Plenum Fans in HVAC Equipment: The Good, the Bad, and the Ugly

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ABSTRACT

This paper provides an overview of the benefits of plenum fans and a detailed discussion of some of the pitfalls. After covering some of the benefits leading to extensive use of plenum fans, included is a brief discussion of the major complaint expressed about plenum fans, which is that plenum fans are less efficient than housed fans. Finally, the bulk of the paper covers problems that can result from poor design practices, sloppy construction, and careless handling.

INTRODUCTION

Plenum fans have become popular with some segments of the HVAC industry because their use can lead to shorter cabinets (read less expensive) and quieter applications. As with all engineering choices, there are trade-offs and pitfalls to be considered.

Plenum fans cannot compete with properly utilized housed fans for efficiency. Properly utilizing a housed fan means a "draw-through" unit. This configuration puts the fan at the discharge end of the unit, drawing the air through the coil. The fan must be connected directly to the ductwork with two to three equivalent diameters of straight duct before any turns. In many applications, however, housed fans are not used in "draw-through" configurations or are not properly ducted. In these configurations, the efficiency gap narrows considerably.

As with any component, you must exercise care in design, construction, and handling of plenum fans. Several examples will be presented, documenting problems that were traced back to problems with frame design, component alignment, and component handling. These issues are important for all fans but some at least can have a uniquely plenum fan twist. For example, any fan can have a problem with frame resonances, making them impossible to balance properly. Plenum fans, however, with their large, flat inlet plates can transmit these resonance frequencies into the airstream like a loud speaker.

Sound and Airflow Data

All data presented in this paper were acquired in a laboratory set up to acquire airflow data in accordance with AMCA Standard 210 (AMCA 1985) and sound data in accordance with AMCA Standard 300 (AMCA 1995). The laboratory is accredited under AMCA 111 (AMCA 1989) for performing the Standard 300 testing. It is not accredited for the 210 tests, but enough certified fans have been tested here to be confident of the airflow measurements. In any case, most of the data presented are comparative; thus, all that really matters is that the measurement procedures be consistent.

Some fan data presented are projected from the test data using methods set out in AMCA standard 301 (AMCA 1990). Also, some of the sound data are not presented as specified in AMCA Standard 300. First of all, some data are presented for the frequencies below the 50 Hz one-third octave band, which are not addressed in the standard. Also, much of the data are presented in one-third octaves, rounded to the nearest tenth dB, rather than summed to full octaves and rounded to the nearest integer. This is done where it better illustrates the point being made. The testing standards list the measurement error for the test procedure to be ±6 dB for the 63 Hz octave band and ±3 dB for the next seven bands. These are for lab-to-lab comparisons. The data presented here were all taken in the same laboratory. Of more importance are the variations from place to place in the lab, which, by standards, must be less than

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0.5 to 3 dB, depending on the one-third octave band. Much of the data were taken from the same position, with the same room setup, on the same day. Here, what is important is the measurement-to-measurement repeatability, which, in this lab, is usually less than 0.5 dB in the first couple of bands and much less in the middle and upper bands.

PLENUM FANS, THE GOOD

It should be no surprise that economic issues control many, if not most, construction projects. Specifying plenum fans can lead to shorter cabinets and, thus, lower unit cost. Furthermore, with the appropriate acoustical treatment of the plenum, one can achieve substantially lower sound emissions with plenum fans over housed fans.

Shorter Cabinets

Although it is not the optimal use of a housed fan, they are frequently positioned in the middle of the unit rather than at the end. In this configuration, using a plenum fan can usually shorten the unit, thus reducing costs. Figure 1 shows drawings for two units laid out using the recommended component spacings. Using the design spacings, the plenum fan unit is about 12 in. (305 mm) shorter than the housed fan unit. Though seemingly a small difference, this still can be a significant savings. Furthermore, these are "recommended" spacings for the most efficient fan size. These are frequently compromised, and a smaller plenum fan may be used so the unit could be substantially shorter. On the other hand, though, this same tendency to infringe on the recommended spacings can contribute to some of the problems that will be discussed later.

Cabinet Attenuation

To estimate the sound power output of a housed fan (DWDI centrifugal airfoil) unit compared to a plenum fan (SWSI centrifugal airfoil) unit, the two units shown in Figure 1 were modeled with a computer program written by the author for the prediction of the sound output of custom air handlers. This is based on the ASHRAE acoustical algorithms (Reynolds and Bledsoe 1991). The fan sound shown is projected from the test data using methods specified in AMCA 301. Fans were selected for 35,000 CFM at 4.0 in.WC (16,517 L/s at 996 Pa) of static pressure across the fan. No inlet screens were assumed so the selection static pressure for the plenum fan was 4.0 in. WC (996 Pa). The housed fan data had to be adjusted for unducted discharge and for losses associated with attaching a discharge diffuser (to spread the discharge more evenly across the filter bank). The selection static pressure for the housed fan was therefore 4.53 in. WC (1,128 Pa). Fans were selected at approximately the same point of operation (position on the fan curve). The housed fan selected was, therefore, a 36.5 in. (927 mm) wheel running at 1,071 rpm (32.2 BHP [24.0 kW]), and the plenum fan selected was a 54.25 in. (1,378 mm) wheel running at 680 rpm (31.8 BHP [23.7 kW]). Being single width, a plenum fan will always require a larger wheel diameter than a DWDI housed fan for the same airflow. For both units, the section between the fan and the rigid filters was modeled with perforated liner. The remaining sections were modeled with solid steel liners. Table 1 shows the basic fan data and the modeled data for the full unit. The sound data for the housed fan (AF01-36) comes from the manufacturer. The plenum fan data were projected from data acquired at the laboratory. In this example, the plenum fan unit is substantially quieter than the housed fan unit. Chances are, however, that with this unit, a smaller plenum fan would be selected, probably a 49 in. (1,245 mm) or even 44.5 in. (1,130 mm) fan, which would allow for a shorter unit, thus saving money, but would also result in a louder unit. It would still be quieter than the housed fan but would then be less efficient. For example, with the 49 in. (1,245 mm) plenum fan, the unit's A-weighted, discharge sound power (LwA) would increase to 89 dB, and motor HP would increase to 32.5 BHP (24.2 kW).

PLENUM FANS, THE BAD

The item most often cited by those adamantly against plenum fan use is that a housed fan is more efficient. As fans are usually tested for a company's catalog, this is undeniably true. Particularly with DWDI housed fans, the test setup bears

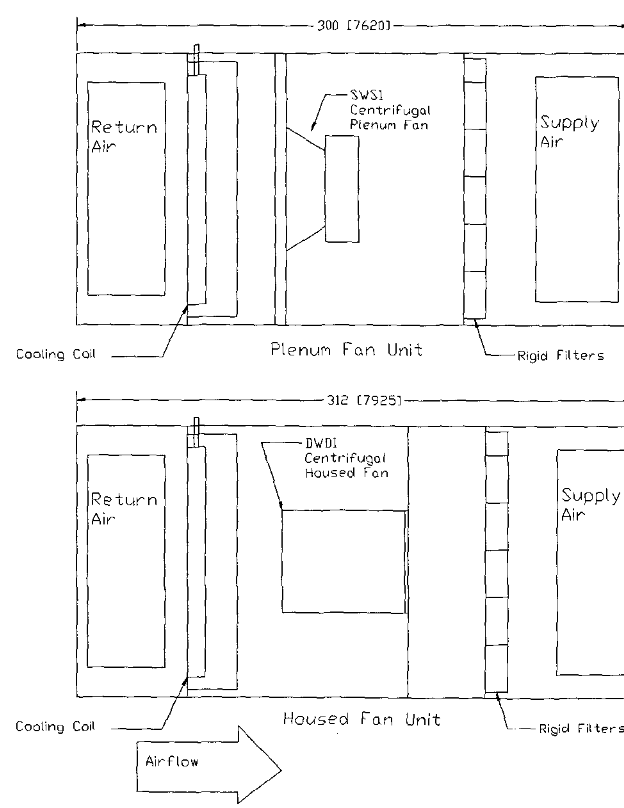

Figure 1 *These drawings show typical layouts of simple, equivalent housed fan and plenum fan-based HVAC equipment. Lengths are in inches and mm.*

Table 1.

Housed Fan Unit:									
	Octave Band Center Frequency								
	62.5	**125**	**250**	**500**	**1,000**	**2,000**	**4,000**	**8,000**	**LwA**
Supply Fan—AF01-36/35,000 CFM at 4.53 in. WC (16,517 L/s at 1,128 Pa)									
Discharge Lw:	107	107	101	96	93	88	79	70	99
Inlet Lw:	107	107	101	96	93	88	79	70	99
Supply Air (2,743 mm × 1,143 mm)									
114 in. × 45 in. Lw:	107	105	98	94	89	79	68	57	96
Return Air (2,743 mm × 1,067 mm)									
108 in. × 42 in. Lw:	100	101	95	90	88	83	74	65	93
Plenum Fan Unit:									
	62.5	**125**	**250**	**500**	**1,000**	**2,000**	**4,000**	**8,000**	**LwA**
Supply Fan—PF02-54/35,000 CFM at 4.00 in. WC (16,517 L/s at 996 Pa)									
Discharge Lw:	91	97	93	92	88	82	85	83	94
Inlet Lw:	86	98	86	83	82	79	82	81	89
Supply Air (2,896 mm × 1,143 mm)									
114 in. × 45 in. Lw:	91	95	87	87	81	70	70	67	87
Return Air (2,743 mm × 1,067 mm)									
108 in. × 42 in. Lw:	79	92	80	77	77	74	77	76	84

Lw—sound power in decibels = 10 log W + 120 dB where W = acoustical power in watts

little resemblance to the ways these fans are configured in practice. The fans are generally tested with the fan at least four or five wheel diameters away from side walls. The fans are generally powered with a shaft of sufficient length such that the motor (dynamometer) is at least one wheel diameter away from the inlet, often more. In use, there is almost always a sheave encumbering one inlet and frequently a belt guard that will nearly block this inlet. Usually, at least one of the two inlets is within a wheel diameter of a wall. Sometimes, the fan is much closer than this. Worse, this is usually the side away from the sheave. Often, housed fans are not used in a draw-through configuration. When not attached directly to the discharge duct, you must account for what may be referred to as the unducted discharge losses. If the next device is a coil, you will probably also have a diffuser of some sort and associated diffuser losses.

Example—Laboratory Data

The opportunity arose from a service department request to investigate some of the losses associated with placing a DWDI housed, centrifugal, airfoil fan in a blow-through configuration. The fan tested had a diameter of 36.5 in. (927 mm). As the purpose of the tests was to determine what could be removed to increase airflow, we unfortunately did not have the time to investigate the losses associated with the sheave. For the initial performance test, the fan was configured with a discharge duct that was two equivalent duct diameters in length, with only the sheave and belts inhibiting the airflow. The final test had the fan configured as installed in the shipped unit, unducted, with an attached discharge diffuser, simulated airflow probes in the fan inlets, and a belt guard. Figures 2 and 3 show these data sets, adjusted by rpm to a matching operating point on the system curve using the fan laws (AMCA 210), along with test data for a couple of plenum fans.

As illustrated, if a housed fan can be used in a draw-through configuration, connected directly to a properly designed discharge duct, it is by far the more energy efficient choice, particularly if you can avoid belt guards. When the assembly was fully configured as outlined above, performance dropped substantially. For the graphs in the two figures, the performance of the encumbered fan was recalculated at a higher rpm using the fans laws (AMCA 210) so as to match performance at the intersection point with the system curve. At this higher rpm, the electrical power draw (kW) shown in Figure 3 increased by more than 6 kW (nearly 7 BHP). It is interesting to note that the belt guard had the largest effect on performance, based on the shift in the fan curve when the belt guard was added to the mix.

The most likely replacement, for the 36.5 in. (927 mm) DWDI fan would be a 49 in. (1,245 mm) SWSI plenum fan. The fan curves in Figures 2 and 3 include the curves for such a fan at an rpm selected to match the airflow and static pressure