fanfacts

NORTHERN BLOWER®

Northern Blower Inc. is one of the most progressive industrial fan companies in North America. In all that we do we are committed to the construction of an excellent product and the provision of outstanding customer service.

Northern Blower quality is a tradition. From our first day we have devoted our best efforts to the production of high grade fan equipment. Every day we strive to improve.

Our sales representatives are located coast-to-coast across the continent. Backed by the factory sales team, Northern Blower representatives are ready to provide product information and application advice whenever you need it.

In our desire to enhance customer service, we have published the *fanfacts* manual. It is designed to provide quick access to a variety of fan related information. While we hope that Northern Blower fans will always be your equipment of choice, we have made this manual quite generic in nature so that it will be of maximum benefit to all users. We are pleased to present it to you, and we hope to work with you now and in the years to come.

NORTHERN BLOWER®

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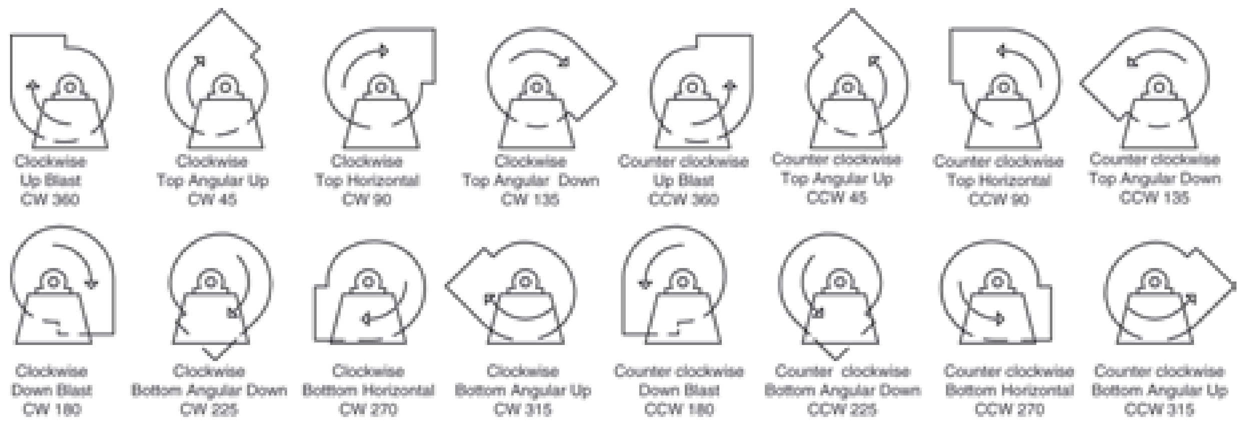
Fan Geometry

Designations for Rotation and Discharge of Centrifugal Fans

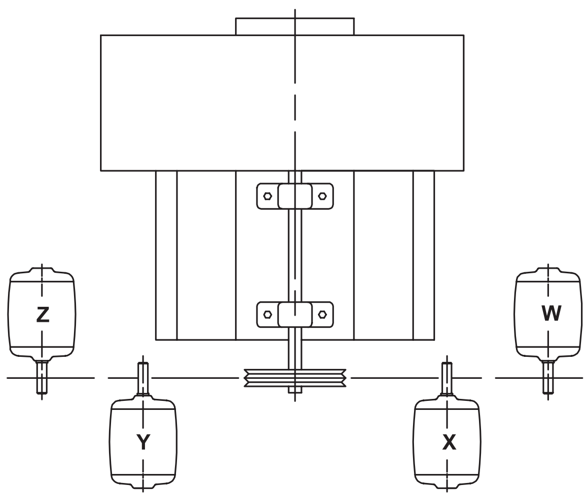


Notes:

1. Direction of rotation is determined from the drive side of the fan.
2. On single inlet fans, the drive side is always considered as the side opposite the fan inlet.
3. On double inlet fans with drives on both sides, the drive side is that with the higher powered drive unit.
4. Direction of discharge is determined in accordance with the diagrams. Angle of discharge is referred to the vertical axis of the fan and designated in degrees from such standard reference axis. Angle of discharge maybe any intermediate angle as required.
5. For a fan inverted for ceiling suspension, or side wall mounting, the direction of rotation and discharge is deter mined when the fan is resting on the floor.

Adapted with permission from AMCA Standards Handbook 99-86

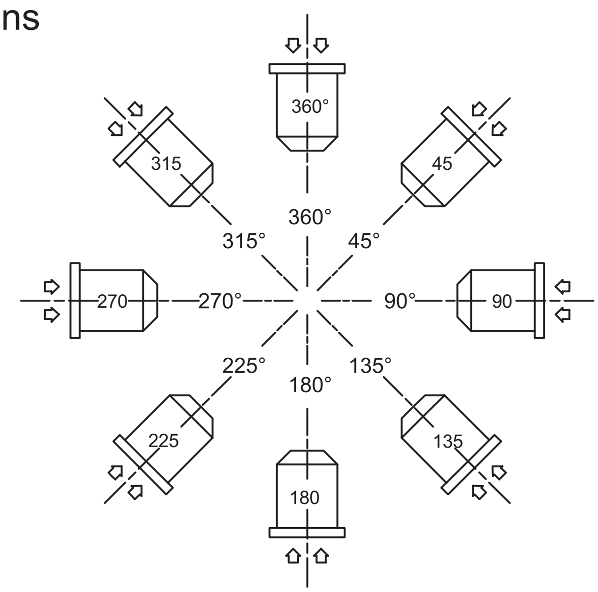
Standard Motor Positions for Centrifugal Fans



1. Motor positions for a centrifugal fan have been given letter designations as shown.
2. These letter designations generally are used only when the motor is mounted separate from the fan proper (i.e. on the ground or on a common fan and motor integral base).

Adapted with permission from AMCA Standards Handbook 99-86

Inlet Box Positions for Centrifugal Fans

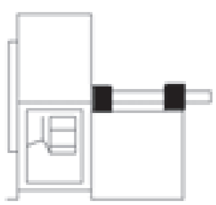


1. The position of the inlet box is determined from the drive side of the fan (as is rotation).
2. On single inlet fans, the drive side is always considered as the side opposite the fan inlet.
3. The angle of the inlet box may be any intermediate angle as required.

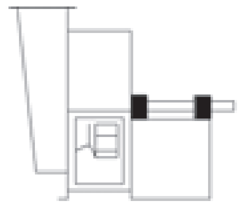
Adapted with permission from AMCA Standards Handbook 99-86

Centrifugal Fan Arrangements

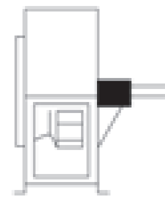
SW - Single Width   DW - Double Width
SI - Single Inlet   DI - Double Inlet



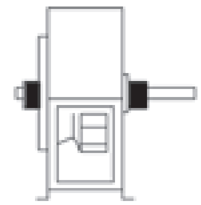
1 SWSI
For belt drive or direct connection. Impeller overhung. Two bearings on base.



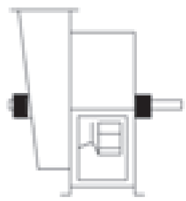
1 SWSI c/w BOX
For belt drive or direct connection. Impeller overhung. Two bearings on base. Inlet box may be self-supporting.



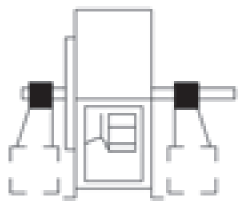
2 SWSI
For belt drive or direct connection. Impeller overhung. Bearings in bracket supported by fan housing.



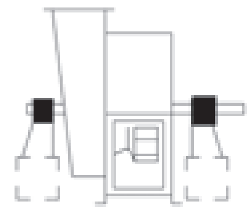
3 SWSI
For belt drive or direct connection. One bearing on each side and supported by fan housing.



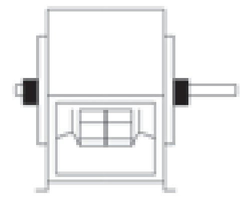
3 SWSI c/w BOX
For belt drive or direct connection. One bearing on each side and supported by fan housing and inlet box. Shaft extending through inlet box.



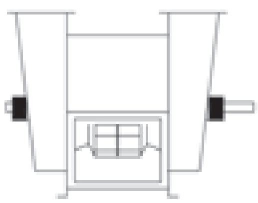
3 SWSI c/w IND. PED.
For belt drive or direct connection. Housing is self-supporting. One bearing on each side supported by independent pedestals.



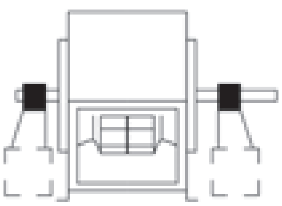
3 SWSI c/w BOX & IND. PED.
For belt drive or direct connection. Housing is self-supporting. One bearing on each side and supported by independent pedestals with shaft extending through inlet box.



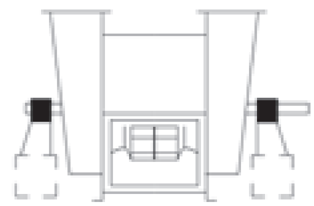
3 DWDI
For belt drive or direct connection. One bearing on each side and supported by fan housing.



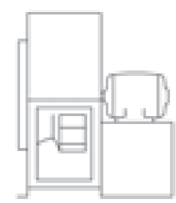
3 DWDI c/w BOXES
For belt drive or direct connection. One bearing on each side and supported by inlet boxes. Shaft extending through inlet boxes.



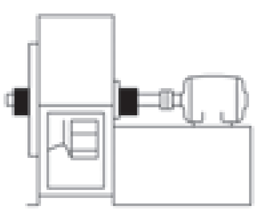
3 DWDI c/w IND. PED.
For belt drive or direct connection. Housing is self-supporting. One bearing on each side and supported by independent pedestals.



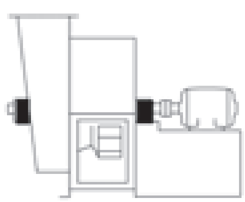
3 DWDI c/w BOXES & IND. PED
For belt drive or direct connection. Housing is self-supporting. One bearing on each side supported by independent pedestals with shaft extending through inlet box.



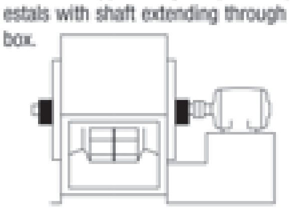
4 SWSI
For direct drive. Impeller overhung on prime mover shaft. No bearings on fan. Prime mover base mounted or integrally directly connected.



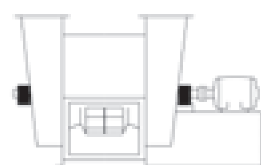
7 SWSI
For belt drive or direct connection. Arrangement 3 plus base for prime mover.



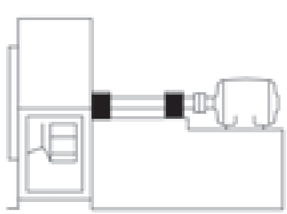
7 SWSI c/w BOX
For belt drive or direct connection. Arrangement 3 plus base for prime mover. Shaft extending through inlet box.



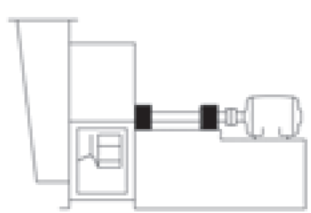
7 DWDI
For belt drive or direct connection. Arrangement 3 plus base for prime mover.



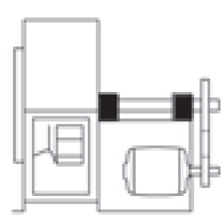
7 DWDI c/w BOXES
For belt drive or direct connection. Arrangement 3 plus base for prime mover. Shaft extending through inlet box.



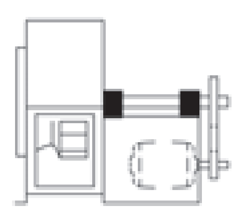
8 SWSI
For belt drive or direct connection. Arrangement 1 plus extended base for prime mover.



8 SWSI c/w BOX
For belt drive or direct connection. Arrangement 1 plus extended base for prime mover.



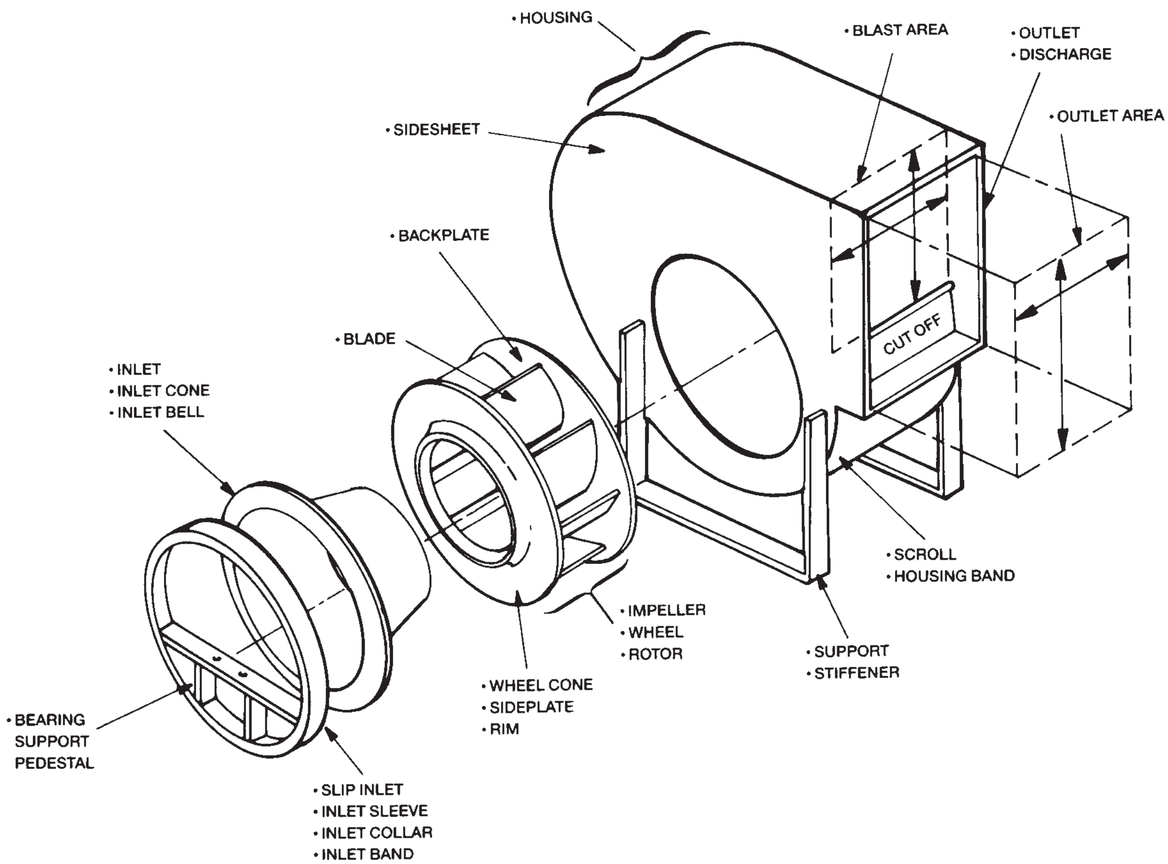
9 SWSI
For belt drive. Impeller overhung, two bearings, with prime mover outside base.



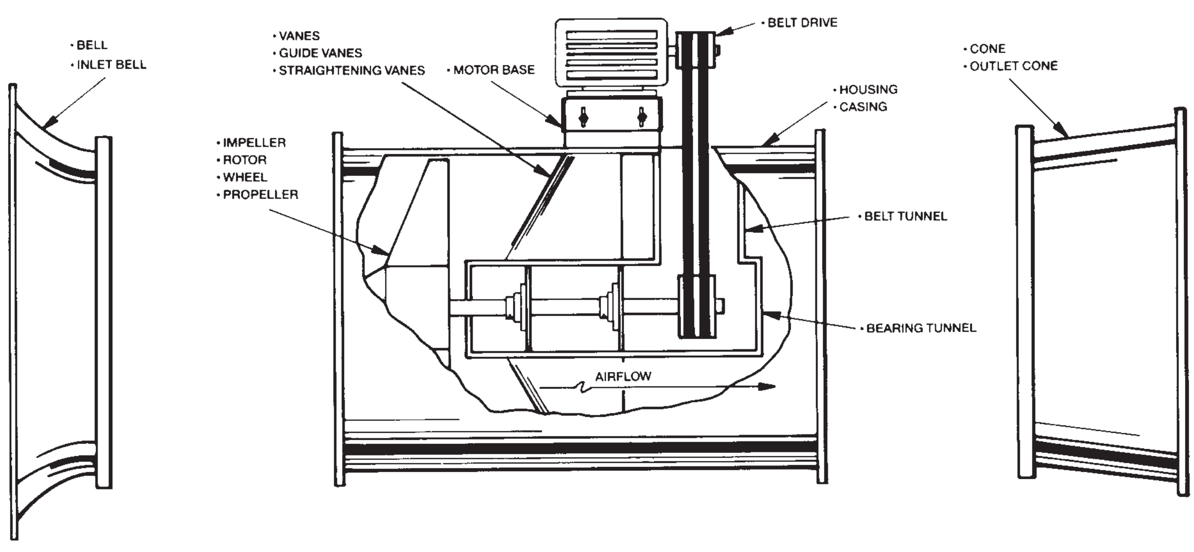
10 SWSI
For belt drive. Impeller overhung, two bearings, with prime mover inside base.

Adapted with permission from AMCA Standards Handbook 99-86

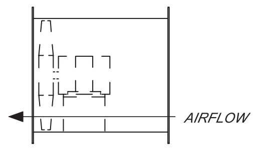
Centrifugal Fan Parts



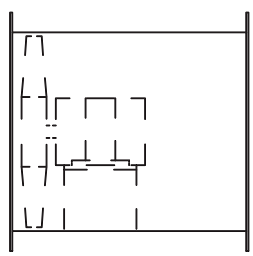
Axial Fan Parts



Axial Fan Arrangements

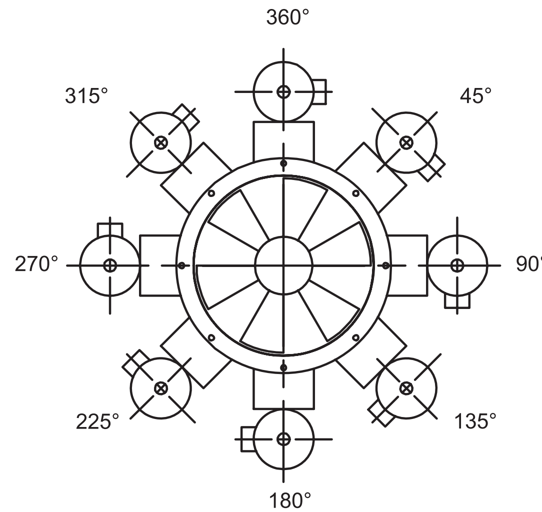


ARRANGEMENT 9



ARRANGEMENT 4

Axial Fan Motor Positions

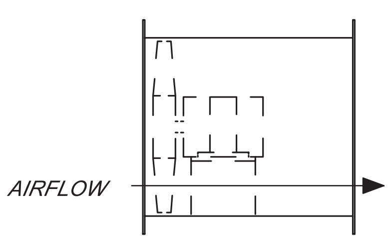


Arrangement 9

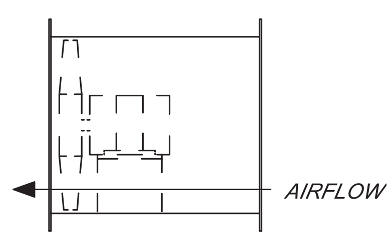
1. The motor can be mounted in one of the positions shown above.
2. The position of the motor is determined from the discharge end of the fan.

Axial Fan Airflow

Airflow through axial fans is designated in two ways. Note: *Not all axial fans are available in both airflow arrangements.*



1) TBOM: through blades, over motor



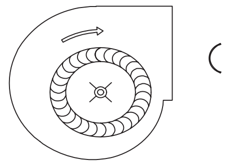
2) OMTB: over motor, through blades

Centrifugal Fan Types

DESCRIPTION | ILLUSTRATION | APPLICATIONS

Forward Curved

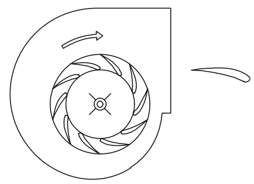
Less efficient than airfoil and backward inclined. Requires the lowest speed of any centrifugal to move a given amount of air. Blades are curved forward in the direction of rotation.



Used for low-pressure heating, ventilating, and air-conditioning systems, ranging from room air-conditioners to residential furnaces.

Airfoil

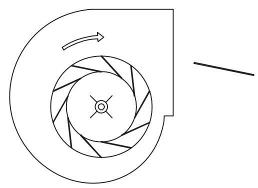
A centrifugal fan type developed to provide high efficiency. Its name is derived from the “airfoil” shape of its blade.



Used on large heating, ventilating, air-conditioning and clean air industrial systems where energy savings are of prime importance.

Backward Inclined

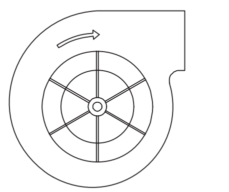
Slightly less efficient than the airfoil. The blades are flat and of single thickness.



Used on large heating, ventilating, air-conditioning and industrial systems where the blade may be subjected to corrosive or erosive environments.

Radial Blade

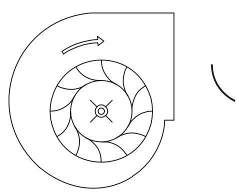
Generally the least efficient of the centrifugal fans. The blades are “radial” to the fan shaft.



For material handling and moderate to high pressure industrial applications.

Radial Tip

More efficient than the radial blade. The blades are radial to the fan shaft at the outer extremity of the impeller, but gradually slope towards the direction of wheel rotation.



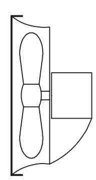
Designed for wear resistance in mildly erosive airstreams.

Inline & Axial Fan Types

DESCRIPTION | ILLUSTRATION | APPLICATIONS

Panel

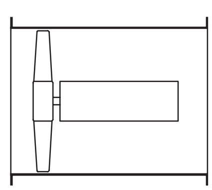
One of the most basic fan designs.



For low-pressure, high-volume applications. Often used for ventilation through a wall.

Tubeaxial

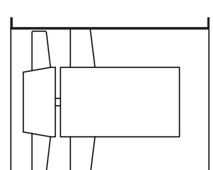
More efficient than the panel fan. Cylindrical housing fits closely to the outside diameter of the wheel.



For low pressure ducted heating, ventilating and air-conditioning systems. Also used in some low pressure industrial applications.

Vaneaxial

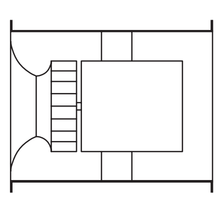
Highest efficiency axial fan. Cylindrical housing fits closely to the outside diameter of the blade tips. The straightening vanes allow for greater efficiency and pressure capacity.



For heating, ventilating, and air-conditioning systems. Good where straight flow and efficiency are required from an axial fan.

Inline Centrifugal

Cylindrical housing is similar to the vaneaxial. Wheel is generally an airfoil or backward inclined type. In this case, the housing does not fit close to the outside diameter of the wheel.



Mostly used for low and medium pressure systems in heating, ventilating, air-conditioning or industrial applications, when a cylindrical housing is geometrically more convenient than a centrifugal configuration.

Centrifugal Fan Class

AMCA Standard 99-2408-69-R83 categorizes centrifugal fans into three performance classes (Class I, II, and III) based on certain minimum operating criteria. A Class I fan offered by any particular manufacturer has a lower allowable minimum operating range than its Class II counterpart. As a result, it is often possible to construct a Class I fan with less mechanical design strength and with less expense than a Class II fan. The same concept applies to a Class II fan versus a Class III fan. Thus, the end result of the AMCA classification system is to allow for a less expensive fan to be constructed for low speed, low pressure applications.

Figure 1 is a reproduction of one chart from AMCA Standard 99-2408-69-R83. It specifies the operating limits of single width centrifugal fan classes (curves for other types of centrifugal fans are available as well). Note that the limits in this chart apply to fans handling air at 70°F and 29.92 inches Hg barometric pressure. When a high temperature application is required, the fan manufacturer should be consulted as to appropriate fan construction.

Example

Q. Given a performance level of $5^1/_2$" SP at 3200 FPM, which fan class is appropriate?

A. This operating point lies well within the boundaries of a Class II fan, and it is appropriate to specify a Class II fan for these conditions.

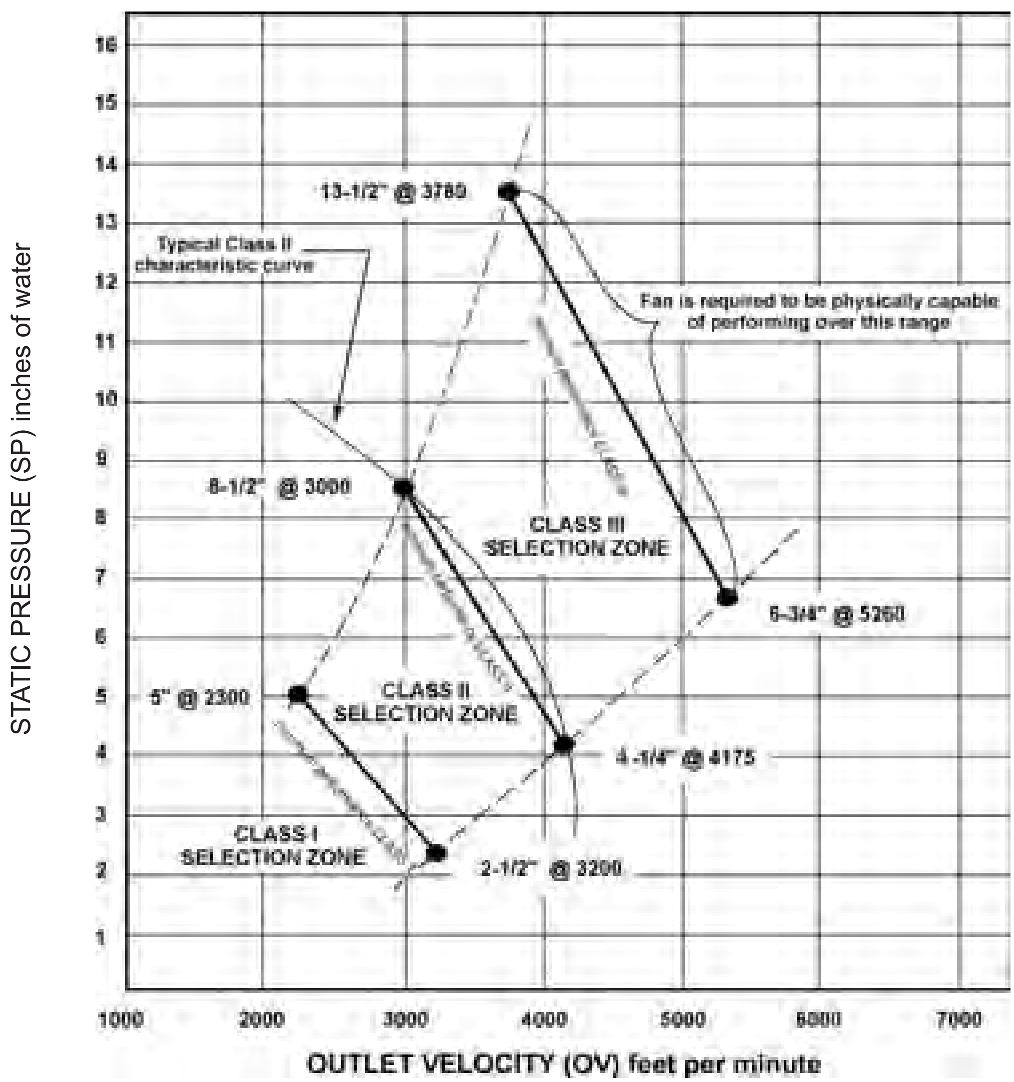


Figure 1 Operating limits for single width centrifugal fans - ventilating airfoils & backwardly inclined.
(Adapted with permission from AMCA Standard 99-2408-69-R83, page 1 of 5)

The Fan Curve

The fan curve is a graphic presentation of fan performance. It is one of the most useful tools available during the fan selection process. While multi-rating tables are convenient (see page 12), performance curves offer additional information such as - how much reserve pressure head exists between the design pressure and the peak static pressure, the maximum power the fan might draw, and the efficiency of operation.

Fan curves are based on laboratory test data and are sometimes referred to as “test curves”. A typical test curve will often define the performance parameters for a specific design and size of fan, operating at a given speed, moving a gas of a given density. Such a curve is illustrated in Figure 2. Inspection of this graph will show that it is actually composed of four separate curves:

- Static Pressure vs. Volume Curve: This plot is the one often referred to as the “fan curve” or “characteristic curve” because it defines all the possible pressure-volume combinations the fan is capable of producing given stated conditions (i.e. rpm and gas density). Note that this curve has two regions - one marked by dashed lines and the other by a solid line. Fans must be selected so that the design point is located on a solid portion of the curve, preferably in an area of high operating efficiency. Operation on the dashed portion of the curve should be avoided as it is a zone of potentially unstable performance. For this reason it is wise to allow some reserve between the peak static pressure and the design pressure to compensate for a higher resistance to flow than anticipated by the design calculation.

- Static Efficiency vs. Volume Curve: In most instances it is desirable to have a fan perform as close to its peak efficiency as possible. The static efficiency vs. volume curve illustrates the efficiency of fan performance at a glance.

- Power vs. Volume Curve: This plot illustrates the power draw of the fan for any point on the characteristic curve.

- System Curve: The system curve defines the volume flow rate versus pressure characteristics of the system in which a fan will be installed. For most applications, the volume flow rate to pressure relationship is governed by the following equation, often called the “duct law”:

$$\frac{P_1}{P_2} = \left(\frac{CFM_1}{CFM_2} \right)^2$$

  Once the system designer has determined the system pressure loss (P) for one flow rate (CFM), it is very easy to calculate the corresponding pressure loss for any other flow rate using this “law”. The system curve is not included on the performance curve when it is issued from the fan manufacturer and its determination is left to the system designer.

At this juncture it is prudent to reiterate that a fan running at a particular speed can have an infinite number of operating points all along its characteristic curve. The fan will interact with the system to produce an operating point at the intersection of the system curve and the fan curve. Note that it is the system in which the fan is installed that will determine the operating point on the fan curve. Thus it is vitally important that the system designer accurately determine the system losses in order to en-sure that the *actual* air flow rate is as close as possible to the *design* air flow rate.

NORTHERN BLOWER®

FAN SERIAL No.:
DRAWN:
SALES OFFICE:
P.O. No.:

CUSTOMER:

PROJECT: D/5010, S/4450, 100% SISW

ENGINEER:

FAN- DESCRIPTION: Design No. 5010 Size 4450 100% SISW Centrifugal

SERVICE: 35000 (CFM), 8 (“W.G.) SP, 56.97 (BHP)
CONDITIONS: Density: 0.075 (lbm/cu.ft), Speed: 1201. (RPM)

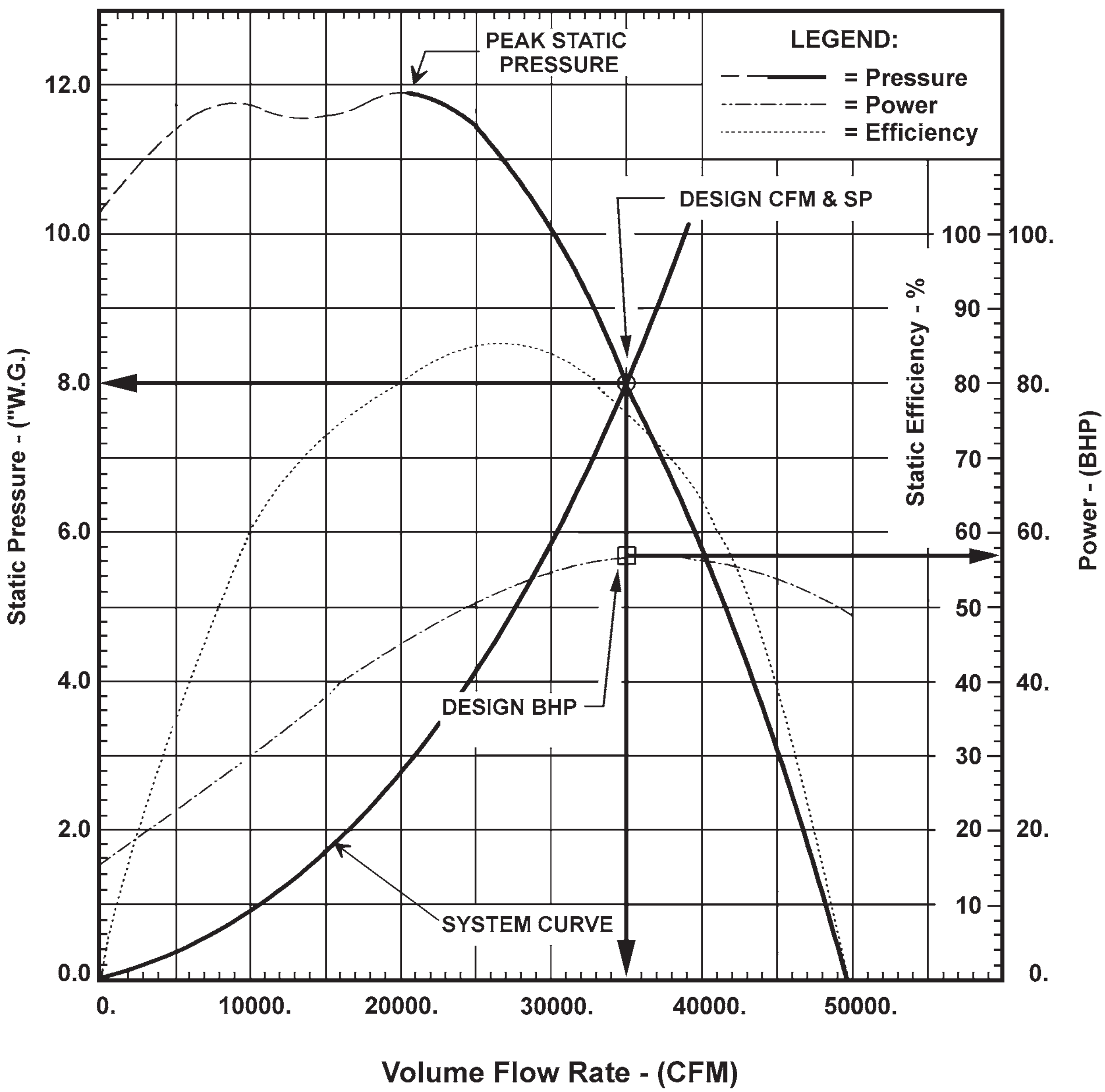


Figure 2 Fan Curve for a Northern Blower Design 5010, Size 4450, running at 1201 RPM

Fan Rating Tables

Fan selection is usually accomplished using performance data published in multi-rating tables. These tables generally show the capacity and pressure range for a given size of fan of a particular design. An example of such a table is illustrated in Figure 3.

From this table it is possible to determine the fan speed (RPM) and power draw (BHP) for various capacity (CFM), outlet velocity (FPM), and pressure (SP) combinations. Some interpolation may be required to accomplish this task.

While a multi-rating table is a very convenient tool, the system designer should always bear in mind the following points:

- Multi-rating tables are published based on data recorded in a laboratory test situation under ideal conditions. The ratings do not account for blockages of the inlet or outlet of the fan by accessories such as screens, guards, or dampers. Appropriate pressure drop corrections should be taken into account whenever these items exist.

- Most tables assume that the gas being handled by the fan is standard air with a density of 0.075 lb_m/cu.ft. If the gas has a density other than 0.075 lb_m/cu.ft., appropriate corrections must be made. A discussion of these corrections is given in this text under the heading of "Fan Laws".

- Rating tables alone say little about the efficiency of fan performance, and do not detail some important selection nuances. For this reason it is useful to examine a fan performance curve (see page 10) before a final fan selection is made.

Note also that any number of fan designs are capable of performing at a given volume-pressure point, but not every design will be suitable for the application at hand. For example, both an airfoil and a radial blade fan are able to produce 10,000 cfm at 10" W.G. static pressure, but the decision as to which fan is best suited for a given job will depend on many other criteria. The relative efficiency of fan operation, the amount of particulate matter in the airstream, and the geometry of the system are just some of the variables which must be taken into account during the selection process. The suitability of different fan designs for particular applications is given some consideration on page 8 of this manual. Your fan manufacturer should be consulted for further detailed information.

Figure 3 A partial section of the multi-rating table for Northern Blower design 5010 single width, size 3650 fan ($36^1/_2$" wheel diameter)

Volume O.V.		2" SP		2-1/2" SP		3" SP		3 1/2" SP		4" SP		4-1/2" SP		5" SP	
CFM	FPM	RPM	BHP	RPM	BHP	RPM	BHP	RPM	BHP	RPM	BHP	RPM	BHP	RPM	BHP
8000	1045	635	3.01												
9000	1175	660	3.44	711	7.24	761	5.09								
10000	1306	686	3.92	736	4.77	783	5.65	827	6.58						
11000	1437	713	4.45	761	5.35	807	6.29	850	7.26	891	8.27	931	9.32		
12000	1567	743	5.05	789	6.04	833	7.00	875	8.03	914	9.07	952	10.14	988	11.26
13000	1698	775	5.72	818	6.73	860	7.79	900	8.85	939	9.95	976	11.09	1012	12.24
14000	1828	810	6.50	849	7.54	888	8.63	927	9.77	965	10.91	1001	12.09	1036	13.29
15000	1959	845	7.34	882	8.43	919	9.56	956	10.75	992	11.97	10.27	13.18	1061	14.44
16000	2090	881	8.23	918	9.46	951	10.58	986	11.81	1020	13.08	1054	14.38	1087	15.68
17000	2220	913	9.20	953	10.51	986	11.75	1018	12.98	1050	14.29	1083	15.64	1114	17.07
18000	2351	953	10.24	988	11.63	1022	12.99	1052	14.28	1082	15.61	1113	17.00	1143	18.42
19000	2481	990	11.36	1024	12.84	1057	14.30	1088	15.75	1116	17.06	1144	18.48	1173	19.94

Fan Laws

Basic Fan Laws

The " Fan Laws " are one of the most fundamental tools used by those involved in fan design, application, and selection. The complete set of equations, which are collectively known as the " fan laws ", are too exhaustive for inclusion in a manual of this type. Instead, we will confine ourselves to the most basic and useful of these equations, which may be used to predict the performance of a fan at speeds and densities other than those listed in a manufacturer's rating catalogue. Please note that the properties of gases are subject to change under certain conditions, and there are consequent limitations to the validity of the basic fan laws. For a complete and detailed explanation of fan laws please consult a suitable fluid mechanics textbook under the heading ' dimensionless parameters '.

KEY TO SYMBOLS	
V - Volume	SP - Static Pressure
P - Power	VP - Velocity Pressure
D - Density	TP - Total Pressure
RPM - Revolutions per Minute	

(1) EFFECT OF VARYING FAN SPEED where the fan size *and* gas density remain constant.

Note from these equations that the volume varies directly with the speed ratio, while the pressure varies by the square, and the power required to drive the fan varies by the cube. The most simple lesson to be learned from these equations is that changes in fan speed are accompanied by relatively greater changes in horsepower. One need only do a sample calculation to realize that a doubling of fan speed will result in an eightfold increase in power consumption. There may be a consequent impact on the size of motor required to drive a fan in such a circumstance.

• Effect on Volume Flow Rate:

$$V_2 = V_1 \times (RPM_2 / RPM_1)$$

• Effect on Pressures:

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

$$VP_2 = VP_1 \times (RPM_2 / RPM_1)^2$$

$$TP_2 = TP_1 \times (RPM_2 / RPM_1)^2$$

• Effect on Power:

$$P_2 = P_1 \times (RPM_2 / RPM_1)^3$$

NOTE: Any system of units may be used for volume, pressure, or power values in the basic fan law equations (i.e. power values may be stated in watts, horsepower, or any other unit system as required).

(2) EFFECT OF VARYING GAS DENSITY where the fan size *and* speed remain constant.

A change in the density of a gas handled by a fan has no impact on the volume flow rate. Only the pressure characteristics and power consumption values vary directly with density changes.

• Effect on Volume Flow Rate:

$$V_2 = V_1$$

• Effect on Pressures:

$$SP_2 = SP_1 \times (D_2 / D_1)$$

$$VP_2 = VP_1 \times (D_2 / D_1)$$

$$TP_2 = TP_1 \times (D_2 / D_1)$$

• Effect on Power:

$$P_2 = P_1 \times (D_2 / D_1)$$

NOTE: The most common influence on density are the effects of temperature and barometric pressure. Almost all fan manufacturers' ratings are published for an air density of 0.075 lb_m / cubic foot, which is the density of dry air at 70° F, and sea level barometric pressure (29.92 in. Hg) .

An Example Calculation Using Basic Fan Laws

The following is an example of the use of the fan laws. Correcting fan performance for a speed change and for density are two of the most common uses of the fan laws outlined on the previous page.

EXAMPLE

Consider an existing size 3650 single width centrifugal fan operating at 12,800 cfm, 2.56 in. W.G. static pressure, 816 rpm, and drawing 6.70 bhp (See Figure 4 below). A change to the system requires a 25% increase in volume flow rate and that the fan be moved outdoors to make room for the new expansion. Once outdoors, the fan will then be handling air at outside temperatures which may drop to -20 degrees Fahrenheit (°F) in winter. Assuming dry air at standard density, calculate the new fan performance requirements and the size of the motor required to drive the fan in its new location.

SOLUTION

The fan laws can be used in the following manner.

The fan laws (listed on page 13) show that the volume flow rate and the speed vary directly, so that a 25% increase in volume flow rate will require a similar increase in the fan speed.

New Fan Speed: 816 x 1.25 = 1020 rpm

Other performance parameters can then be calculated:

Volume Flow Rate: 12,800 x (1020/816) = 16,000 cfm
Static Pressure: 2.56 x $(1020/816)^2$ = 4 in. W.G.
Brakehorsepower: 6.70 x $(1020/816)^3$ = 13.08 bhp

We now know that at standard temperature, a 15 hp motor would be a suitable driver.

As the fan will be required to move cold air, the performance must be corrected for density. The temperature must be expressed as absolute temperature, or degrees Rankine (°R) for these calculations. °R = °F + 460. In this case, standard temperature (T_1 = 70°F) is 70 + 460 = 530°R, and the design temperature (T_2 = -20°F) is -20 + 460 = 440°R. The final relationship required is that density varies indirectly with the absolute temperature ratio as $d_2 = d_1 \times T_1/T_2$. The new performance parameters will then be:

Density (at -20°F): 0.075 x 530/440 = 0.090 lb_m/ft^3
Volume Flow Rate: remains the same.
Static Pressure: 4 x (0.090/0.075) = 4.8 in. W.G.
Brake horsepower: 13.08 x (0.090/0.075) = 15.76 bhp

For the outdoor application, a 20 hp motor would be required to drive the fan.

Volume O.V.		2" SP		2-1/2" SP		3" SP		3 1/2" SP		4" SP		4-1/2" SP		5" SP	
CFM	FPM	RPM	BHP	RPM	BHP	RPM	BHP	RPM	BHP	RPM	BHP	RPM	BHP	RPM	BHP
8000	1045	635	3.01												
9000	1175	660	3.44	711	7.24	761	5.09								
10000	1306	686	3.92	736	4.77	783	5.65	827	6.58						
11000	1437	713	4.45	761	5.35	807	6.29	850	7.26	891	8.27	931	9.32		
12000	1567	743	5.05	789	6.04	833	7.00	875	8.03	914	9.07	952	10.14	988	11.26
13000	1698	775	5.72	818	6.73	860	7.79	900	8.85	939	9.95	976	11.09	1012	12.24
14000	1828	810	6.50	849	7.54	888	8.63	927	9.77	965	10.91	1001	12.09	1036	13.29
15000	1959	845	7.34	882	8.43	919	9.56	956	10.75	992	11.97	10.27	13.18	1061	14.44
16000	2090	881	8.23	918	9.46	951	10.58	986	11.81	1020	13.08	1054	14.38	1087	15.68
17000	2220	913	9.20	953	10.51	986	11.75	1018	12.98	1050	14.29	1083	15.64	1114	17.07
18000	2351	953	10.24	988	11.63	1022	12.99	1052	14.28	1082	15.61	1113	17.00	1143	18.42
19000	2481	990	11.36	1024	12.84	1057	14.30	1088	15.75	1116	17.06	1144	18.48	1173	19.94

from interpolation
12,800 CFM,
2.56" W.G.,
816 RPM,
6.7 BHP
70°F
0.075 lb_m/ft^3

Figure 4 A partial section of the multi-rating table for Northern Blower design 5010 single width, size 3650 fan ($36^1/_2$" wheel diameter)

An Example Using the Air-Density Correction Factors

When the air or gas density changes due to a change in temperature only, the new density can be calculated by the equation $d_2 = d_1 \times T_1/T_2$ as demonstrated in the example on page 14. The fan laws can then be used to calculate the new performance characteristics.

When the air or gas density changes due to a change in altitude or altitude and temperature, the new performance characteristics may be calculated using the Air-Density Correction Factors. This example illustrates how to use the Air-Density Correction Factors Chart in the context of selecting a fan.

Example: Suppose you are selecting a fan for an operating condition of 14000 CFM at 2” SP, 450°F and 4000’ altitude and handling dry air.

Selection: Since fan manufacturers’ catalogued tables are for 70°F and standard density, the design performance must be at these standard conditions to allow for selection from these tables.

The following procedure is often used:

First, the correction factor is selected from the Air-Density Correction Factors Chart (Figure 5) at 450°F and 4000’ altitude. The correction factor is 2.00.

Because CFM is constant for constant RPM, the selection CFM = the operating CFM = 14000 CFM.

Selection SP = Operating SP x Air-Density Correction Factor = 2” x 2.00 = 4” SP at 70 °F and standard density.

It is now known that the fan required must be able to move 14000 CFM at 4” SP when handling air at 70°F and standard density. These parameters are used to select a fan from a performance table. Using Figure 4, you can select a size 3650 fan at 965 RPM, 10.91 BHP, at 70°F and standard density.

With this selection BHP, you can then calculate the operating BHP at 450°F and 4000’ altitude.

Operating BHP = Selection BHP / Air Density Correction Factor = 10.91 / 2.00 = 5.46

Thus, the final operating parameters are: 14000 CFM, 2” SP, 5.46 BHP, and 965 RPM at 450°F and 4000’ altitude.

Note that changes in density have an effect on the SP and HP draw of a fan rotating at a constant RPM.

AIR TEMP °F	Air Density Correction Facotor										
	Elevation (Feet) above Sea Level										
	0	500	1000	1500	2000	2500	3000	3500	4000	4500	5000
-40°	.79	.81	.82	.84	.85	.87	.88	.90	.92	.93	.95
0°	.87	.89	.91	.92	.94	.96	.98	.99	1.01	1.03	1.05
40°	.94	.96	.98	1.00	1.02	1.04	1.06	1.08	1.10	1.12	1.14
70°	1.00	1.02	1.04	1.06	1.08	1.10	1.12	1.14	1.16	1.18	1.20
80°	1.02	1.04	1.06	1.08	1.10	1.12	1.14	1.16	1.19	1.21	1.23
100°	1.06	1.08	1.10	1.12	1.14	1.16	1.19	1.21	1.23	1.25	1.28
120°	1.09	1.12	1.14	1.16	1.18	1.20	1.23	1.25	1.28	1.30	1.32
140°	1.13	1.15	1.18	1.20	1.22	1.25	1.27	1.29	1.32	1.34	1.37
160°	1.17	1.19	1.22	1.24	1.26	1.29	1.31	1.34	1.36	1.39	1.42
180°	1.21	1.23	1.26	1.28	1.30	1.33	1.36	1.38	1.41	1.43	1.46
200°	1.25	1.27	1.29	1.32	1.34	1.37	1.40	1.42	1.45	1.48	1.51
250°	1.34	1.36	1.39	1.42	1.45	1.47	1.50	1.53	1.56	1.59	1.62
300°	1.43	1.46	1.49	1.52	1.55	1.58	1.61	1.64	1.67	1.70	1.74
350°	1.53	1.56	1.59	1.62	1.65	1.68	1.72	1.75	1.78	1.81	1.85
400°	1.62	1.65	1.69	1.72	1.75	1.79	1.82	1.85	1.89	1.93	1.96
450°	1.72	1.75	1.79	1.82	1.86	1.89	1.93	1.96	2.00	2.04	2.08
500°	1.81	1.85	1.88	1.92	1.96	1.99	2.03	2.07	2.11	2.15	2.19
550°	1.91	1.94	1.98	2.02	2.06	2.10	2.14	2.18	2.22	2.26	2.30
600°	2.00	2.04	2.08	2.12	2.16	2.20	2.24	2.29	2.33	2.38	2.42
650°	2.10	2.14	2.18	2.22	2.26	2.31	2.35	2.40	2.44	2.49	2.54
700°	2.19	2.23	2.27	2.32	2.36	2.41	2.46	2.50	2.55	2.60	2.65
750°	2.28	2.33	2.37	2.42	2.47	2.51	2.56	2.61	2.66	2.71	2.76
800°	2.38	2.43	2.48	2.52	2.57	2.62	2.66	2.72	2.76	2.81	2.86

Figure 5 Air-Density Correction Factors Chart ($C_F = .075\ lb_m/ft^3 / D_a$)

Common Terminology

A-Weighting Network: (a.k.a. dBA)

The term "A-Weighted" applies to individual octave band sound levels which have been adjusted to account for the response of the human ear to sound pressure level. It also refers to a single number logarithmic summation of such adjusted octave band values. This number is useful in comparing measured values to endurance limits in the workplace.

Access Door: Door(s) mounted on a fan to provide access to the fan interior for maintenance inspection.

Air Horsepower: (abbr. AHP) The work done by a fan on the gas it moves. This value also may be thought of as the power required to drive the fan if it was 100% efficient, and it is often calculated with the following equation -

$$AHP = (CFM \times TP) / 6356$$

Air: A gaseous mixture of Oxygen, Nitrogen, Hydrogen and other elements in lesser amounts. Standard air is dry air at 70° F and 29.92 inches Hg barometric pressure, and has a density of approximately 0.075 lb_m / cubic foot.

Airfoil: See *page 8*.

AMCA: Air Movement and Control Association Inc. AMCA is a non-profit trade association that publishes standards and test procedures for air handling equipment. The AMCA laboratory uses standard methods to test fans. They also certify ratings of air moving equipment from various manufacturers.

Arrangement: A convention for specifying the drive and bearing location on a fan. For details *see pages 5 and 7* of this manual. Axial Fan: A fan in which the air flows parallel to the shaft, or axially.

Backward Inclined: See *page 8*.

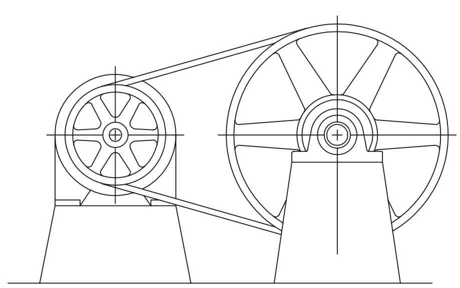
Bearing Life: The life of a rolling bearing is defined as the number of constant speed operating hours (or revolutions) which the bearing is capable of enduring before the first sign of fatigue occurs on the raceways or the rolling elements.

Bearing L_{10} Life: (a.k.a. B_{10} life or nominal life) The number of operating hours (or revolutions) that 90% of a sufficiently large sample of apparently identical bearings can survive when operating under identical conditions at a given constant speed.

Bearing L_{50} Life: (a.k.a. B_{50} life or median life) The number of operating hours (or revolutions) that 50% of a sufficiently large sample of apparently identical bearings can survive when operating under identical conditions at a given constant speed.

$$L_{50} \cong 5L_{10}$$

Belt Drive: A power transfer mechanism composed of "rubber" belts and sheaves or pulleys. Typically one sheave will be mounted on an electric motor and the other on a fan, and the two will be connected by the belts.



belt drive

Belt Guard: A component designed to cover a belt drive mechanism. Its principal function is to guard against human injury.

belt guard

Blast Gate: A sliding or rotating damper composed of only one blade. It is a crude device used to regulate the volume of gas flow by decreasing the open area through which the gas might travel.

Brake Horsepower: A synonym for net horsepower. One (1) horsepower is developed when working at the rate of 550 ft-lb per second. The brake horsepower is generally taken as the power required at the fan shaft.

Centrifugal Fan: Any fan with a scroll shaped housing geometry. The airflow enters the impeller axially and exits radially outward.

Class: A numerical description of the class of construction of a fan. This method of categorization was developed to classify fans based on mini-mum operating characteristics. Three official Air Movement and Control Association (AMCA) classes exist - Class I, II, & III. For all practical purposes (where the fan size is constant) the larger the class numeral, the greater the minimum performance capability and price of a fan.
See *page 9.*

Cooling Wheel: (a.k.a. heat slinger) A heat dissipating device formed in a circular shape with radial fins. It is usually constructed from a highly conductive alloy, such as aluminum, and is attached to the fan shaft. It protects fan bearings from shaft conveyed heat in high temperature applications.

cooling wheel

Cubic Feet per Minute: (abbr.CFM) A description of volume flow rate in English Units.

Damper: A mechanical device which acts to regulate the volume of air transported by a fan. See also Inlet Damper, Outlet Damper, and Variable Inlet Vanes.

damper

Decibel: (abbr. dB) The decibel is a dimensionless unit used for measuring sound power or any other sound property that is proportional to sound power. The decibel is calculated on a logarithmic scale which transforms otherwise unwieldy values into a work-able size.

Density Factor: The ratio of the density of standard air to the actual gas density (typ. 0.075 / D_a). It is a dimensionless factor.

Dry Bulb Temperature: (abbr. T_{db}) The temperature of the atmosphere as measured by a dry temperature sensor (i.e. a thermometer).

Efficiency: The ratio of useful energy delivered by a dynamic system to the energy supplied to it. See Static Efficiency or Total Efficiency.

End Reflection: The phenomenon which occurs when a sound is transmitted from a small space, such as a duct, into a larger area, such as a room. Some of the sound is reflected back into the smaller area.

Entry Loss: A pressure drop caused by mechanical energy losses as air decelerates at the entrance of a duct or a pipe. This loss can be minimized by providing a smooth rounded orifice at the duct opening.

Evasé: An expansion transition located directly on the fan discharge. It is used to convert some of the kinetic energy (velocity pressure) of the air into potential energy (static pressure).

Fan: A device designed to move air. It consists of a rotating impeller and some type of stationary housing which may or may not totally enclose the impeller.

Fan Characteristic Curve: (a.k.a. fan curve) A curve plot of the pressure vs. volume characteristics of a fan running at a given speed handling a gas of a given density. It is usually accompanied by a power consumption curve and may be combined with an efficiency curve. The fan curve is one of the most useful analytical tools available when selecting a fan. For more details see *page 10.*

Flanged Inlet: A round or rectangular facing circumscribing the inlet of a fan. It is usually provided with an arrangement of bolt holes to allow for the mechanical attachment of ducting to the fan inlet.

Flanged Outlet: A round or rectangular facing circumscribing the outlet of a fan. It is usually provided with an arrangement of bolt holes to allow for the mechanical attachment of ducting to the fan outlet.

Forward Curve: See page 8.

Free Field: A free field is defined as a sound field in which the effects of boundaries or surrounding objects are negligible. As a “rule of thumb”, in a free field, the sound pressure will de-cay at a rate of 6 dB for each doubling of distance from the location of the sound source.

Heat Slinger: See cooling wheel

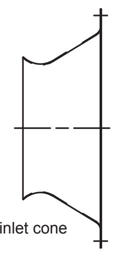
Impeller: The impeller is a rotating device which transmits energy to the air or gas through which it moves. It is often called a wheel or rotor.

Inch of Water: (a.k.a. Water Gauge, abbr. W.G.) The pressure exerted by a column of water one inch high at 68 °F. It is the most common unit of pressure measurement used in the fan industry.

Inlet Box: Inlet boxes can be considered to be a special type of duct elbow which directs air into the inlet(s) of the fan. They are used to turn the airflow and/or protect the fan bearings from the air stream. See diagram, page 5.

Inlet Box Damper: An air volume control device generally mounted on an inlet box. It is composed of several blades mounted on shafts in a frame-work. The position of the blades may be changed by rotating the shafts, and the orifice area through which the air might pass is varied. Like the variable inlet vane it acts to change the shape of the fan curve, so that it essentially causes the fan to perform as though it were smaller in physical size. In most applications, it is less efficient than the variable inlet vane, but more suited to dirty airstreams and temperature extremes.

Inlet Cone: A streamlining device used to reduce entrance losses at the inlet of a fan. It is most often found in clean air applications where high efficiency is a priority.

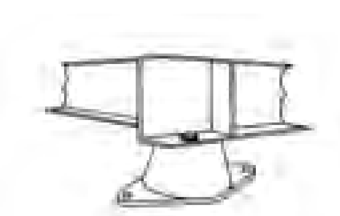


inlet cone

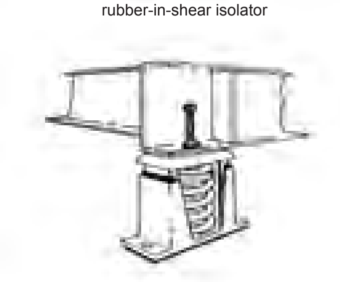
Integral Base: (a.k.a. unitary base)
A frame made from structural steel channels designed to provide a common mounting platform for a fan and its electric motor. If the integral base is supported on vibration isolators it is referred to as a *Vibration Isolation* Base.

integral base

Isolator: An elastic media placed between the fan and its foundation for the purpose of reducing the transmission of vibration. The two most common units found in fan applications are the rubber-in-shear isolator and the coil spring isolator.



rubber-in-shear isolator



coil spring isolator

Near Field: The near field can be considered to extend out from the source of sound a distance equal to the wavelength of the lowest frequency of interest. In this area, the sound pressure levels are the result of the sound radiated from various parts of the source. Sound measurements made within the near field can be misleading as the sound waves generated at one location on the source will tend to interfere with sound waves generated at other locations on the source.

Open Drip Proof: (abbr. ODP) An open electric motor where the ventilation openings are made so that its operation is not impaired when drops of liquid or solid particles strike or enter the enclosure at any angle from 0 to 15 degrees downward from the vertical axis. The motor is cooled by drawing ambient air into the enclosure and circulating it directly over the windings. Open drip proof motors are designed for use in relatively clean, dry, and non-corrosive environments.

Octave Band: The range of sound frequency that may be heard by a human being is conventionally divided into eight octave bands. An octave band ranges from one frequency to twice that frequency.

Outlet Damper: An air volume control device mounted on a fan outlet. It is composed of several blades mounted on shafts in a framework. The position of the blades may be changed by rotating the shafts, and the orifice area through which the air might pass is varied. This mechanism has no effect on the shape of the fan curve, and it functions only to artificially change the system resistance "seen" by the fan. In almost all applications, it is less efficient than the inlet damper or variable inlet vane, but may be a better alternative than a blast gate.

Pressure: Force per unit area. See Static, Total and Velocity Pressure.

Radial Blade: See *page 8.*

Relative Humidity: The ratio of the partial pressure of the water vapour in a mixture to the partial pressure of the water vapour in a saturated mixture at the same temperature.

$$\varphi = (p_v \ / \ p_s)$$

Reverberant Field: In a reverberant field, the sound level is a function of not only the original sound radiated by a source, but also of the reflected sound from surrounding surfaces. Factors which affect the sound pressure level in a room include - the sound power level of the sound source, the size of the room, and the acoustic properties of the reflective surfaces. In a true reverberant field, the sound pressure level does not vary with distance from the sound source.

Revolutions per Minute: (abbr. RPM) The number of times a fan impeller (or shaft) revolves per minute.

Seals:

Labyrinth Seal: An elaborate bearing seal constructed from a dynamic component which rotates with the fan shaft, and a stationary component which is attached to the bearing housing.

Taconite Seal: Taconite is a flint-like rock which contains iron. A "taconite" seal refers to a type of bearing seal which is used to protect internal bearing parts from the invasion of very fine, abrasive dust particles (e.g. cement , potash or "taconite" dust).

Sound Power: (abbr. W) Sound power is a measure of the absolute sound energy that is radiated by a source per unit of time.

Sound Power Level: (abbr. L_W) A logarithmic expression comparing the sound power from a source to a reference power. Sound power level is measured in decibels, generally using a reference power of 1 picowatt or 10^{-12} watts.

$$L_W = 10 \log_{10} (W/10^{-12})$$

Sound Pressure Level: (abbr. L_P) The sound pressure level, measured in decibels, is a comparison of the sound pressure recorded at a particular location to a reference pressure. Distance from the sound source, from other sound sources and from reflective surfaces all affect the sound pressure level. The reference pressure is generally 0.0002 microbar or dynes/ cm^2.

$$L_P = 20 \log_{10} (\text{Pressure[dynes]}/0.0002)$$

Spark Resistant Construction: Various construction techniques utilized by fan manufacturers to reduce the probability of an explosion which might be caused when two ferrous fan parts strike in a volatile gaseous environment.

Static Efficiency: The ratio of the static air power to the fan input power. This can be calculated by multiplying the fan total efficiency by the ratio of fan static pressure to fan total pressure.

Static Pressure: That pressure which exerts itself at a right angle to a surface (e.g. the pressure which tends to burst or collapse a balloon). The energy carried in the air as static pressure is used in part to overcome frictional resistance of the air against the duct surface as well as the resistance offered by all other parts of the system. The fan static pressure is equal to the fan total pressure less the fan velocity pressure, which is the mathematic equivalent of the difference between the static pressure at the fan outlet and the total pressure at the fan inlet.

System: The system consists of the elements through which the air flows on either side of the fan. This could be ductwork, filters, venturi etc.

System Curve: A plot of volume flow rate vs. pressure (static or total) for air flow through the system served by the fan. This curve can be superimposed on a plot of a fan curve to obtain the operating design point of the fan. See graph on *page 11*.

Tip Speed: The peripheral speed of a fan impeller.

Tip Speed (ft/min) = p x dia.(ft.) x rpm

Total Efficiency: The ratio of the total air power to the fan input power. This value is usually expressed as a percentage. Power losses can be due to turbulence, leakage, and friction.

Total Pressure: (abbr. TP) The sum of the static pressure and velocity pressure at any given point in a system.

$$TP = VP + SP$$

Fan total pressure is the sum of the fan velocity pressure and the fan static pressure.

Totally Enclosed Fan Cooled: (abbr. TEFC) A totally enclosed electric motor designed to prevent the free exchange of air between the inside and the outside of the motor, but not designed so that it is air tight. The motor is cooled by means of an integral fan which draws air across the enclosure. TEFC motors are used in outdoor applications, and other abusive environments.

Totally Enclosed Fan Cooled Exposion-Proof: (abbr. TEFC-XP) An explosion proof electric motor de-signed to withstand an explosion of a specified gas or vapour which may occur within it, and designed to pre-vent the ignition of the specified gas or vapour surrounding the machine by sparks flashes or explosions which might occur within the machine casing.

Tubeaxial Fan: See *page 8.*

Vaneaxial Fan: See *page 8.*

Variable Inlet Vane: (a.k.a. Inlet Control Vane, abbr. V.I.V.) An air volume control device mounted on the fan inlet, or integrally constructed into the fan's inlet cone . It is composed of several blades mounted radially on shafts. The position of the blades may be changed by rotating the shafts, and the area through which the air might pass is increased or decreased as required. The variable inlet vane also acts to pre-spin the air as it enters the fan, and, consequently, it is one of the more efficient means of air volume flow control. Its effect is to change the shape of the fan curve. It is particularly suited to clean airstreams.

variable inlet vane

Velocity Pressure: The pressure necessary to maintain the movement of air (kinetic energy). Fan velocity pressure is the pressure which corresponds to the average velocity at the fan outlet.

Vibration Isolation Base: See *Integral Base.*

Wet-Bulb Temperature: (abbr. T_{wb}) The temperature as measured if a thermometer bulb is covered with absorbent material, wet with distilled water and exposed to the atmosphere. Evaporation cools the water and the thermometer bulb to the wet-bulb temperature.

Wheel: See *Impeller.*

Vibration

The presence of vibration is not desirable in any piece of mechanical equipment, and fans are no exception. Excess vibration can cause premature failure of critical parts which might result in high maintenance costs and expensive down-time. As a consequence, it is quite common to find a “vibration clause” written into many fan specifications. Such clauses are generally an attempt to define the allowable vibration limits of operating fan equipment. Nonetheless, vibration remains an oft misunderstood phenomena, so we will devote some effort towards explaining the concepts involved.

The causes of fan vibration may be placed loosely into two general categories:

(i) vibration that is a result of rotating part unbalance.
(ii) vibration that arises from mechanical sources (drive misalignment, improper belt tension, bent shafts and aerodynamic force, to name a few).

Most fan manufacturers dynamically balance their impellers to ensure that their product does not suffer from rotating part unbalance. Care is taken when the fan components are assembled to remove possible mechanical sources of vibration. However, the mass and rigidity of the foundation, duct connections, and all aspects of the final installation process also contribute to the overall vibration level of the fan. For this reason, equipment manufacturers are reluctant to commit themselves to a vibration specification when they have no control over these conditions.

Vibration specifications are written in many different forms, and it is no surprise that some are better than others. Many are quoted in terms of vibration displacement as shown in the following example:

> *The fan will be made to run with a maximum vibration amplitude not to exceed 1.0 mil (0.001 inches) peak-to-peak displacement at the specific operating speed of the fan.*

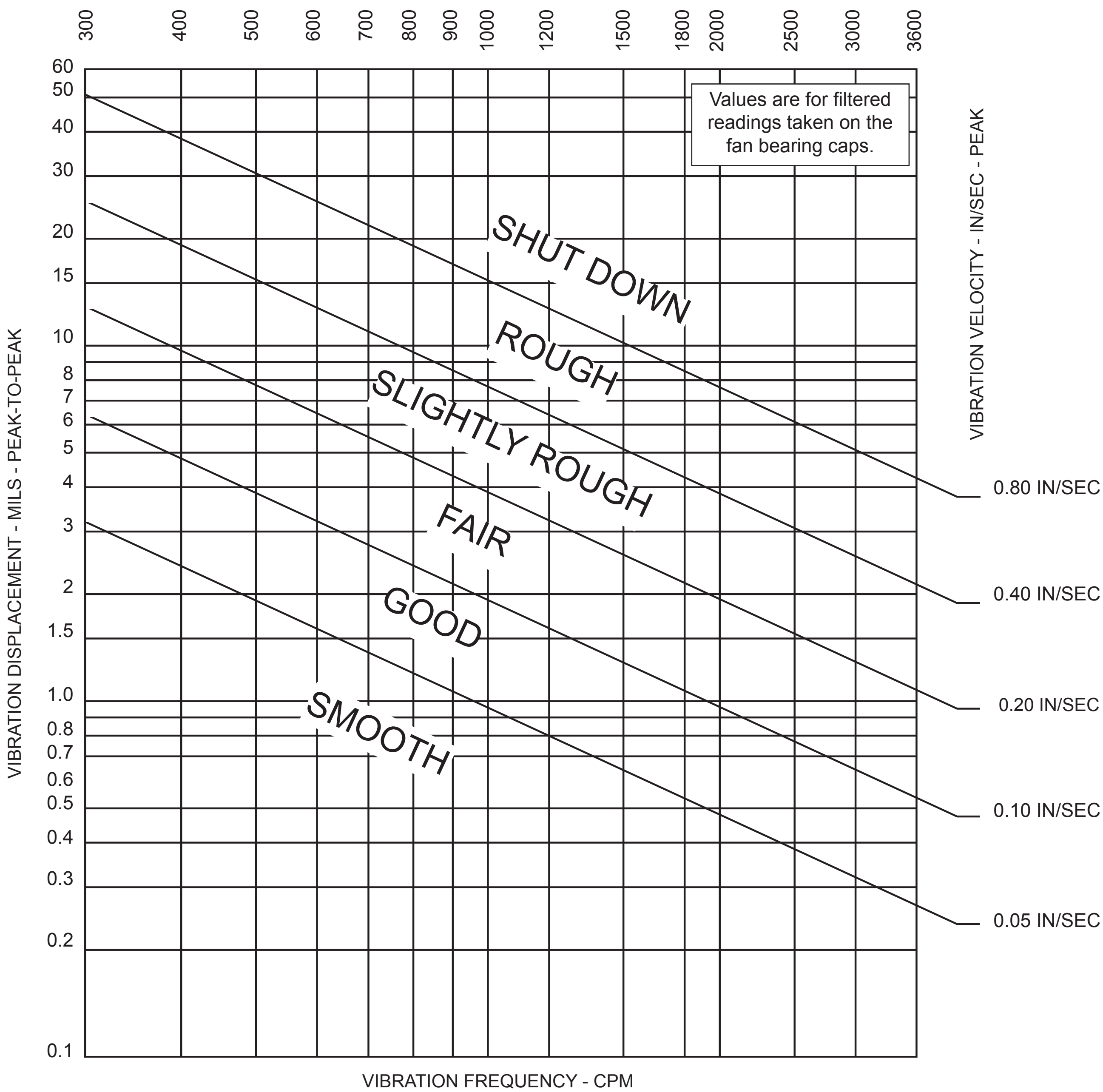
While displacement specifications are well intentioned, they are often inadequate, as acceptable levels of vibration displacement will vary with the driven speed of the fan, as shown in the Fan Vibration Severity Chart (Figure 6). From this chart it is evident that 1 mil of displacement at an operating speed of 1200 rpm is considered a “good” vibration level, while the same displacement at 3600 rpm is deemed to be “slightly rough”. Consequently, the use of a displacement specification is sufficient only when the operating speed of the fan is already known. At the time of specification this is rarely the case.

The most comprehensive method of specifying vibration limits is one which uses vibration velocity to define an acceptable vibration level. This single velocity level de-fines severity for all operating speeds. The following statement may be considered a useful model when drafting vibration specifications:

> *The fan will be made to run with a maximum vibration velocity not to exceed 0.10 inches per second as measured on the fan bearings.*

Once a fan has been installed on the job-site, the determination of its vibration characteristics is generally accomplished with a piece of electronic equipment known as a “vibration analyzer”. If the level of vibration is unacceptable, knowledge of the frequency of vibration allows for diagnosis. It is known that vibration due to rotating part unbalance will occur only at fan speed (i.e. if a fan is driven at 1800 rpm and its wheel is out of balance, the resulting vibration will have a frequency of 1800 cpm), while vibrations due to mechanical sources will exist at different frequencies, including the fan speed.

Fan Vibration Severity Chart



Guidelines for Interpreting the Classifications on the Severity Chart

Smooth: Alignment, balance, and the integrity of the support structure must be near perfect and the vibration from sources other than the fan equipment must be low.

Good: Requires reasonable care on installation, proper foundation, good balance on the rotating components, and good alignment of the running gear.

Fair: Fan equipment can operate in this region, but imperfections are indicated.

Slightly Rough: Requires service. Continued use in this condition will reduce equipment life. Monitor equipment for deterioration.

Rough: Requires service. Dangerous operating conditions for fan equipment. Shut equipment down.

Shut Down: Do not operate fan equipment. Potential for catastrophic failure.

Figure 6 Fan Vibration Severity Chart - to be used as a basic guide only.

Vibration Terms and Definitions

Balance: Unbalance is caused by the non-symmetrical mass distribution about the rotational axis of the rotor. As a result, the heavier side exerts a larger centrifugal force than the lighter side. Balancing therefore consists of redistributing the mass of the rotor so that its mass becomes symmetrically distributed about the shaft or axis of rotation.

Vibration: Vibration is simply the motion of an object back and forth from its position of rest. Consider a weight suspended on a spring as illustrated in Figure 7.

Frequency: Frequency is a measure of the number of cycles for a given interval of time. An 1800 rpm motor has a rotational frequency of 1800 cycles per minute(cpm), or 30 cycles per second (Hz).

Displacement: The total distance traversed by the vibrating part, from one extreme limit to the other extreme limit of travel, is referred to as the “peak-to-peak displacement”. Displacement is generally expressed in mils where 1 mil equals 0.001 inch. The metric unit of measure is the micron and one micron is equal to one-millionth of a meter.

Velocity: Velocity is the rate of change of position with respect to time. Refer to the weight on the spring example above. At the top (and bottom) limit of travel, the velocity is zero as the weight comes to rest before changing direction. The weight then accelerates from its position of rest to a maximum velocity at the neutral axis. Since the velocity is constantly changing, the highest or peak velocity is selected for measurement. Velocity is measured in inches per second. In the metric system, the unit is millimeters per second or microns per second.

$$\text{PEAK VELOCITY (in/sec)} = \frac{52.3 \times \text{FREQUENCY (rpm)} \times \text{DISPLACEMENT (mils)}}{1{,}000{,}000}$$

Acceleration: Acceleration is the rate of change of velocity. An examination of Figure 7 will show that the spring exerts its maximum force on the weight at the bottom extreme limit of travel (At the top limit of travel, gravity acts on the weight). At this point, the acceleration is at its maximum magnitude. As the weight approaches the neutral axis, the velocity increases to a maximum while the acceleration decreases to zero. The unit of measurement is inches/sec^2 or, in the metric system, cm/sec^2.

Filter Out Reading: Most machinery vibration is a complex function consisting of vibrations at many different frequencies. Vibration measuring instruments have the capability of measuring this total vibration function on a “filter out” basis. The “filter out” reading is the sum of all the individual vibrations detected by the vibration measurement instrument.

Filter In Reading: Most sophisticated vibration measurement instruments have a narrow filter band which allows the instrument to measure the vibration at a specific frequency while ignoring the vibration at other frequencies...very similar to tuning into a station on a radio. The filter system in the instrument makes it possible to isolate high vibration and the frequency at which it occurs. Knowing this frequency allows the operator to locate and identify the nature of the problem.

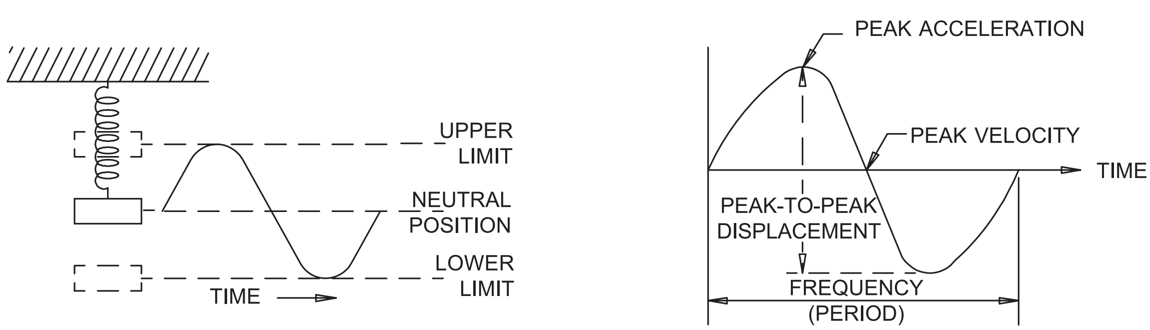


Figure 7 Spring and Weight System

Fan Trouble-Shooting

PROBLEMS	PROBABLE CAUSES
INSUFFICIENT AIR FLOW	. duct elbows near fan inlet or outlet . restricted fan inlet or outlet . impeller rotating in wrong direction . fan speed lower than design . system resistance higher than design . dampers shut . faulty ductwork . dirty or clogged filters and/or coils . inlet or outlet screens clogged
EXCESSIVE AIR FLOW	. system resistance less than design . fan speed too high . filters not in place . registers or grilles not installed . improper damper adjustment
EXCESSIVE HORSEPOWER DRAW	. fan speed higher than design . gas density higher than design . impeller rotating in wrong direction . static pressure less than anticipated . fan size or type not appropriate for application
EXCESSIVE VIBRATION	. accumulated material on impeller . worn or corroded impeller . bent shaft . impeller or sheaves loose on shaft . motor out of balance . impeller out of balance . sheaves eccentric or out of balance . bearing or drive misalignment . mismatched belts . belts too loose or too tight . loose or worn bearings . loose bearing bolts . loose fan mounting bolts . weak or resonant foundation . foundation unlevel . structures not crossbraced . fan operating in unstable system condition
INOPERATIVE FAN	. blown fuse . broken belts . loose sheave . motor too small . wrong voltage

Miscellaneous Formulae and Tables

MOTOR

Torque (in-lbs) $= \dfrac{63025 \times hp}{rpm}$

For 3 phase motors:

BHP output $= \dfrac{E \times I \times ME \times Pf \times 1.73}{746}$

KW input $= \dfrac{E \times I \times Pf \times 1.73}{1000}$

For 1 phase motors:

BHP output $= \dfrac{E \times I \times ME \times Pf}{746}$

KW input $= \dfrac{E \times I \times Pf}{1000}$

FAN

$$\text{Fan BHP} = \frac{CFM \times TP}{6356 \times Eff_T} = \frac{CFM \times SP}{6356 \times Eff_S}$$

$$VP = \left(\frac{CFM}{A \times 4005}\right)^2 = \left(\frac{V}{4005}\right)^2$$

Tip Speed (FPM) $= \pi \times D \times RPM$

$$WR^2_{\text{related to motor}} = WR^2_{fan} \times \left(\frac{RPM_{fan}}{RPM_{motor}}\right)^2 \times S.F.$$

STEEL GAUGES AND WEIGHTS

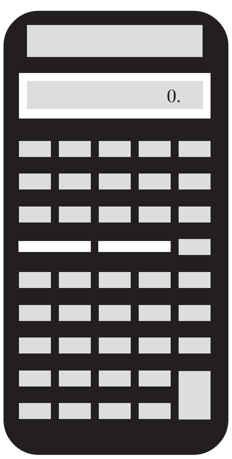
GAUGE	THICKNESS		WEIGHT	
	Inches	mm	lb/Ft²	Kg/m²
000	3/8	9.5250	15.300	74.754
00	11/32	8.7313	14.025	68.525
0	5/16	7.9375	12.750	62.295
1	9/32	7.1450	11.475	56.066
2	1/4	6.3500	10.200	48.836
3	.2391	6.0731	10.000	48.859
4	.2242	5.6947	9.375	45.805
5	.2092	5.3137	8.750	42.752
6	.1943	4.9352	8.125	39.698
7	.1793	4.5542	7.500	36.644
8	.1644	4.1758	6.875	33.591
9	.1495	3.7973	6.250	30.537
10	.1345	3.4163	5.625	27.483
-	1/8	3.1750	5.100	24.917
11	.1196	3.0378	5.000	24.429
12	.1046	2.6568	4.375	21.376
13	.0897	2.2784	3.750	18.322
14	.0747	1.8974	3.125	15.268
15	0.673	1.7094	2.813	13.744
16	.0598	1.5189	2.500	12.215
17	.0538	1.3665	2.250	10.993
18	.0478	1.2141	2.000	9.772
19	.0418	1.0617	1.750	8.550
20	.0359	0.9119	1.500	7.329
21	.0329	.8357	1.375	6.718
22	.0299	.7595	1.250	6.107
23	.0269	.6833	1.125	5.497
24	.0239	.6071	1.000	4.886
25	.0209	.5309	0.875	4.275
26	.0179	.4547	.750	3.664
27	.0164	.4166	.688	3.361

ABOVE ARE MANUFACTURERS' STANDARD GAUGES. WEIGHTS ARE BASED ON DENSITY OF 501.84 LB/FT ^{3}.

MISCELLANEOUS

Area of a Circle, $A = \pi \times (\text{radius})^2$
$= .25 \times \pi \times (\text{diameter})^2$

Circumference of a Circle, $C = 2 \times \pi \times \text{radius}$
$= \pi \times \text{diameter}$



KEY TO SYMBOLS

BHP	brake horsepower
CFM	air volume flow (ft³/min)
D	impeller outside diameter (feet)
E	volts
Eff	fan efficiency (decimal)
I	amps
Kw	kilowatts
ME	motor efficiency (decimal)
Pf	power factor
S.F.	service factor
SP	static pressure (inches W.G.)
TP	total pressure (inches W.G.)
V	velocity (ft/min)
VP	velocity pressure (inches W.G).
WR²	moment of inertia

PROPERTIES OF METALS

Material	Approx. Density (lb/in³)	Approx. Coefficient of Expansion (in/in/°Fx10⁻⁵)	Approx. Melting Point (°F)
Aluminum	0.09751	13.0	1000
Brass	0.30903	10.4	1650
Bronze	0.29456	10.0	1910
Copper	0.32176	9.3	1980
Steel	0.28332	6.3	2370 -2550

METRIC PREFIXES

1 000 000 000 000	$= 10^{12}$	tera	T
1 000 000 000	$= 10^{9}$	giga	G
1 000 000	$= 10^{6}$	mega	M
1 000	$= 10^{3}$	kilo	k
100	$= 10^{2}$	hecto	h
10	$= 10^{1}$	deca	dk
0.1	$= 10^{-1}$	deci	d
0.01	$= 10^{-2}$	centi	c
0.001	$= 10^{-3}$	milli	m
0.000 001	$= 10^{-6}$	micro	m
0.000 000 001	$= 10^{-9}$	nano	n
0.000 000 000 001	$= 10^{-12}$	pico	p

Conversion Factors

ANGLES

MULTIPLY	BY	TO OBTAIN
degrees (angle)	60	min
degrees (angle)	0.01745	radians

AREA

MULTIPLY	BY	TO OBTAIN
in^2	6.452	cm^2
in^2	.006944	ft^2
in^2	635.2	mm^2
ft^2	144	in^2
ft^2	.09290	m^2
m^2	10.76	ft^2

DENSITY

MULTIPLY	BY	TO OBTAIN
lb/ft^3	16.02	kg/m^3
kg/m^3	.06243	lb/ft^3

LENGTH

MULTIPLY	BY	TO OBTAIN
miles	5280	ft
miles	1.609	km
miles	1760	yd
cm	0.3937	in
cm	0.01	m
cm	10	mm
mm	.00328	ft
mm	.03937	in
m	3.281	ft
m	39.37	in
ft	0.3048	m
in	2.540	cm

MOMENT OF INERTIA

MULTIPLY	BY	TO OBTAIN
lb-ft^2	.04214	kg-m^2
kg-m^2	23.730	lb-ft^2

POWER

MULTIPLY	BY	TO OBTAIN
hp	33000	ft-lbs/min
hp	550	ft-lbs/sec
hp	745.7	W
hp	76.04	kg-m/sec
hp	1.014	hpm
hpm	75.00	kg-m/sec
Watts (W)	.00134	hp

PRESSURE

MULTIPLY	BY	TO OBTAIN
in. Hg	0.03342	Atm
in. Hg	0.49115	lb/in^2
in. Hg	13.619	in. WG
in. Hg	3386.4	Pa
in. Hg	25.4	mm Hg
in. WG	0.07343	in. Hg
in. WG	0.03607	lb/in^2
in. WG	248.36	Pa
in. WG	1.8651	mm Hg
lb/in^2	2.036	in. Hg
lb/in^2	27.728	in. WG
lb/in^2	6894.8	Pa
lb/in^2	51.715	mm Hg
oz/in^2	1.732	in. WG
Pa	.00403	in. WG
Pa	.00015	lb/in^2
Pa	.00030	in. Hg
Pa	.00750	mm Hg
mm Hg	.53616	in. WG
mm Hg	133.32	Pa
mm Hg	.01934	lb/in^2
mm Hg	.03937	in. Hg

ROTATING SPEED

MULTIPLY	BY	TO OBTAIN
radians/sec	0.1592	rps
radians/sec	9.549	rpm
rpm	0.1047	rad/sec
rpm	0.01667	rps
rps	6.283	rad/sec
rps	60	rpm
Hertz	1	rps
Hertz	60	rpm

TEMPERATURE

MULTIPLY	BY	TO OBTAIN
°C + 17.78	1.8	°F
°C + 273	1	°K
°F - 32	5/9	°C
°F + 460	1	°R

TORQUE

MULTIPLY	BY	TO OBTAIN
lbf-in	.11298	Nm
Nm	8.8511	lbf-in

VELOCITY

MULTIPLY	BY	TO OBTAIN
ft/min	0.5080	cm/sec
ft/min	.00508	m/sec
ft/min	18.288	m/hr
cm/sec	1.969	ft/min
cm/sec	.03281	ft/sec
m/sec	196.8	ft/min
m/sec	3600	m/hr
m/hr.	.05468	ft/min
m/hr.	.00028	m/sec

VOLUME

MULTIPLY	BY	TO OBTAIN
m^3	10^6	cm^3
m^3	35.31	ft^3
in^3	16.39	cm^3
in^3	5.787×10^{-4}	ft^3
ft^3	1728	in^3
ft^3	0.02832	m^3

VOLUME FLOW

MULTIPLY	BY	TO OBTAIN
CFM	0.4720	l/sec
CFM	.000472	m^3/sec
CFM	.02832	m^3/min
CFM	1.6990	m^3/hr
m^3/sec	2118.9	CFM
m^3/sec	60	m^3/min
m^3/sec	3600	m^3/hr
m^3/sec	1000	l/sec
m^3/min	35.314	CFM
m^3/min	.01667	m^3/sec
m^3/min	60	m^3/hr
m^3/min	16.667	l/sec
m^3/hr	.58858	CFM
m^3/hr	.00028	m^3/sec
m^3/hr	.01667	m^3/min
m^3/hr	.27778	l/sec
l/sec	2.1189	CFM
l/sec	.00100	m^3/sec
l/sec	.06	m^3/min
l/sec	3.6	m^3/hr

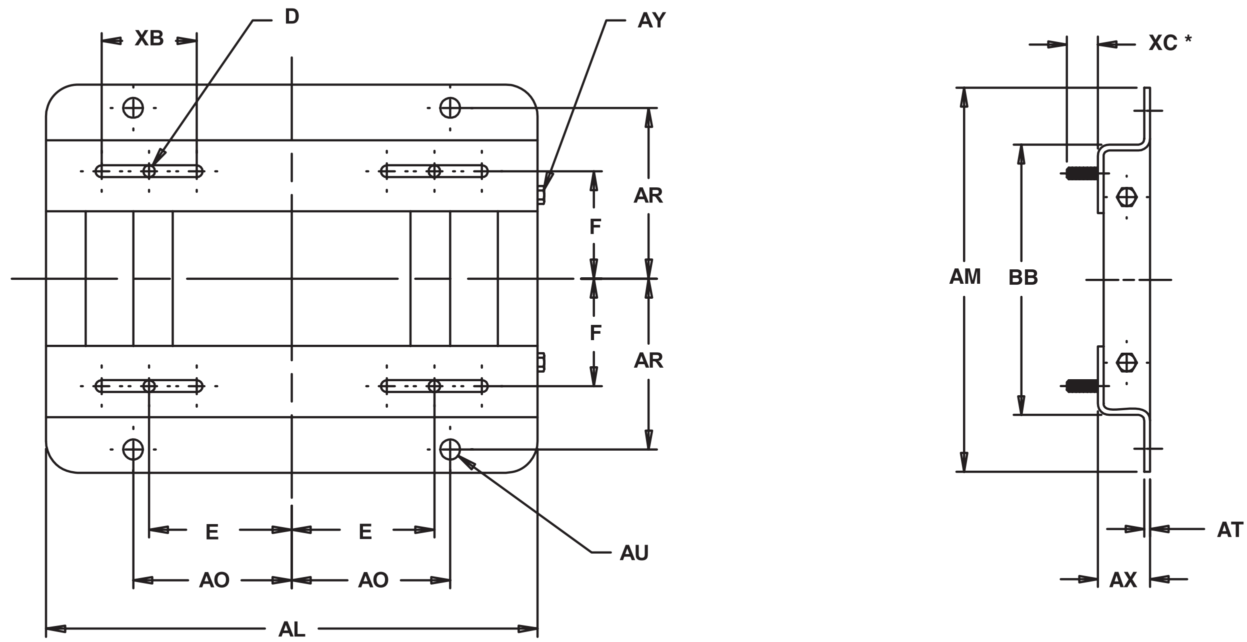
WEIGHT

MULTIPLY	BY	TO OBTAIN
grain	.064798	g
grain	.000143	lb
oz	0.0625	lbs
oz	28.3495	g
g	0.03527	oz
g	.002205	lb
g	15.43	grain
lb	16	oz
lb	453.5924	g
lb	7000	grain

ABBREVIATIONS

Atm	Atmospheres
CFM	cubic feet per minute
cm	centimeter
ft	feet
g	grams
Hg	mercury
hp	horsepower
hpm	metric horsepower
hr	hour
in	inch
kg	kilogram
km	kilometer
l	liters
lb	pounds
lbf	pound-force
m	meter
min	minutes
mm	millimeter
Nm	Newton-meter
oz	ounces
Pa	pascals
rad	Radians
rpm	revolutions per minute
rps	revolutions per second
sec	second
W	Watts
WG	water gauge
yd	yard
°C	degrees Celcius
°F	degrees Fahrenheit
°K	degrees Kelvin
°R	degrees Rankine

Motor Slide Base Dimensions

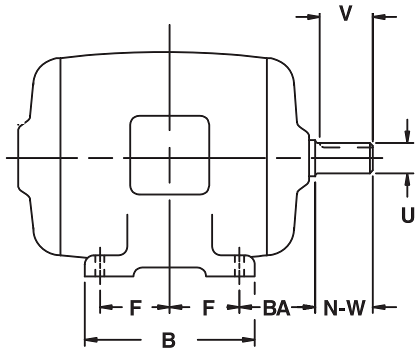


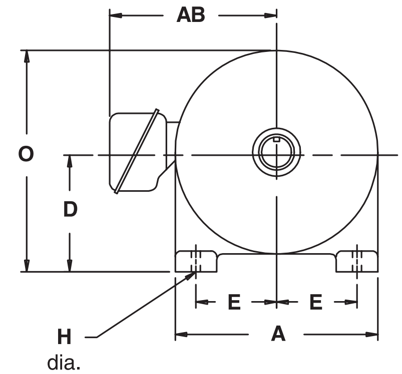
MOTOR** FRAME SIZE	DIMENSIONS* (inches)													WEIGHT (lb.)
	D MTG. BOLT DIA	E	F	AL	AM	AO	AR	AT	AU	AX	AY ADJ. BOLT	BB	XB	
48	5/16	2-1/8	1-3/8	10	6-1/4	3-1/2	2-3/4	.078	3/8	1-1/8	3/8	4-1/4	3	2
56	5/16	2-7/16	1-1/2	10-5/8	6-1/2	3-13/16	2-7/8	.078	3/8	1-1/8	3/8	4-1/2	3	3
143	5/16	2-3/4	2	10-1/2	7-1/2	3-3/4	3-3/8	.119	3/8	1-1/8	3/8	5-1/2	3	5
145	5/16	2-3/4	2-1/2	10-1/2	8-1/2	3-3/4	3-7/8	.119	3/8	1-1/8	3/8	6-1/2	3	6
182	3/8	3-3/4	2-1/4	12-3/4	9-1/2	4-1/2	4-1/4	.134	1/2	1-1/2	1/2	6-1/2	3	9
184	3/8	3-3/4	2-3/4	12-3/4	10-1/2	4-1/2	4-3/4	.134	1/2	1-1/2	1/2	7-1/2	3	9
213	3/8	4-1/4	2-3/4	15	11	5-1/4	4-3/4	.164	1/2	1-3/4	1/2	7-1/2	3-1/2	13
215	3/8	4-1/4	3-1/2	15	12-1/2	5-1/4	5-1/2	.164	1/2	1-3/4	1/2	9	3-1/2	15
254	1/2	5	4-1/8	17-3/4	15-1/8	6-1/4	6-5/8	3/16	5/8	2	5/8	10-3/4	4	17
256	1/2	5	5	17-3/4	16-7/8	6-1/4	7-1/2	3/16	5/8	2	5/8	12-1/2	4	18
284	1/2	5-1/2	4-3/4	19-3/4	16-7/8	7	7-1/2	3/16	5/8	2	5/8	12-1/2	4-1/2	21
286	1/2	5-1/2	5-1/2	19-3/4	18-3/8	7	8-1/4	3/16	5/8	2	5/8	14	4-1/2	22
324	5/8	6-1/4	5-1/4	22-3/4	19-1/4	8	8-1/2	3/16	3/4	2-1/2	3/4	14	5-1/4	31
326	5/8	6-1/4	6	22-3/4	20-3/4	8	9-1/4	3/16	3/4	2-1/2	3/4	15-1/2	5-1/4	32
364	5/8	7	5-5/8	25-1/2	20-1/2	9	9-1/8	1/4	3/4	2-1/2	3/4	15-1/2	6	44
365	5/8	7	6-1/8	25-1/2	21-1/2	9	9-5/8	1/4	3/4	2-1/2	3/4	16-1/2	6	45
404	3/4	8	6-1/8	28-3/4	22-3/8	10	9-7/8	1/4	7/8	3	3/4	16-1/2	7	60
405	3/4	8	6-7/8	28-3/4	23-7/8	10	10-5/8	1/4	7/8	3	3/4	18	7	61
444	3/4	9	7-1/4	31-1/4	24-5/8	11	11	1/4	7/8	3	3/4	19-1/4	7-1/2	67
445	3/4	9	8-1/4	31-1/4	26-5/8	11	12	1/4	7/8	3	3/4	21-1/4	7-1/2	69

1) Northern Blower motor bases for frames 48 & 56 have no adjusting bolt.
2) Northern Blower motor bases for frames 143 to 184 are supplied with one adjusting bolt.
3) Northern Blower motor bases for frames 213 and larger are supplied with two adjusting bolts.

* Some dimensions vary from one base manufacturer to another.
** Dimensions are the same for ‘ U ‘ or ‘ T ‘ frame motor bases.

Motor Dimensions





NEMA FRAME	HP*	DIMENSIONS (inches)											
		A	B	D	E	F	H	N-W	O	U	KEY SQ.	V	BA
48		6-1/2	3-1/2	3	2-1/8	1-3/8	11/32	1-1/2	5-7/8	1/2	flat	1-1/2	2-1/2
56		6-1/2	4	3-1/2	2-7/16	1-1/2	11/32	1-7/8	6-7/8	5/8	3/16	1-7/8	2-3/4
143T	1	7	6	3-1/2	2-3/4	2	11/32	2-1/4	7-1/4	7/8	3/16	2	2-1/4
145T	1-1/2	7	6	3-1/2	2-3/4	2-1/2	11/32	2-1/4	7-1/4	7/8	3/16	2	2-1/4
182	1	7	7	3-1/2	2-3/4	2-1/4	13/32	2-1/4	9	7/8	3/16	2-1/2	2-3/4
184	1-1/2	9	8	4-1/2	3-3/4	2-3/4	13/32	2-1/4	9	7/8	3/16	2-1/2	2-3/4
182T	3	9	6-1/2	4-1/2	3-3/4	2-1/4	13/32	2-3/4	9-3/8	1-1/8	1/4	2-1/2	2-3/4
184T	5	9	7-1/2	4-1/2	3-3/4	2-3/4	13/32	2-3/4	9-3/8	1-1/8	1/4	2-1/2	2-3/4
213	3	10-1/2	7	5-1/4	4-1/4	2-3/4	13/32	3	10-1/4	1-1/8	1/4	3-1/8	3-1/2
215	5	10-1/2	8-1/2	5-1/4	4-1/4	3-1/2	13/32	3	10-1/4	1-1/8	1/4	3-1/8	3-1/2
213T	7-1/2	10-1/2	7-1/2	5-1/4	4-1/4	2-3/4	13/32	3-3/8	11	1-3/8	5/16	3-1/8	3-1/2
215T	10	10-1/2	9	5-1/4	4-1/4	3-1/2	13/32	3-3/8	11	1-3/8	5/16	3-1/8	3-1/2
254U	7-1/2	12-1/2	10	6-1/4	5	4-1/8	17/32	3-3/4	13	1-3/8	5/16	3-3/4	4-1/4
256U	10	12-1/2	12	6-1/4	5	5	17/32	3-3/4	13	1-3/8	5/16	3-3/4	4-1/4
254T	15	12-1/2	10-7/8	6-1/4	5	4-1/8	17/32	4	13-1/2	1-5/8	3/8	3-3/4	4-1/4
256T	20	12-1/2	12-1/2	6-1/4	5	5	17/32	4	13-1/2	1-5/8	3/8	3-3/4	4-1/4
284U	15	14	12-1/2	7	5-1/2	4-3/4	17/32	4-7/8	14-5/8	1-5/8	3/8	4-3/8	4-3/4
286U	20	14	14	7	5-1/2	5-1/2	17/32	4-7/8	14-5/8	1-5/8	3/8	4-3/8	4-3/4
284T	25	14	12-1/2	7	5-1/2	4-3/4	17/32	4-5/8	14-5/8	1-7/8	1/2	4-3/8	4-3/4
286T	30	14	14	7	5-1/2	5-1/2	17/32	4-5/8	14-5/8	1-7/8	1/2	4-3/8	4-3/4
324U	25	16	13-1/2	8	6-1/4	5-1/4	21/32	5-5/8	16-5/8	1-7/8	1/2	5	5-1/4
326U	30	16	15	8	6-1/4	6	21/32	5-5/8	16-5/8	1-7/8	1/2	5	5-1/4
324T	40	16	14	8	6-1/4	5-1/4	21/32	5-1/4	16-5/8	2-1/8	1/2	5	5-1/4
326T	50	16	15-1/2	8	6-1/4	6	21/32	5-1/4	16-5/8	2-1/8	1/2	5	5-1/4
364U	40	18	14-1/2	9	7	5-3/8	21/32	6-3/8	18-3/4	2-1/8	1/2	5-5/8	5-7/8
365U	50	18	15-1/2	9	7	6-1/8	21/32	6-3/8	18-3/4	2-1/8	1/2	5-5/8	5-7/8
364T	60	18	15-1/4	9	7	5-5/8	21/32	5-7/8	18-3/4	2-3/8	5/8	5-5/8	5-7/8
365T	75	18	16-1/4	9	7	6-1/8	21/32	5-7/8	18-3/4	2-3/8	5/8	5-5/8	5-7/8
404U		20	16	10	8	6-1/8	13/16	7-1/8	20-1/4	2-3/8	5/8	7	6-5/8
405U	60	20	17-1/2	10	8	6-7/8	13/16	7-1/8	20-1/4	2-3/8	5/8	7	6-5/8
404T		20	16-1/4	10	8	6-1/8	13/16	7-1/4	21	2-7/8	3/4	7	6-3/8
405T	100	20	17-7/8	10	8	6-7/8	13/16	7-1/4	21	2-7/8	3/4	7	6-3/8
444U	75	22	18-1/2	11	9	7-1/4	13/16	8-5/8	22-1/4	2-7/8	3/4	8-1/4	7-1/2
445U	100	22	20-1/2	11	9	8-1/4	13/16	8-5/8	22-1/4	2-7/8	3/4	8-1/4	7-1/2
444T	125	22	18-1/2	11	9	7-1/4	13/16	8-1/2	23	3-3/8	7/8	8-1/4	7-1/2
445T		22	20-1/2	11	9	8-1/4	13/16	8-1/2	23	3-3/8	7/8	8-1/4	7-1/2

NOTE: all dimensional data shown on this page is approximate, and may vary between models, manufacturers, and types of enclosures. Contact your motor manufacturer when exact dimensions are required.

* Horsepowers listed are for standard TEFC 1800 rpm motors only, and may vary between models, manufacturers, and types of enclosures.

NORTHERN BLOWER®