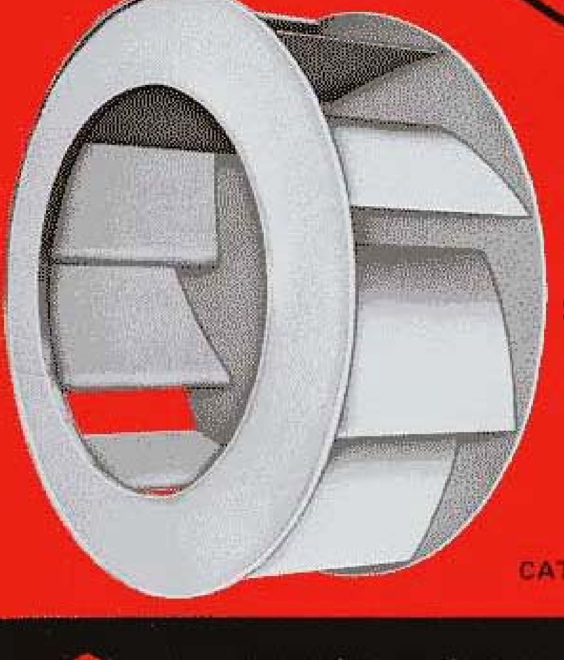
SHELDONS

CENTRIFUGAL FANS

ULTRAFOIL®

AND

UNIFOIL®



SWSI and DWDI
Classes 1, 2 & 3
Sizes 12" to 89"

CAT. No. 355 (Issue 177)

INDEX

SHELDONS ENGINEERING

Sheldons

Sheldons Engineering is a long-standing member of AMCA – "The Air Movement and Control Association" which acts as an independent association for fan manufacturers.

As you may be aware, Sheldons Engineering has manufactured fans for over 100 years. Our product offering includes standard centrifugal and axial fans as well as custom manufactured units to meet our customers' requirements. Our fans are used in applications requiring up to 80" wg static pressure and flows from 500-1,000,000 cfm.

Our web site at www.sheldonsengineering.com contains all of these notes for your reference. We would list our complete product offering, but Sheldons is and has always been a custom manufacturer. Please contact us to discuss your application.

Sheldons Engineering has realized the need for more detailed information to be provided for the selection of fans, particularly the effect of appurtenances and site conditions on the fan performance in the field.

Since site conditions seldom approach laboratory conditions, it is necessary to make some practical allowance for the effect of site conditions on fan ratings. Therefore, in this catalog, Sheldons have attempted to provide the Consulting Engineer and end user with practical guide lines to assist in arriving at a fan selection that is reasonably valid for the particular application, taking into account the many variables that have been found to affect fan performance in the field. Further detailed information on these effects can be found in AMCA Publications 201, 202 and 203.

ENERGY CONSERVATION

System designs today demand more accurate estimates of energy consumption. By considering all the factors affecting fan performance shown in this catalog, and applying all the losses involved, the consulting engineer and end user are provided with all the information to make an accurate determination of fan horsepower, resulting in valuable savings.

The AMCA seal as shown below is a guarantee of performance resulting from a rigorous test program undertaken in accordance with AMCA Standard 210-74. Sheldons modern Research and Development facility, a fully approved AMCA test laboratory, was completely re-equipped to meet the updated test procedures outlined in the 1974 AMCA test code.

AMCA CERTIFIED RATINGS

Because of variation of performance with size, it is not practical to rate all fan sizes from a single fan test. There fore, in order to provide optimum performance, especially in the smaller sizes, Sheldons have tested 5 basic fan sizes, both SWSI and DWDI. These tested fans are then rated up to larger sizes and presented according to the requirements of AMCA Publication 211 for air performance, to provide conservative ratings.

To ensure integrity of ratings, AMCA requires that published fan performance data clearly state the basis from which the ratings were obtained, and any limitations noted. In addition, rigorous check testing of random fan sizes and speeds, are made regularly by AMCA staff to ensure published ratings are being met by production fans.

Sheldons certifies that the Type UNF Unifoil and Type ULF Ultrafoil Centrifugal fans shown herein are licensed to bear the AMCA Seal, The ratings shown are based on tests made in accordance with AMCA Standard 210, and comply with the requirements of the AMCA Certified Ratings Program. The AMCA Certified Ratings Seal applies to air capacities only

AMCA CLASS LIMITS

In accordance with AMCA Standard 2408-69, the class operating limits have been defined at certain outlet velocities and static pressures. Values beyond these limits can be published but the class limits must be capable of being met by the class of the fan. With most fan manufacturers, the upper and lower limits of the class range are met at two slightly different speeds, and in this case, the higher speed will determine the fan design capability. The class limitation line shown on each fan-rating curve has been drawn at this higher speed. It is suggested that when a required performance lies at a speed 5% less than the class limit, then the next higher fan class be chosen. In this way, slight increases in fan speed to suit site conditions will not put the fan beyond the class design limit. In this catalog Sheldons provide standard information in Class 1, 2 and 3 fans, both single width and double width. For larger or heavier duty fans for extra heavy-duty commercial or industrial applications, consult Sheldons nearest sales office.

SOUND

Overall sound power levels in eight octave bands have been calculated for the speed range covered in each fan size, and are shown in a table on each performance curve for the area of maximum efficiency. For other speeds simple interpolation between the given speeds is quite satisfactory.
Sound levels vary slightly with the point of operation and may be approximately compensated for by further addition of the dB values noted in the zones defined by the radial lines on each performance-rating curve.
When calculating sound levels in either inlet or outlet ductwork, deduct 3 dB from the overall sound **power** levels to arrive at sound power levels at the fan inlet or outlet. Sound **pressure** levels must be calculated from information available to the system designer or end user, using room absorption effects. Sound pressure levels in typical equipment rooms, can generally be estimated by Sheldons from their field tests on comparable applications.

SPARK-RESISTANT CONSTRUCTION

For fans handling potentially explosive mixtures, AMCA standard sparkproof construction is available in Arr. 1, Arr. 2, Arr. 4, Arr. 8 and Arr. 9 only. Bearings are not located in the gas stream.
There are three classifications of construction as follows:

Type A - "All parts of the fan in the air or gas being handled shall be made of non-ferrous material." Aluminum wheel, housing and inlet, with monel shaft. All other parts are standard steel construction. (Generally limited to Class 2 and/or 250° F.)

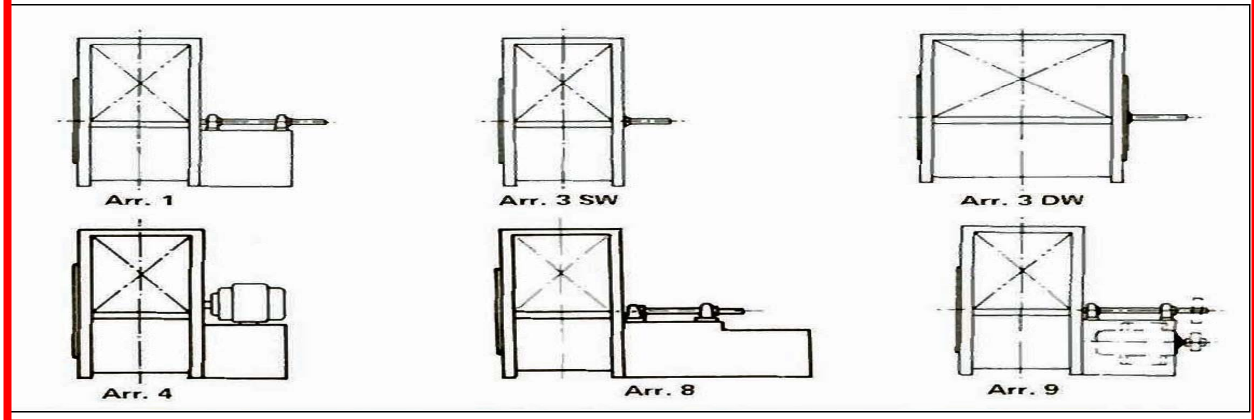
Type B - "Fan shall have an entirely non-ferrous wheel and non-ferrous ring about the opening through which the shaft passes." Aluminum wheel and aluminum disc where the shaft passes through fan case. All other parts are standard
steel construction. (Generally limited to Class 2 and/or 250° F.)

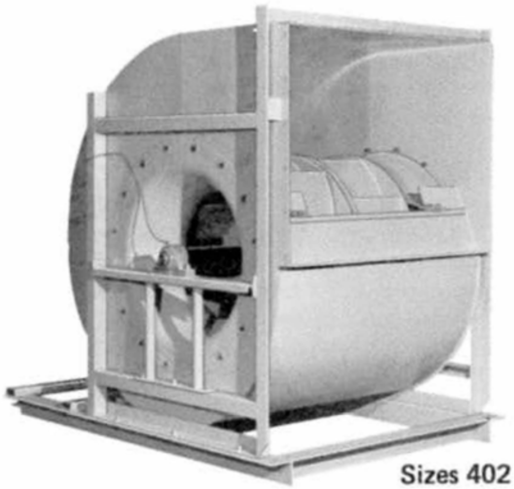
Type C - "Fan shall be so constructed that a shift of the wheel or shaft will not permit two ferrous parts to rub or strike." Aluminum inlet and aluminum disc where the shaft passes through the fan case. All other parts are standard steel construction. (Generally limited to 500° F. operation).

Temperature Limitations

Air or Gas Temp °F	Arr 1 & 9	Arr 3	% of Max Speed Limit
Up to 130°F	**Standard Bearings**	**Standard Bearings**	**100%**
130°-200°F	**Standard Bearings**	**Not Applicable**	**94%**
200°-300°F	**Standard Bearings w/ cooling disc**	**Not Applicable**	**89%**
Above 300°F	**Consult Sheldons Engineering**		

Arrangements





Sizes 402 - 890 Ultrafoil

Sizes 122 - 222 Unifoil
Sizes 245 - 365 Ultrafoil

FEATURES

- **Streamlined Inlet Cone**

Inlet spinnings are designed for the smooth acceleration of airflow into the wheel, with clearances designed to maintain efficiency and reduce noise to a minimum.
On smaller sizes in Classes 1 and 2, an inlet collar is provided for duct connection. On sizes 402 and above, a rectangular flange on the fan case provides connection to ductwork.

- **Bearing Pedestals**

Heavy-duty bearing pedestals of structural steel with extra stiffeners on Class 3 fans, maintain accurate alignment, prevent distortion due to belt pull, and provide minimum resistance to the airflow. For actual effect of bearing in the air stream on the performance of Arr. 3 fans, see note on fan performance curves on page *5.*

- **Shafts**

Shafts are ground or machined to close tolerances to ensure accurate fits with bearings and hub. Shafts are designed to accommodate at least a *5%* increase in speed over the maximum speed for each class limit. All hubs are keyed to the shaft.

- **Housing**

Fabricated from heavy gauge steel, continuously welded for strength and air-tight construction. Heavy angle reinforcement on the fan case increases the stiffness and provides support for the bearing pedestals on large fans. In many instances, fan housings size 660 and larger, must be split and shipped knocked-down due to their size and configuration.

- **Wheel**

Type ULF Ultrafoil wheels have double-thickness airfoil blades, accurately die-formed and jig welded to ensure consistent profile to maintain maximum design efficiency. Type UNF Unifoil wheels have single-thickness airfoil blades, accurately die-formed and jig welded to the shroud and back plate. Reinforcing rings are added a t higher tip speeds. All wheels have heavy-duty cast iron or aluminum hubs. Shrouds are formed from a one piece spinning, thus ensuring minimum inlet clearance to maintain design efficiency.

- **Balancing**

All wheels are balanced statically and dynamically on precision electronic balancing machines, for both single plane and two plane components, to close tolerances.

- **Flanged Outlets**

Available on all classes of construction.

ACCESSORIES

- **Quick Opening Access Doors**

Located on fan scroll, with cast aluminum clamps that release and rotate after slackening nut. Door is gasketed to eliminate leakage.

- **Drains**

Plugged drain can be furnished in lowest part of scroll.

- **Shaft Seals**

Simple shaft seals of impregnated compressed Aramid fibers, mineral wool, or UHMW are available to prevent excess leakage of air along the shaft. Carbon seals or stuffing box seals are also available for special applications.

- **Inlet Screens**

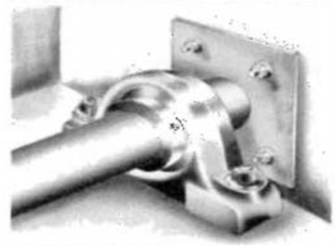
Inlet Screens are available on all open inlet fans. With large open wire mesh, friction loss is negligible for all classes and sizes of fans. If finer mesh screen is required, additional friction losses must be considered.

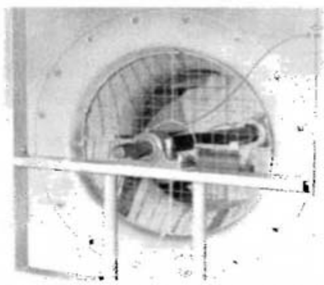
- **Back Draft Damper and Cowl**

On smaller sized fans, up to size *365,* automatic back draft dampers are available for roof mounted exhaust fans. Outlet cowls to prevent wind-blown rain and snow entering the fan are also recommended, and are available as an optional extra.

- **Cooling Discs**

On temperatures above 200° F., a cooling disc between the fan case and the inboard bearing is required to prevent heat conduction along the shaft. This allows cooling air to be drawn over the bearing and radiates the shaft heat to atmosphere.





BEARINGS

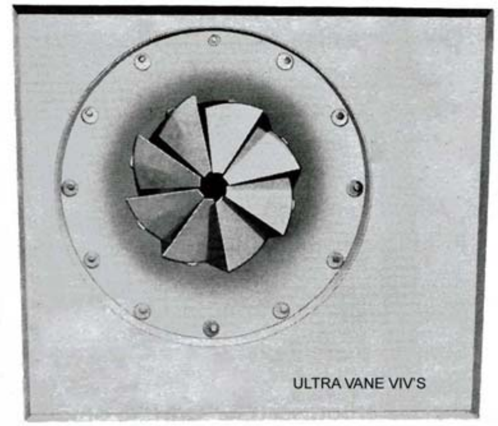
Grease lubricated anti-friction ball or roller bearings are standard on Sheldons Ultrafoil and Unifoil fans, sized for an average life expectancy of at least 150,000 hours. On larger fans and some Class 3 fans, split pillow block bearings are used having a higher bearing life. In some instances, oil lubricated roller bearings may be necessary.

Extended grease pipes may be fitted on all bearings that are normally inaccessible, to bring the grease point beyond the inlet duct.
Sleeve bearings are available for Type ULF Ultrafoils size 365 and larger, but require longer shafts and modifications to the standard bearing supports. Sleeve bearings are not normally used on commercial applications.

AERODYNAMIC LOSSES OF ARR. 3 FANS

Bearings and bearing supports in the inlet of SWSI, Arr. 3 fans, can cause a slight reduction in fan performance from the ratings published for SWSI, Arr. 1 fans. Inlet bearing losses for Arr. 3 fans are shown on each SWSI performance-rating curve for Class 1, 2 and 3 fans.
Class 1 DWDI fans have been tested and rated with bearings in the inlet and therefore no losses need normally be considered for this class. However, losses for other classes of DWDI fans must be considered.

METHODS OF FLOW CONTROL



Variable inlet vanes (V.I.V.'s)
On modern office building projects, considerable economy in power consumption can be obtained by reducing air conditioning and ventilation requirements during low use periods, such as, evenings, nights and weekends. Sheldons' variable inlet vanes (V.I.V.'s) provide this economy of energy consumption by providing controlled pre-swirl to the fan wheel, thus reducing the inlet velocity relative to the wheel blades.
Although higher in initial cost the use of variable inlet vanes results in a large long term savings in horsepower over the conventional outlet louver damper control.

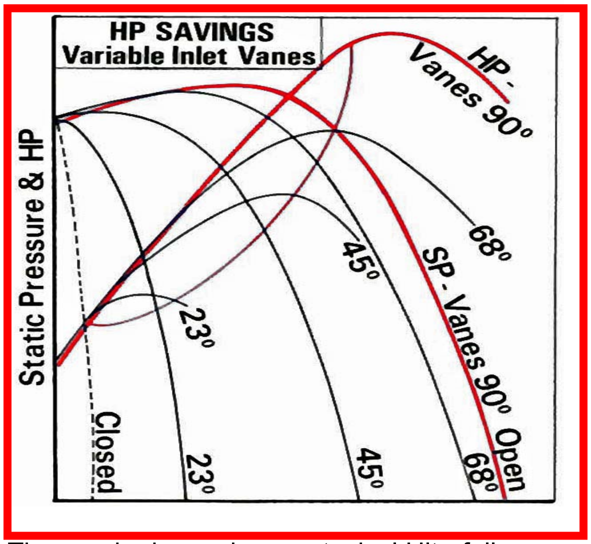
Construction
All vanes are supported in nylon or oilite bearings, running on cadmium plated or stainless steel shafts to reduce the incidence of rust and/or corrosion, when applied to air conditioning systems. To reduce aerodynamic losses, Sheldon V.I.V.'s have airfoil shaped vanes with no supports in the fan inlet.

Although the standard VIV's are designed for the maximum shut-off pressure for the class of construction of the fan, they are supplied mainly for flow control and should not be used in applications requiring tight shutoff. DWDI fans have inter-connected sets of V.I.V.'s to ensure equal flow into each inlet. For tighter sealing, rubber seals can be provided at the periphery and edges of each vane.
Damper operating torques are available on request.

SOUND LEVELS V.I.V.'s
Slight changes in the basic sound level occur at different vane angles. Tests made on a size 245 SWSI Ultrafoil, Arr. 1 fan a t 1000 rpm, show the following net changes, which generally apply over the useful working range of the fan curve:
Speed and diameter, as well as point of rating, will make slight changes to these values, but they do serve as a useful guide to the sound effect of variable inlet vanes at low flows.

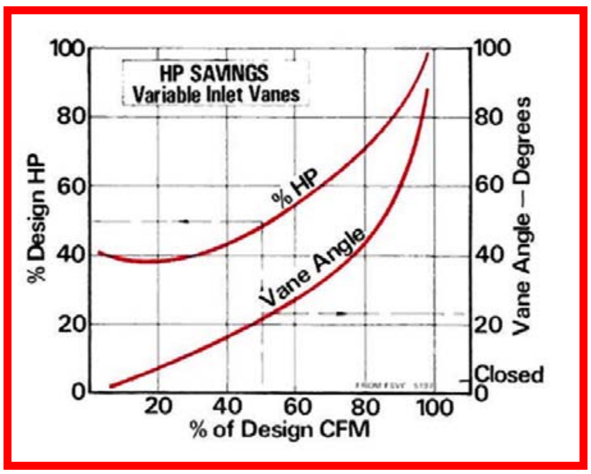
	OCTAVE BANDS FOR VIV DAMPERS - Hz							
VIV's Open	**63**	**125**	**250**	**500**	**1000**	**2000**	**4000**	**8000**
100%	+4	+3	+4	+4	+4	0	-2	+2
75%	+3	+1	+4	+4	+3	0	-1	+2
50%	+4	+2	+5	+4	+3	-1	-3	+2
25%	+4	+2	+5	+5	+4	+3	+2	-2
Closed	0	-3	+1	+1	-1	-4	-4	-3

Fig 1



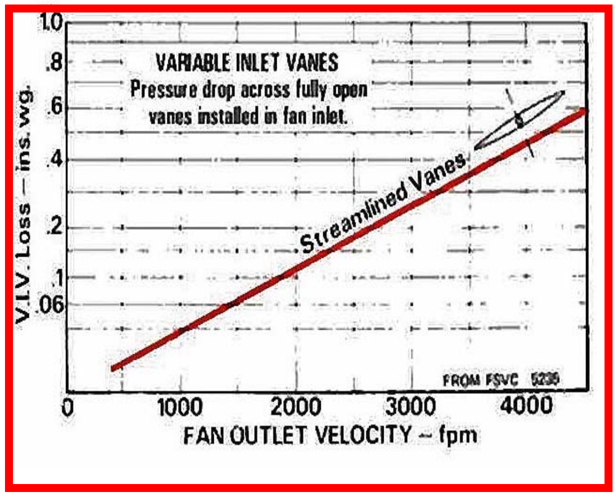
The graph above shows a typical Ultrafoll performance curve with variable inlet control Note the significant change in the horsepower curve with inlet vane control, based on a constant system resistance.

Fig 2



Savings In horsepower for a constant system resistance can be estimated from the chart above, as a percentage of the horsepower with the vanes fully open. The approximate vane angle can also be read directly from the graph for any fan given percentage of design airflow.

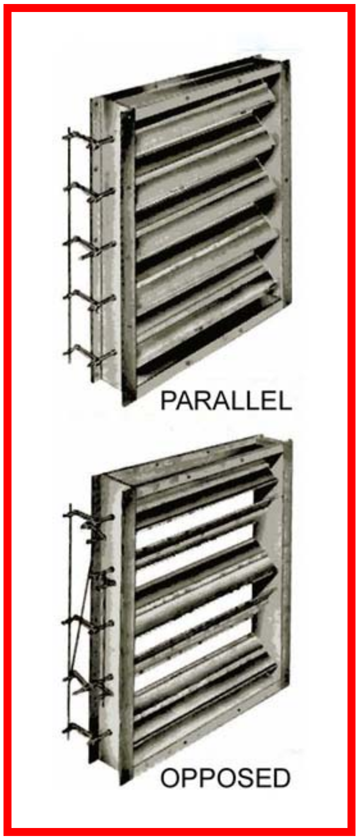
Fig 3



Due to the slight restriction of airflow caused by the vanes in the wide-open position, a small correction to the fan performance obtained from the rating curves must be made. V.I.V. losses are conveniently related to the outlet shown on the performance rating curves. Determine the fan outlet velocity, enter the chart at this value, and read off the fully Open V.I.V. loss. Add this loss to the required system static pressure corrected to standard air conditions before entering the selection.

OUTLET DAMPERS

For the lowest initial cost, the use of parallel blade outlet dampers recommended. Although slightly higher in initial cost, opposed blade dampers overcome the non-linear control characteristic of the parallel blade damper to a large extent. For both parallel and opposed blade dampers, horsepower reduction is achieved by forcing the fan to operate lower on its own horsepower curve, due to the lower flow rate in the dampened condition. Outlet dampers can be designed for any catalog pressure and for high temperature if necessary. All louver dampers are supplied with channel frames for bolting directly to the fan discharge flange.

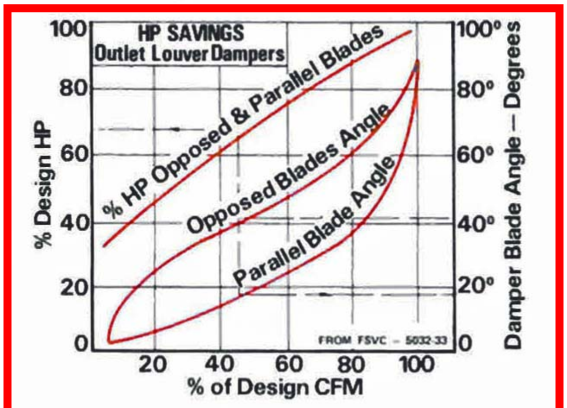


Sound levels

Slight changes in the basic sound levels occur at different damper blade angles. Tests on a 365 SWSI Ultrafoil fan at 1000 rpm with opposed blade dampers, show the following net changes, which generally apply over the useful working range of the fan curve.

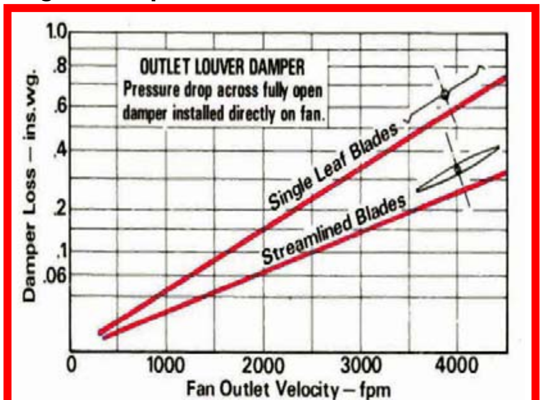
	OCTAVE BANDS FOR OUTLET DAMPERS - Hz							
Damper Blades Open	**63**	**125**	**250**	**500**	**1000**	**2000**	**4000**	**8000**
100%	+1	+2	+4	+4	+6	+7	+3	+2
75%	+2	+1	+1	+2	+6	+7	+2	-4
50%	+8	+3	+6	+5	+8	+7	+1	-3
25%	+8	+3	+6	+4	+8	+9	+3	-5
Closed	+6	+2	+5	+4	+8	+8	+1	-5

Fig. 4 Horsepower Savings

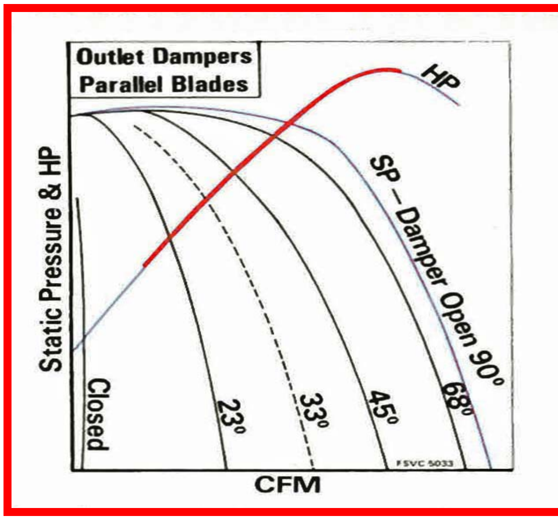


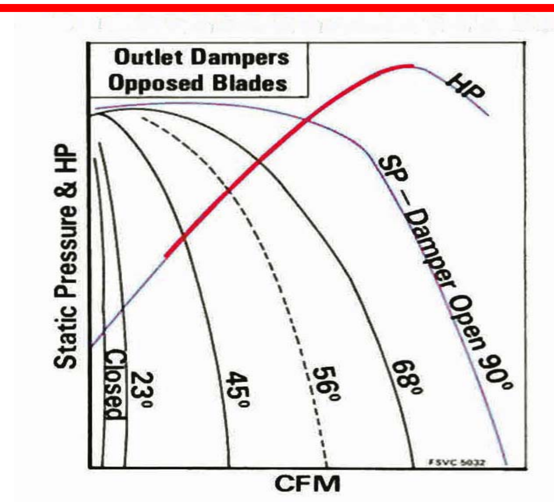
Saving in horsepower for a constant system resistance can be estimated from the charts above, as a percentage of the let horsepower with the damper blades fully open. The approximate damper blade angle can also be read directly from the chart for any given percentage of design airflow.

Fig. 5 Damper Losses



Due to the slight resistance to the airflow caused by the out let dampers in the wide-open position, a small correction to
The fan performance obtained from the rating curves must be made. Outlet damper losses are added to the required static pressure **before** entering the selection curves, and are conveniently related to the fan outlet velocity shown on the performance curves.





Typical Ultrafoil performance curves with parallel blade and opposed blade outlet dampers at a constant system resistance, are shown above based on actual tests. Note that the horsepower at all blade angles follows the fan characteristic horsepower curve. Unifoil fans with outlet dampers show similar characteristics.

APPLICATION DATA

For some time, fan manufacturers have been concerned with the effect of inlet and outlet connections on the actual performance of a fan in its installed system, Published fan ratings are based on tests conducted under laboratory conditions, and rated in accordance with AMCA procedures to ensure uniformity in presentation for comparison purposes. Since most installations are unlikely to provide laboratory type conditions, it is necessary to make some allowances in the fan selection for,

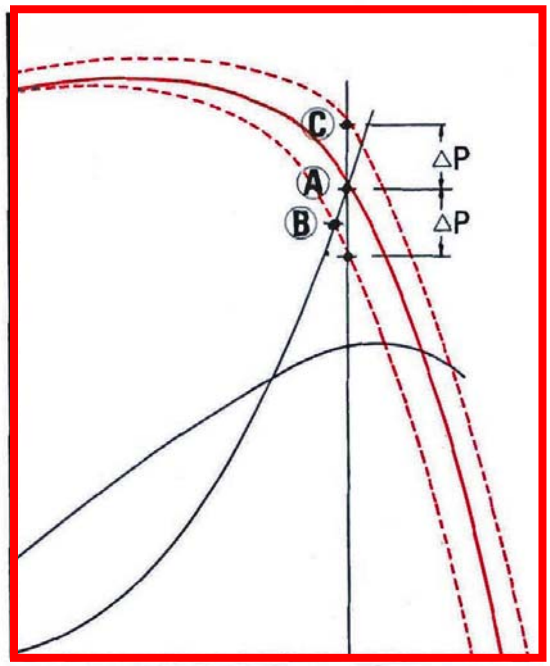
(a) Site conditions that cause poor inlet and outlet connections, and
(b) Additional fan accessories that could affect fan performance.

(a) SITE CONDITIONS

Incorrect fan inlet and outlet connections have an adverse effect on fan performance. In general, the effect of the system on the fan causes the fan to lose both volume and pressure, as shown by the lower broken line in Fig.8.

This loss in performance at a given flow rate, (shown as ΔP in Fig. 8) causes the fan to operate at point B, rather than point A, on the system curve. This loss in performance can be compensated for by adding ΔP to the final system static pressure, and selecting the fan

Fig. 8 Compensating for loss in performance

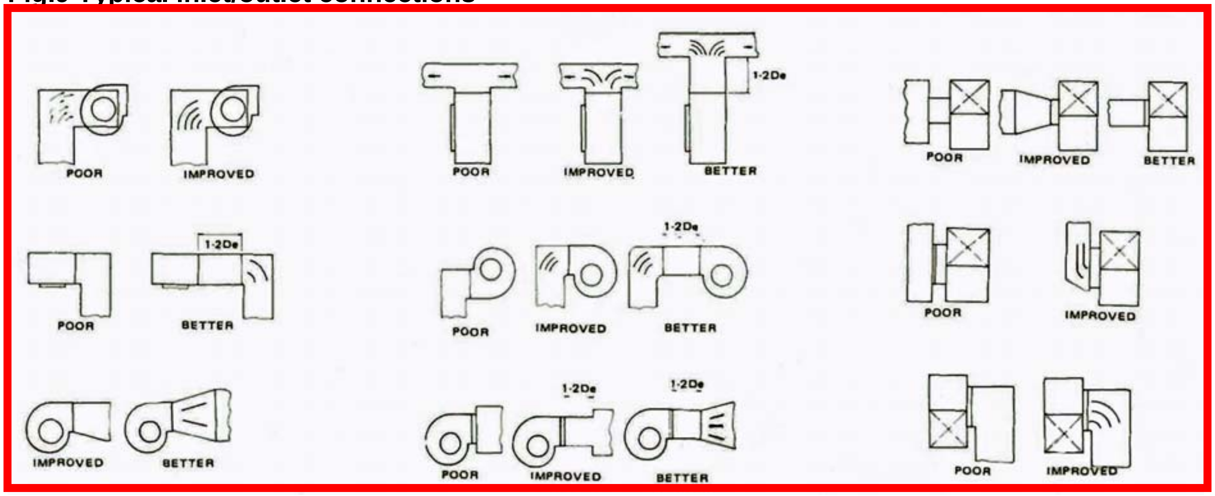


for point C. The fan will then operate at point A in the system, and give the required design flow and pressure. System effects vary with velocity, and therefore can be directly related to the fan outlet velocity pressure, which is shown on the performance rating curves. Values of the effect of poor inlet and outlet conditions on the system can usually be compensated for by using what are known as System Effect Factors (**SEF's**). These are clearly detailed in AMCA publication #201 for various inlet and outlet connections. It should be noted that this fan performance loss due to the system effect occurs in addition to any friction loss through the connection. Sheldons have shown some typical poor inlet and outlet connections in Fig. 9 below, indicating where poor connections can be improved by minor alterations to the duct layout. Applying information contained in AMCA publication #201 can make a more detailed review of the subject of SEF's.

(b) FAN ACCESSORIES

The rating curves shown in this catalog are based on SWSI Arr. 1 fans, and DWDI Arr. 3 fans. When SWSI Arr. 3 fans are selected, the aerodynamic loss caused by the obstruction of the bearings in the inlet must be added to the static pressure before the fan selection is made. The value of the bearing loss for each class of fan is shown at the bottom of each rating curve. When applying variable inlet vanes or outlet louver dampers to a fan, the pressure drop across them in the fully open position, can be obtained from Figs. 3 and 5 shown on pages 6 and 7. Belt guards offer some resistance to airflow, which must be considered when large horsepower drives are being used.

Fig.9 Typical inlet/outlet connections



GENERAL APPLICATION NOTES

1. The fan performance is most adversely affected by poor connections located directly at the fan inlet or outlet connections. Therefore, every attempt should be made in duct layout to ensure that elbows are located as far upstream or downstream as possible to reduce their effect on fan performance.
2. Sheldons recommend that the fan outlet be provided with a straight duct at least one wheel diameter in length to ensure more fully developed flow from the fan discharge, as shown in Fig. 9.
3. When DW fans are used in plenums, the distance between the fan case and the plenum wall may affect the fan performance. For calculation of these effects, see the information below.
4. Arr. 3 fans with bearings in the air stream are not suitable for handling contaminated air such as exists in laundry exhausts, or any temperatures higher than approximately 130° F.
5. For higher temperatures and for contaminated air or gases, it is recommended that Arr. 1 or Arr. 9 fans, which have their bearings outside the air stream, be used. For temperature limitations on fans, see table on page 3.
6. It is recommended that the fan and motor be installed on a common structural base supported on vibration isolators, and that all ductwork be connected to the fan by means of flexible connections, to prevent the transmission of vibration to the building structure.

DWDI FANS IN PLENUMS

Since all DWDI fans are tested with open inlets, the use of freestanding DWDI fans inside built-up air conditioning plenums requires that some allowances be made for the size of the plenum on the fan performance. Table 3 below, adapted from AMCA Publication #201, shows the approximate combined system effect and friction loss at any given fan outlet velocity for fans located inside plenums and for various distances from the plenum wall. These losses must be added to the system static pressure corrected to standard air conditions before the final fan selection is made.

Example: A 365 DWDI fan is located inside a plenum 22" from the inside wall. The fan outlet velocity is 2530 fpm, giving a velocity pressure (VP_o) of 0.40" wg. at standard air conditions.

L = 22/36.5 - .6. From Table 3, loss is .41 x VP_o,

I.E. Loss = .41 x .40 = .16" wg.

Note: For Standard air:
$VP_o = (\text{Velocity}/4005)^2$ “wg.

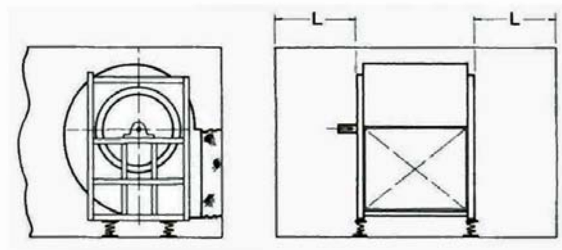


Table 3

L – distance from fan case to wall of plenum as fraction of wheel diameter	Loss as fraction of VP at fan outlet
0.3D	0.84
0.4D	0.61
0.5D	0.48
0.6D	0.41
0.7D	0.38
0.8D	0.35
1.0D	0.29
1.2D	0.23

DRIVE LOSSES

In common with most fan manufacturers, the rating curves shown in this catalog have been determined from tests on direct connected fans only. For any belt-driven fan application, an allowance for V-belt drive losses must therefore be added to the horsepower obtained from the rating curves, as shown. This curve, shown on Fig. 10, is adapted from AMCA Publication #203 and is based on tests and experience, and some variation from these values may be expected.

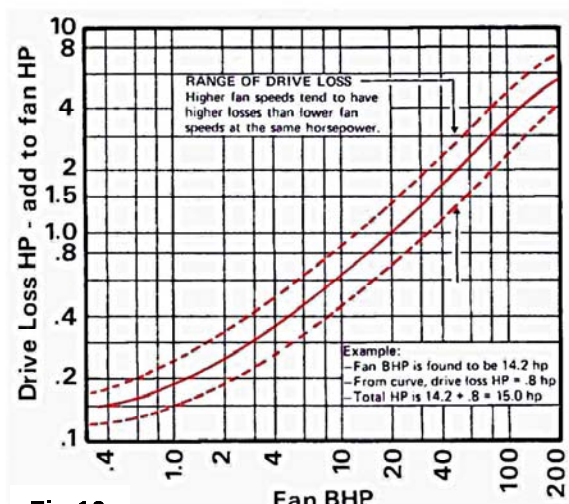


Fig 10

SELECTION PROCEDURE

Performance rating curves shown in this catalog are based on the following:

1) Standard air at .075 lb/ft^3, which is approximately 68° F, 50% R.H. and 29.92" Hg barometric pressure. Standard air in S.I. units is 1.2 kg/m^3, which is approximately 20° C, 50% R.H. and 101.3 kPa barometric pressure.

2) Fans are tested with outlet ducts and open inlets, but apply equally to fans with inlet ducts.

3) Tests do not include any optional appurtenances such as, variable inlet vanes, outlet dampers, inlet screens, belt guards, or any other obstruction in the air stream, or any V-belt drive losses.

By following the itemized procedure listed here, all necessary factors for the correct selection of the fan can be made easily and accurately, in order to achieve the desired performance in the field.

1) From the system design, the amount of air required will have been calculated. The elevation and air temperature will also be known.

2) From available duct design procedures, calculate the resistance of the duct system in ins. wg. This is the system static pressure.

3) To select fans at a density other than .075 lb/ft^3 or 1.2 kg/m^3, a correction must be made both before and after selection, as follows:

   a) For the given temperature and elevation, select the correct density.
   b) Divide the actual static pressure by the density correction factor to obtain the equivalent system static pressure at standard density.

Note: - The CFM does not change with density.

4) The next step is to select the fan size. Since the value of the losses of the accessories cannot be assessed until the final fan size is determined, it is necessary to use the equivalent static pressure as an initial guide only to the approximate operating point on the performance rating curves.

Several fan sizes can be chosen for any given performance, depending on whether low first cost, low energy consumption, low sound levels, designed duct velocity, or space limitations, are made important.

Generally, if the point of operation is selected near the top of the fan pressure curve (A) the fan will be the most efficient and most quiet, but it will be the largest.

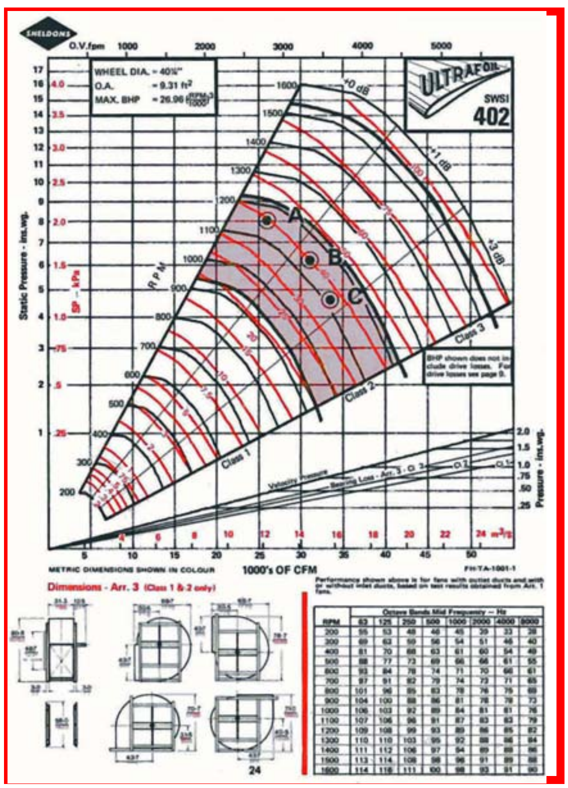
If the point of operation is selected slightly further down the curve (B) the fan will be smaller and slightly less efficient and provide nominally the same sound levels as (A).

For economy in first cost, choose the fan at (C) which will give a higher outlet velocity and slightly higher sound levels.

5) Having selected the fan size, calculate the losses due to the accessories.

   a) For variable inlet vanes and outlet dampers, enter Figs. 3 and 5 on pages 6 and 7, at the outlet velocity of the chosen fan and read off the pressure loss at standard density.
   b) For DWDI fans in plenums, calculate the combined system effect and friction loss for the appropriate outlet velocity of the fan at standard density, as shown on page 9.
   c) For SWSI, Arr. 3 fans, read off from the rating curve the loss in ins. wg of the bearing in the inlet for Class 1, Class 2, and Class 3 fans, at standard density.

6) SEF's. The effect of unusual inlet and outlet conditions on the fan performance can be calculated from AMCA Publication #201,and these effects added as losses in ins. wg at standard density to the equivalent static pressure.



SELECTION PROCEEDURE CON'T

7) Add all accessory losses and SEF losses to the system fan static pressure at standard density to obtain the equivalent selection static pressure. Reenter the performance rating curves for the chosen fan size with the equivalent selection static pressure and the required CFM.

8) Estimate the fan speed at the intersection of the CFM and the equivalent selection static pressure values.

9) Estimate the fan BHP at this same intersection point, interpolating between the constant HP curves shown.

Note: The HP values shown on the performance rating curves are the actual fan BHP, but have been listed in convention motor HP sizes far convenience in selecting the approximate motor HP size.

10) Add the drive loss HP from Fig. 10 on page 9 to the HP obtained from the fan curve, to obtain the overall fan BHP at standard density. This overall HP must now be multiplied by the density correction factor obtained in step 3 (a) on page 10, to give the fan HP at site conditions.

Using the above procedures, the fan will deliver the air volume and required system static pressure at the calculated speed and HP.

11) Sound. Using the calculated fan speed, read the octave band sound power levels from the table shown below the performance rating curves, interpolating between the listed speeds as necessary. Since these sound levels are for fans operating near maximum efficiency, an approximate correction can be made for other points of rating by adding the dB values noted in the zones defined by the radial lines in which the performance has been selected.

12) Dimensions. Basic dimensions of Arr. 3, Class 1 fans are conveniently provided below each performance curve for quick and easy reference for layout purposes. Dimensions for Class 2 and 3 fans and for other arrangements are shown on pages 54 through 58.

13) Metric (S.I.) units have also been shown in color on the rating curves for easy reference.

WK^2 AND START TIMES

Table 4		SW			DW		
	Size	Class 1	Class 2	Class 3	Class1	Class 2	Class 3
Unifoil	122	1.7	2.0	Not Available See Engineering	3.5	4.0	Not Available See Engineering
	135	2.5	3.1		5.0	6.2	
	150	4.3	4.0		8.6	8.0	
	165	6.6	7.8		13.2	15.6	
	182	11.3	12.6		22.6	25.6	
	200	17.3	18.7		34.6	37.4	
	222	31.5	33.4		63.0	66.8	
Ultrafoil	245	43	51	55	102	103	100
	270	57	57	75	114	114	150
	300	97	97	118	194	194	236
	330	152	152	176	304	304	352
	365	218	218	255	436	436	510
	402	365	365	431	730	730	862
	445	732	732	870	1463	1463	1740
	490	1116	1116	1268	2232	2232	2536
	542	1733	1733	1809	3466	3466	3618
	600	2845	2845	3536	5690	5690	7072
	660	4215	4251	4380	8430	8430	8760
	730	7120	7120	7746	14240	14240	15492
	807	10750	10750	11839	21500	21500	23678
	890	15878	15878	17134	31756	31756	34268

The WK^2 values shown above are in lb/ft^2

Low horsepower motors used on some applications such as large low pressure centrifugal return air fans, may not be capable of starting the fan within a reasonable time.
Sheldons provide an approximate method of calculating the start times of all fans shown in this catalog, using the formula below. Values of WK^2 for all fans listed in this catalog are shown above.

$t = (WK^2 \times N^2) / 1.62 \times HP_m$

t = Start time in seconds.
WK^2 = Polar moment of inertia - $lb.ft^2$.
N = Fan speed in 1000's of rpm.
HP_m = Motor horsepower.

START TIMES
10 seconds or less - satisfactory
11 to 15 seconds - probably satisfactory.
15 to 20 seconds - check with starter and motor manufacturer.
Over 20 seconds - not recommended.

In order to compare published motor WK^2 capability with that of the fan on V-belt drive applications, it is necessary to convert the fan WK^2 to the equivalent WK^2 referred to the motor shaft, by applying the following formula:

$WK^2_m = (WK^2_f) \times (N_f / N_m)^2$

Where WK^2_m = equivalent fan WK^2 at motor speed
WK^2_f = fan WK^2 at fan speed
N_f = fan speed
N_m = motor speed