.0218	0.5236
3	7.069	0.0491	0.7854
4	12.57	0.0873	1.047
5	19.63	0.1364	1.309
6	28.27	0.1964	1.571
7	38.48	0.2673	1.833
8	50.27	0.3491	2.094
9	63.62	0.4418	2.356
10	78.54	0.5454	2.618
11	95.03	0.6600	2.880
12	113.1	0.7854	3.142
13	132.7	0.9218	3.403
14	153.9	1.069	3.665
15	176.7	1.227	3.927
16	201.0	1.396	4.189
17	227.0	1.576	4.451
18	254.7	1.767	4.712
19	283.5	1.969	4.974
20	314.2	2.182	5.236
21	346.3	2.405	5.498
22	380.1	2.640	5.760
23	415.5	2.885	6.021
24	452.4	3.142	6.283
25	490.9	3.409	6.545
26	530.9	3.687	6.807
27	572.5	3.976	7.069
28	615.7	4.276	7.330
29	660.5	4.587	7.592
30	706.8	4.909	7.854
31	754.7	5.241	8.116
32	804.2	5.585	8.378
33	855.3	5.940	8.639
34	907.9	6.305	8.901
35	962.1	6.681	9.163
36	1017.8	7.069	9.425
37	1075.2	7.467	9.686
38	1134.1	7.876	9.948

Continued on next page

Diameter (inches)	Area (sq. in.)	Area (sq. ft.)	Circumference (feet)
39	1194.5	8.296	10.210
40	1256.6	8.727	10.47
41	1320.2	9.168	10.73
42	1385.4	9.621	10.99
43	1452.2	10.08	11.26
44	1520.5	10.56	11.52
45	1590.4	11.04	11.78
46	1661.9	11.54	12.04
47	1734.9	12.05	12.30
48	1809.5	12.57	12.57
49	1885.7	13.09	12.83
50	1963.5	13.64	13.09
51	2043	14.19	13.35
52	2124	14.75	13.61
53	2206	15.32	13.88
54	2290	15.90	14.14
55	2376	16.50	14.40
56	2463	17.10	14.66
57	2552	17.72	14.92
58	2642	18.35	15.18
59	2734	18.99	15.45
60	2827	19.63	15.71
61	2922	20.29	15.97
62	3019	20.97	16.23
63	3117	21.65	16.49
64	3217	22.34	16.76
65	3318	23.04	17.02
66	3421	23.76	17.28
67	3526	24.48	17.54
68	3632	25.22	17.80
69	3739	25.97	18.06
70	3848	26.73	18.33
71	3959	27.49	18.59
72	4072	28.27	18.85
73	4185	29.07	19.11
74	4301	29.87	19.37
75	4418	30.68	19.63
76	4536	31.50	19.90

Continued on next page

Area & Circumference of Circles (cont.)

Diameter (inches)	Area (sq. in.)	Area (sq. ft.)	Circumference (feet)
77	4657	32.34	20.16
78	4778	33.18	20.42
79	4902	34.04	20.68
80	5027	34.91	20.94
81	5153	35.78	21.21
82	5281	36.67	21.47
83	5411	37.57	21.73
84	5542	38.48	21.99
85	5675	39.41	22.25
86	5809	40.34	22.51
87	5945	41.28	22.78
88	6082	42.24	23.04
89	6221	43.20	23.30
90	6362	44.18	23.56
91	6504	45.17	23.82
92	6648	46.16	24.09
93	6793	47.17	24.35
94	6940	48.19	24.61
95	7088	49.22	24.87
96	7238	50.27	25.13
97	7390	51.32	25.39
98	7543	52.38	25.66
99	7698	53.46	25.92
100	7855	54.54	26.18

Circle Formulas

$$\textit{Area (square inches)} = \pi \times r(in) = \frac{\pi \times d(in)^2}{4}$$

$$\textit{Area (square feet)} = \frac{\pi \times r(in)^2}{144} = \frac{\pi \times d(in)^2}{576}$$

$$\textit{Circumference (feet)} = \frac{\pi \times d(in)}{12}$$

Where:

d = Diameter

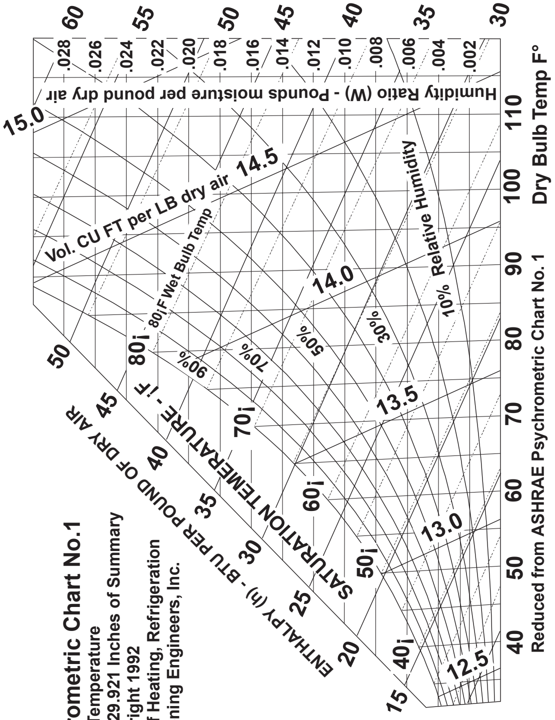
r = Radius

Fractions of an Inch

Decimal and Metric Equivalents

Inches Fraction	Inches Decimal	mm	Inches Fraction	Inches Decimal	mm
1/64	0.01562	0.397	17/64	0.26562	6.747
1/32	0.03125	0.794	9/32	0.28125	7.144
3/64	0.04688	1.191	19/64	0.29688	7.541
1/16	0.06250	1.588	5/16	0.31250	7.938
5/64	0.07812	1.984	21/64	0.32812	8.334
3/32	0.09375	2.381	11/32	0.34375	8.731
7/64	0.10938	2.778	23/64	0.35938	9.128
1/8	0.12500	3.175	3/8	0.37500	9.525
9/64	0.14062	3.572	25/64	0.39062	9.922
5/32	0.15625	3.969	13/32	0.40625	10.319
11/64	0.17188	4.366	27/64	0.42188	10.716
3/16	0.18750	4.763	7/16	0.43750	11.113
13/64	0.20312	5.159	29/64	0.45312	11.509
7/32	0.21875	5.556	15/32	0.46875	11.906
15/64	0.23438	5.953	31/64	0.48438	12.303
1/4	0.25000	6.350	1/2	0.50000	12.700
33/64	0.51562	13.097	49/64	0.76562	19.447
17/32	0.53125	13.494	25/32	0.78125	19.844
35/64	0.54688	13.891	51/64	0.79688	20.241
9/16	0.56250	14.288	13/16	0.81250	20.638
37/64	0.57812	14.684	53/64	0.82812	21.034
19/32	0.59375	15.081	27.32	0.84375	21.431
39.64	0.60938	15.478	55/64	0.85938	21.828
5/8	0.62500	15.875	7/8	0.87500	22.225
41/64	0.64062	16.272	57/64	0.89062	22.622
21/32	0.65625	16.669	29/32	0.90625	23.019
43/64	0.67188	17.066	59/64	0.92188	23.416
11/16	0.68750	17.463	15/16	0.93750	23.813
45/64	0.70312	17.859	61/64	0.95312	24.209
23/32	0.71875	18.256	31/32	0.96875	24.606
47/64	0.73438	18.653	63/64	0.98438	25.004
3/4	0.75000	19.050	1/1	1.00000	25.400

Pyschrometric Chart



ASHRAE Psychrometric Chart No.1
Normal Temperature
Barometric Pressure: 29.921 Inches of Summary
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Index

F

Index

Q

R

S

Notes

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