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2 Air Handling Units

VAV terminals provide conditioned air to a zone or space. Conditioned air to the VAV terminals come from air handling units typically located in a building's mechanical room. The following sections describe the basic components that are normally found in air handling units.

Basic Fan Types

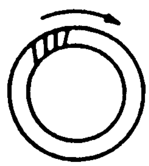
Two types of fans are generally used in HVAC work: centrifugal and axial. Centrifugal fans are most commonly used in VAV systems, particularly the forward curved and airfoil types.

Centrifugal Fans

With centrifugal fans, airflow is perpendicular to the shaft and induced by the wheel. The forward curved (FC) centrifugal fan (Figure A-2) travels at relatively low speeds and is used for producing high volumes at low static pressure. The fan will surge, but the magnitude is less than for other types. Another advantage is that it has a wide operating range. The low cost and slow speed of the FC fan are additional advantages that minimize shaft and bearing size. One disadvantage is the shape of its performance curve. It could allow overloading of the motor if system static pressure decreases. It has an inherently weak structure, and therefore is not generally capable of the high speeds necessary for developing higher static pressures.

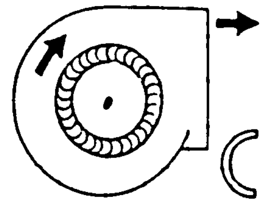
The airfoil fan (AF) (Figure A-3) is another type of centrifugal fan. It travels at about twice the speed of the FC fan. Generally, the larger the fan, the greater the efficiency. The magnitude of the AF fan's surge is also greater than that of the FC fan. Its higher speeds and bearing sizes, along with nonoverloading brake horsepower (BHP), allow higher efficiency but make proper wheel balance more critical. Also, as block-tight static pressure is approached, unstable operation may occur.

IMPELLER DESIGN



Efficiency is somewhat less than airfoil and backwardly curved bladed fans. Usually fabricated of light-weight and low cost construction. Has 24 to 64 shallow blades with both the heel and tip curved forward. Air leaves wheel at velocity greater than wheel tip speed and primary energy transferred to the air is by use of high velocity in the wheel. For given duty, wheel will be the smallest of all centrifugal types and operate at lowest speed.

HOUSING DESIGN



Scroll is similar to and often identical to other centrifugal fan designs. The fit between the wheel and inlet is not as critical as on airfoil and backwardly inclined bladed fans.

Figure A-2. Forward Curved Centrifugal Fan.

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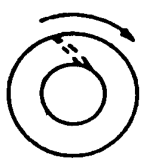
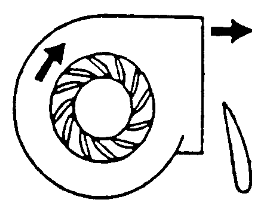
IMPELLER DESIGN



Highest efficiency of all centrifugal fan designs. 10 to 16 blades of airfoil contour curved away from the direction of rotation. Air leaves the impeller at a velocity less than its tip speed and relatively deep blades provide for efficient expansion within the blade passages. For given duty this will be the highest speed of the centrifugal designs.

HOUSING DESIGN



Scroll type. Usually designed to permit efficient conversion of velocity pressure to static pressure thus permitting a high static efficiency. Essential that clearance and alignment between wheel and inlet bell be very close in order to reach the maximum efficiency capability.

Concentric housings can also be used as in power roof ventilators. Since there is efficient pressure conversion in the wheel.

Figure A-3. Airfoil Centrifugal Fan.

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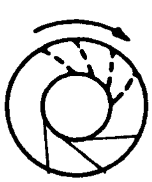
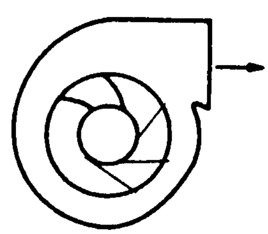
Two additional types of centrifugal fans are the backwardly curved or backwardly inclined fan (Figure A-4), and the radial tip or radial blade fan (Figure A-5).

IMPELLER DESIGN



Efficiency is only slightly less than that of airfoil fans. Backwardly inclined or backwardly curved blades are single thickness. 10 to 16 blades curved or inclined away from the direction of rotation. Efficient for the same reasons given for the airfoil fan above.

HOUSING DESIGN



Utilizes the same housing configuration as the airfoil design.

Figure A-4. Backwardly Curved or Backwardly Inclined Centrifugal Fan.

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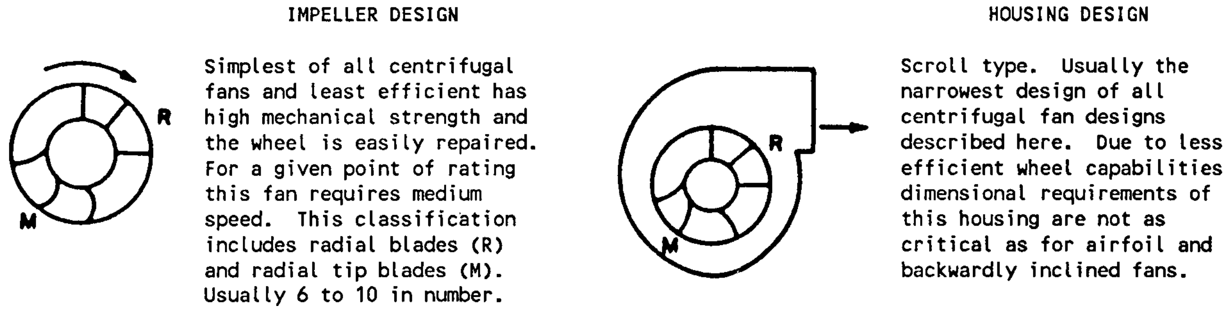

Figure A-5. Radial Tip or Radial Blade Centrifugal Fan.

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Forward curved fans are stable and economical, but airfoil fans are better for discharging air into a plenum, and for multizone units and others having higher pressure through the system. The airfoil fan requires about 75 to 80 percent wide open volume, but the maximum BHP of the forward curved blade wheel is 100 percent wide open. This is a disadvantage for systems with large pressure fluctuations. Since fans operate best at peak efficiency, it is important to choose their size properly for quiet performance. For static pressures above 2 in. w.g., the backward inclined and airfoil fan are used. Below 2 in. w.g., the forward curved fan is best as far as noise is concerned.

Axial Fans

In axial fans, the airflow is parallel to the shaft. Axial fans include propeller (Figure A-6), tubeaxial (Figure A-7), and vaneaxial (Figure A-8).

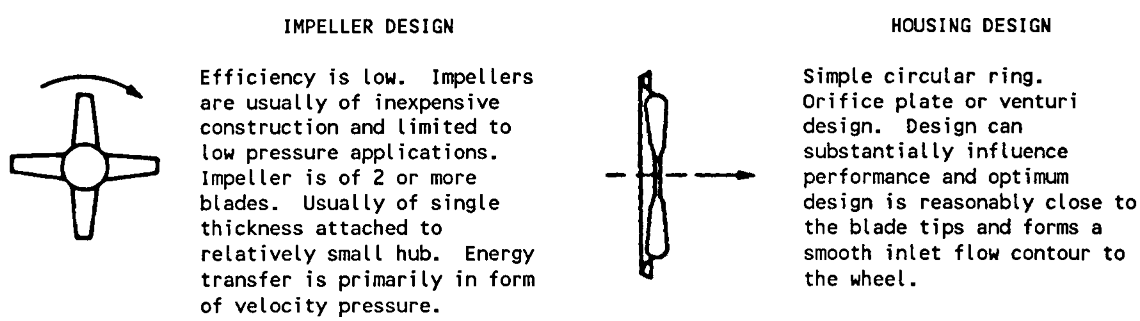

Figure A-6. Propeller Fan.

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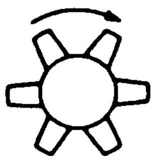
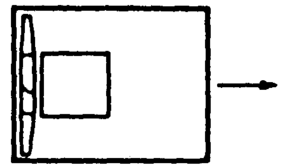
IMPELLER DESIGN



Somewhat more efficient than propeller fan design and is capable of developing a more useful static pressure range. Number of blades usually from 4 to 8 and hub is usually less than 50% of fan tip diameter. Blades can be of airfoil or single thickness cross-section.



HOUSING DESIGN

Cylindrical tube formed so that the running clearance between the wheel tip and tube is close. This results in significant improvement over propeller fans.

Figure A-7. Tubeaxial Fan.

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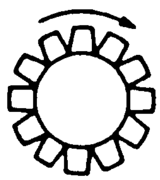
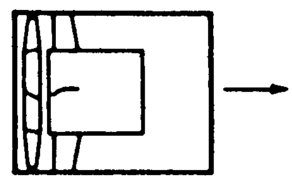
IMPELLER DESIGN



Good design of blades permits medium to high pressure capability at good efficiency. The most efficient fans of this type have airfoil blades. Blades are fixed or adjustable pitch types and hub is usually greater than 50% of fan tip diameter.



HOUSING DESIGN

Cylindrical tube closely fitted to the outer diameter of blade tips and fitted with a set of guide vanes. Upstream or downstream from the impeller. Guide vanes convert the rotary energy imparted to the air and increase pressure and efficiency of fan.

Figure A-8. Vaneaxial Fan.

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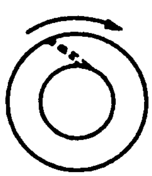
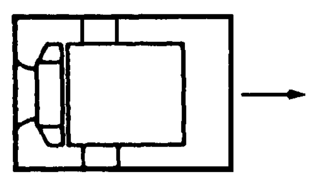
Other Fan Types

Two additional fan types that may be encountered are the tubular-centrifugal fan (Figure A-9), and the inline centrifugal duct fan (Figure A-10).

IMPELLER DESIGN



This fan usually has a wheel similar to the airfoil. Backwardly inclined or backwardly curved blade as described above. (However this fan wheel type is of lower efficiency when used in fan of this type.) Mixed flow impellers are sometimes used.



HOUSING DESIGN

Cylindrical shell similar to a vaneaxial fan housing except the outer diameter of the wheel does not run close to the housing. Air is discharged radially from the wheel and must change direction by 90 degrees to flow through the guide vane section.

Figure A-9. Tubular-Centrifugal Fan.

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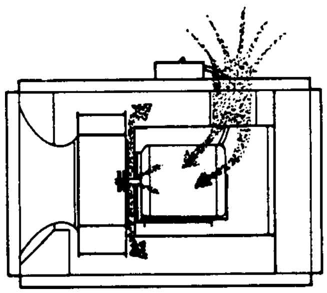


Superior aerodynamic performance is provided by deep venturi inlet combined with median-foil wheels. Casing sizes and internal baffling selected for optimum airflow. More air per dollar first-cost and more air per dollar of operating cost are provided by the high air handling efficiency. The same design details that produce maximum aerodynamic efficiency also assure quiet operation.

Figure A-10. Inline Centrifugal Duct Fan.

Source: Carnes Co., Verona, WI. Used with permission.

Fan Classes

Fans are classified according to certain construction features such as thickness of metal, type of bracing, etc. Fan classification is usually shown in the manufacturer's performance data. The TAB technician should be aware of the fan classification as this will affect whether or not the operating conditions of the fan can be altered in order to balance the system.

Fan Laws and Sizing

Shapes of performance curves for various fan types, and other information about static pressure, BHP, and rotations of the wheel per minute may be plotted on a fan curve (Figure A-11). Also, selection of fans to fit a system may be found by plotting on this curve.

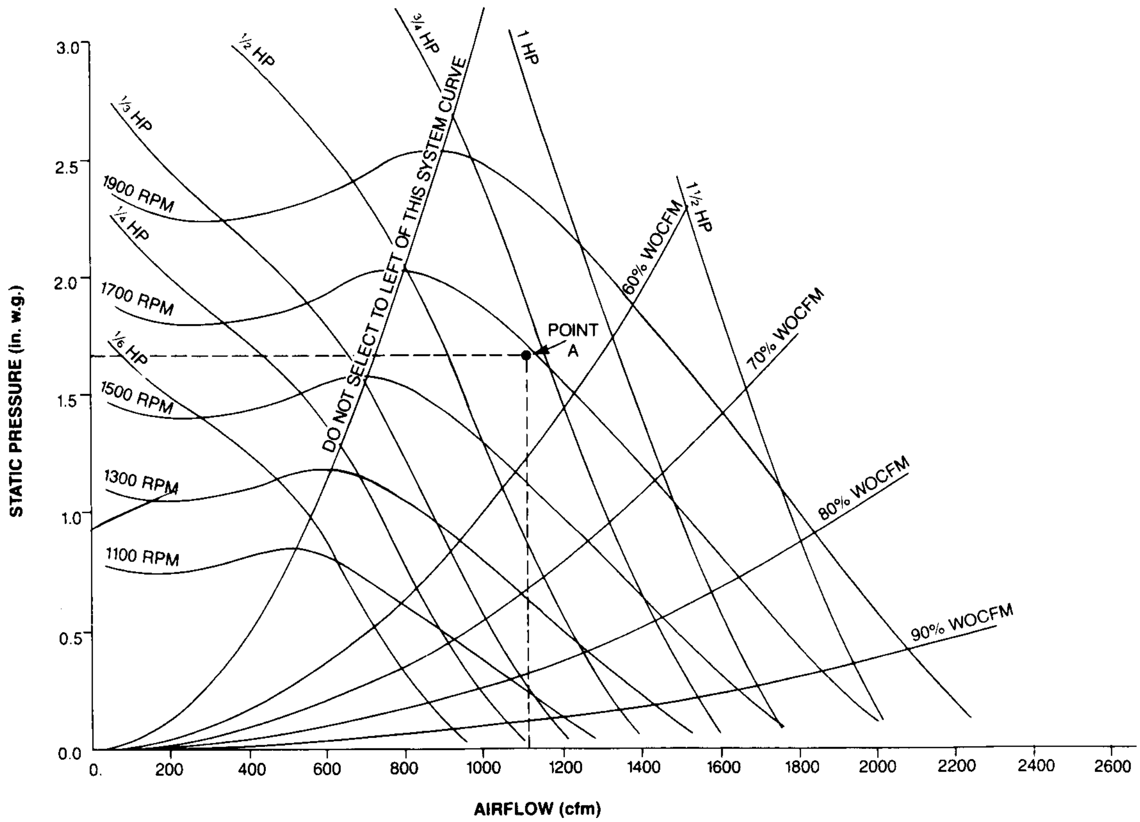


Figure A-11. Typical FC Fan Curves.

The following examples demonstrate applications of fan laws.

Fan Laws:

$$\frac{Q_2}{Q_1} = \frac{rpm_2}{rpm_1} \qquad \frac{bhp_2}{bhp_1} = \left(\frac{rpm_2}{rpm_1}\right)^3 \qquad \frac{P_2}{P_1} = \left(\frac{Q_2}{Q_1}\right)^2$$

where:

	Q	=	airflow (cfm)
	rpm	=	revolutions/minute
	P	=	system pressure (in. w.g.)
	bhp	=	brake horsepower

Example 1:

A fan must be speeded up to supply 13,000 cfm. The airflow is presently measured at 10,000 cfm at 2.0 in. w.g. static pressure. What will be the new fan speed, if the present fan speed is 660 rpm?

From $\frac{Q_2}{Q_1} = \frac{rpm_2}{rpm_1}$ then $rpm_2 = 660 \times \frac{13{,}000}{10{,}000} = 858\,rpm$

Example 2:

A duct system is operating at 2.0 in. w.g. with an airflow of 10,000 cfm. If the airflow is increased to 13,000 cfm without any other change, what is the new duct system pressure?

From $P_2 = P_1 \times \left(\frac{Q_2}{Q_1}\right)^2$ then $P_2 = 2.0 \times \left(\frac{13{,}000}{10{,}000}\right)^2$

$$P_2 = 2.0 \times (1.3)^2 = 3.38\,in.\,w.g.$$

From the fan curve in Figure A-12, 3.38 in. w.g. of static pressure at 13,000 cfm requires an estimated 860 rpm. (When using this equation, the system pressure can be in terms of either total pressure or static pressure.)

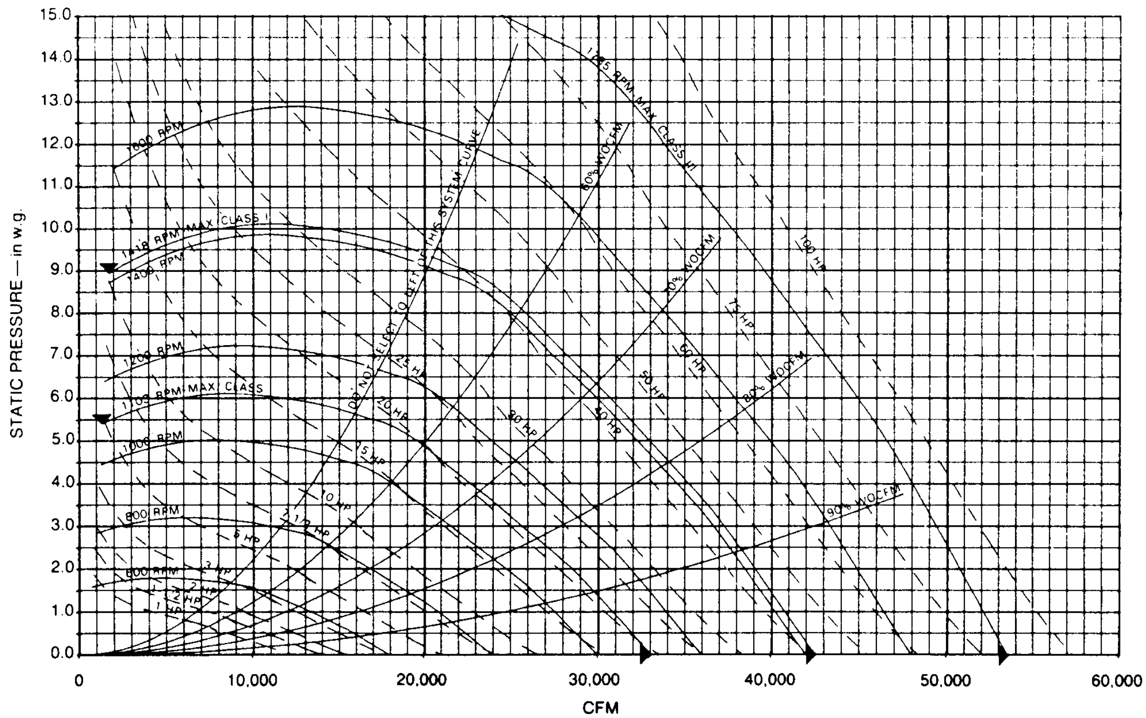


Figure A-12. Fan Curves.

Example 3:

The same system used in Example 2 has a 5 HP motor operating at 4.0 bhp. Find the bhp that would be required if the airflow was increased to 13,000 cfm.

From $bhp_2 = bhp_1 \times \left(\frac{Q_2}{Q_1}\right)^3$ then $bhp_2 = 4.0 \times \left(\frac{13{,}000}{10{,}000}\right)^3$

$$bhp_2 = 4.0 \times (1.3)^3 = 8.79\ bhp$$

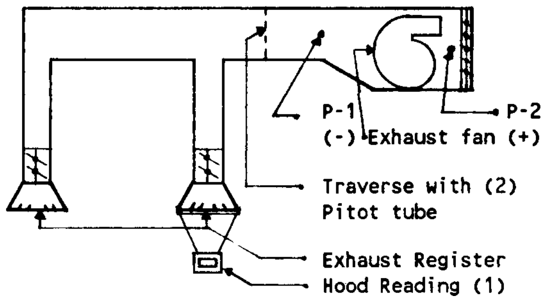
The 5 HP motor would be inadequate, and a 10 HP motor would be required.

The next three examples show how fans are sized. Before proceeding further, some terms used in the examples will be defined first:

SP	Static Pressure:	The normal force per unit area that would be exerted by the moving air on a balloon immersed in it if it were carried along by the air.
ISP	Internal Static Pressure:	The sum of the pressure drops across components inside air handling units such as filters, cooling and heating coils, hot and cool deck dampers, etc.
ESP	External Static Pressure:	The sum of the pressure drop across components external to the air handling unit such as terminal boxes, elbows, diffusers, volume dampers, and all other friction causing elements in the duct system.
SPs	Static Pressure (suction)	
SPd	Static Pressure (discharge)	
ISPcc	Internal Static Pressure (cooling coil)	
ISPhc	Internal Static Pressure (heating coil)	
ISPf	Internal Static Pressure (filters)	

Fan SP = SPd - SPs

Example 4: Single Zone Exhaust System



Find:

1. P-1 Neg. (Pressure required to move air from face of exhaust grill to P-1.)
2. P-2 Pos. (Pressure required to move air through discharge louver. P-2 will be 0 in. w.g. depending on discharge configuration.)
3. cfm (Determined by hood reading at exhaust grill and pitot tube traverse.)
4. Static Pressure (Taken with pitot tube.)
5. Fan Horsepower

P-1 = SPs = -.64 in. w.g.

P-2 = SPd = .44 in. w.g.

Fan SP = SPd - SPs = .44 - (-.64) = 1.08 in. w.g.

From hood and pitot tube readings, cfm was found to be 6,509

By interpolation on Table A-1, fan bhp = 2.47.

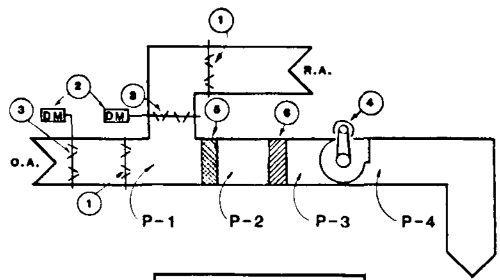
Table A-1. Fan Ratings.

VOL CFM	OUT VEL FPM	VEL PRES IN H₂O	0.125 S P RPM	0.125 S P BHP	0.250 S P RPM	0.250 S P BHP	0.375 S P RPM	0.375 S P BHP	0.500 S P RPM	0.500 S P BHP	0.625 S P RPM	0.625 S P BHP	0.750 S P RPM	0.750 S P BHP	0.875 S P RPM	0.875 S P BHP	1.000 S P RPM	1.000 S P BHP	1.250 S P RPM	1.250 S P BHP	1.500 S P RPM	1.500 S P BHP	1.750 S P RPM	1.750 S P BHP	2.000 S P RPM	2.000 S P BHP
2264	800	0.04	398	0.10	456	0.15	507	0.21	557	0.26	608	0.32	656	0.40	703	0.47	747	0.55								
2547	900	0.05	434	0.13	487	0.19	536	0.25	578	0.30	624	0.37	669	0.44	712	0.51	755	0.60	835	0.78						
2830	1000	0.06	472	0.17	519	0.23	566	0.29	608	0.36	645	0.42	686	0.49	727	0.57	767	0.65	843	0.83	916	1.03				
3113	1100	0.08	510	0.21	552	0.27	595	0.34	636	0.42	675	0.49	708	0.56	745	0.63	782	0.71	855	0.89	924	1.10	991	1.31		
3396	1200	0.09	549	0.26	587	0.33	627	0.40	666	0.48	702	0.56	738	0.64	768	0.71	802	0.79	870	0.97	936	1.17	999	1.39	1062	1.63
3679	1300	0.11	589	0.32	624	0.39	661	0.47	697	0.55	731	0.64	765	0.73	798	0.81	825	0.89	888	1.07	950	1.26	1012	1.48	1070	1.71
3962	1400	0.12	629	0.39	662	0.46	695	0.54	729	0.63	762	0.72	794	0.81	826	0.91	856	1.01	909	1.17	967	1.37	1026	1.58	1083	1.82
4245	1500	0.14	668	0.46	700	0.54	730	0.62	762	0.72	794	0.81	825	0.91	854	1.01	884	1.12	936	1.30	989	1.50	1043	1.71	1097	1.94
4528	1600	0.16	709	0.55	739	0.63	767	0.72	796	0.81	827	0.91	856	1.02	884	1.13	912	1.23	967	1.46	1013	1.64	1063	1.86	1114	2.09
4811	1700	0.18	749	0.65	778	0.74	805	0.83	832	0.92	860	1.03	888	1.14	915	1.25	942	1.36	994	1.59	1044	1.82	1087	2.01	1134	2.25
5094	1800	0.20	790	0.75	818	0.85	843	0.95	868	1.06	894	1.15	921	1.26	948	1.38	973	1.50	1023	1.74	1073	1.99	1115	2.21	1157	2.43
5377	1900	0.23	830	0.88	857	0.98	882	1.08	906	1.19	930	1.29	955	1.40	980	1.53	1005	1.65	1053	1.90	1100	2.16	1146	2.42	1185	2.64
5660	2000	0.25	872	1.01	897	1.12	921	1.23	944	1.33	966	1.44	989	1.56	1014	1.68	1038	1.81	1084	2.08	1129	2.34	1173	2.61	1217	2.89
5943	2100	0.27	913	1.18	937	1.27	960	1.39	982	1.50	1004	1.61	1025	1.73	1048	1.85	1071	1.99	1116	2.28	1160	2.54	1202	2.82	1245	3.12
6226	2200	0.30	954	1.32	977	1.44	999	1.56	1021	1.68	1042	1.80	1062	1.91	1083	2.04	1104	2.17	1148	2.46	1191	2.75	1231	3.04	1272	3.34
6509	2300	0.33	995	1.50	1017	1.62	1039	1.75	1059	1.87	1080	1.99	1100	2.12	1119	2.24	1139	2.38	1181	2.67	1222	2.97	1262	3.28	1301	3.58
6792	2400	0.36	1037	1.70	1057	1.82	1079	1.95	1099	2.08	1118	2.21	1137	2.34	1156	2.47	1175	2.60	1215	2.90	1255	3.20	1293	3.52	1331	3.84
7358	2600	0.42	1120	2.13	1139	2.26	1159	2.40	1178	2.55	1196	2.68	1214	2.82	1231	2.97	1248	3.10	1284	3.40	1321	3.72	1358	4.06	1393	4.40
7924	2800	0.49	1204	2.64	1221	2.78	1239	2.93	1257	3.08	1274	3.23	1291	3.38	1308	3.53	1324	3.69	1356	3.99	1389	4.32	1424	4.67	1458	5.03
8490	3000	0.56	1287	3.23	1303	3.38	1320	3.53	1337	3.70	1353	3.86	1370	4.02	1385	4.18	1401	4.34	1431	4.67	1461	5.00	1492	5.35	1525	5.73
9056	3200	0.64	1371	3.90	1386	4.06	1401	4.21	1417	4.39	1433	4.56	1448	4.74	1464	4.91	1478	5.08	1507	5.43	1535	5.77	1563	6.13	1593	6.51
9622	3400	0.72	1465	4.66	1469	4.82	1483	4.99	1498	5.16	1513	5.35	1528	5.54	1542	5.72	1556	5.91	1583	6.27	1611	6.64	1637	7.00	1664	7.39
0188	3600	0.81	1539	5.51	1552	5.68	1566	5.85	1579	6.04	1594	6.24	1608	6.43	1621	6.63	1635	6.82	1661	7.20	1687	7.59	1713	7.99	1737	8.37
0754	3800	0.90	1623	6.46	1636	6.64	1648	6.82	1661	7.01	1674	7.21	1688	7.42	1701	7.63	1714	7.84	1740	8.25	1764	8.65	1788	9.06	1813	9.48
1320	4000	1.00	1707	7.52	1719	7.70	1731	7.89	1743	8.09	1755	8.29	1769	8.52	1781	8.74	1794	8.96	1818	9.39	1841	9.80	1865	10.24	1888	10.68

Pressure class limits:

Class	Maximum RPM
I	1550
II	2140

Example 5: Single Zone System



COMPONENT LIST	
1	MANUAL OPPOSED BLADE BALANCING DAMPER
2	DAMPER MOTOR
3	MIXING DAMPER
4	SUPPLY FAN
5	FILTER
6	COOLING COIL

Find:

P-1 = ESPs = .53 in. w.g.

ISPf and ISPcc: pressure drops available from manufacturer's data

P-2 = ISPf + ESPs = .08 + .53 = .61 in. w.g.

P-3 = ISPcc + ISPf + ESPs = (-.18) + (-.08) + (-.61) = -.87 in. w.g.

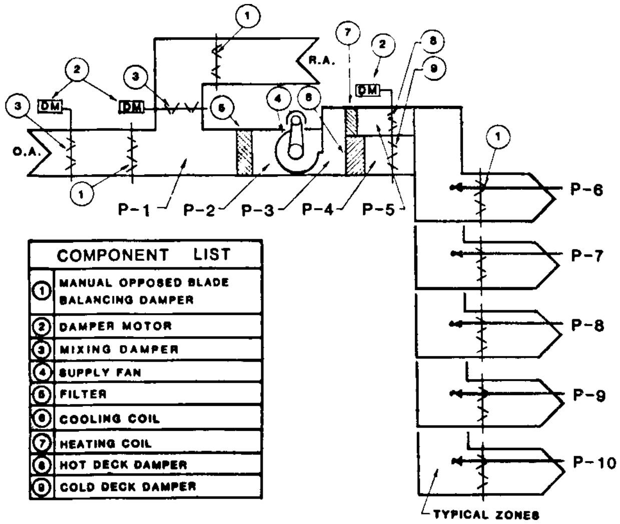
P-4 = ESPd = 1.15 in. w.g. (measured with pitot tube)

cfm = 7,924 (determined by hood and pitot readings)

Fan SP = (P-4) - (-P-3) = 1.15 - (-.85) = 2.00 in. w.g.

From Table A-1, fan bhp = 5.03.

Example 6: Multizone System



cfm = 10,188, amount of air required to supply zones 6-10

P-1 = -.40

P-2 = ISPf + (P-1) = (-.08) + (-.40) = -.48 in. w.g.

P-3 = 1.15 in. w.g.

P-4 = 1.15 - ISPcc = 1.15 - .15 = 1.00 in. w.g.

P-5 = 1.15 - ISPhc = 1.15 - .05 = 1.10 in. w.g.

In cooling mode (readings for P-6 through P-10 are obtained by pressure gauge measurements):

ESPsys6 = (P-4) - (P-6) = 1.0 - .32 = .68 in. w.g.

ESPsys7 = (P-4) - (P-7) = 1.0 - .25 = .75 in. w.g.

ESPsys8 = (P-4) - (P-8) = 1.0 - .30 = .70 in. w.g.

ESPsys9 = (P-4) - (P-9) = 1.0 - .10 = .90 in. w.g.

ESPsys10 = (P-4) - (P-10) = 1.0 - .20 = .80 in. w.g.

FAN SP = (P-3) - (P-2) = 1.15 - (-.48) = 1.63 in. w.g.

By interpolation on the fan rating table, fan bhp = 7.79.

The calculations show that P-9 has developed the largest static pressure required in the system. Therefore, fan size is based on the static pressure required at P-9, and use of the cooling coil (wet). All remaining systems require balancing with manual dampers.

Fan Curves Vs. System Curves

System resistance curves or system curves are a plot of cfm vs. static pressure in a system. This shows a graphical representation of the system's resistance to air flow. Each system will have its own system curve that is represented by a single line. This curve will remain unchanged until there is a change to the system, such as dirt or moisture buildup, or a change in position of the outlet dampers.

When the system curve and a fan performance curve are plotted together, the intersection of the two curves will be the operating point of that system. The figure below shows a typical system curve plotted with two fan curves. This example illustrates the effects of a 10 percent increase in fan speed without a change to the system itself. The operating point moves upward along the system curve resulting in an increase in both cfm and static pressure.

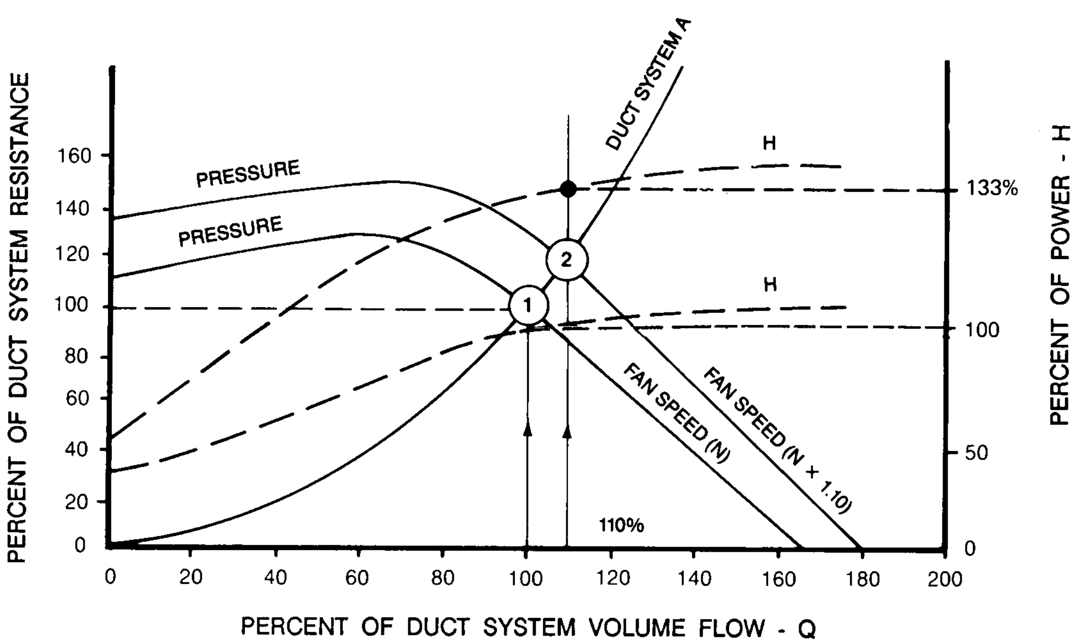


Figure A-13. Typical System Curve With Two Fan Curves.

Environmental Systems Technology, W. D. Bevirt, 1984. Reprinted with permission of the National Environmental Balancing Bureau.

However, if a change is made to the system that will shift the whole system curve to the right, the operating point will move downward along the fan curve. This results in an increase in cfm with less static pressure in the system.

The fan and system curves can be used to help troubleshoot different problems that may occur in the system. They can also be used to model the effects of different changes to the system or fan. This helps in predicting what changes will produce the best results. This method, however, does not produce exact results. Therefore, when searching for exact answers, the appropriate fan laws must be used.

Fan Discharge Control

The four common methods of controlling the effects of any fan: (1) discharge damper control, (2) inlet vane control, (3) variable pitch control, and (4) speed control. Figure A-14 shows the approximate power savings that can be obtained by reducing air quantities for the four methods of capacity control.

From a power consumption standpoint, variable speed motors and blade pitch control are the most efficient. Inlet vanes save some power, while discharge dampers throttling at the fan save little. From a first-cost standpoint, dampers are the least costly. Inlet vanes and blade pitch control follow, with variable speed motors being the most expensive.

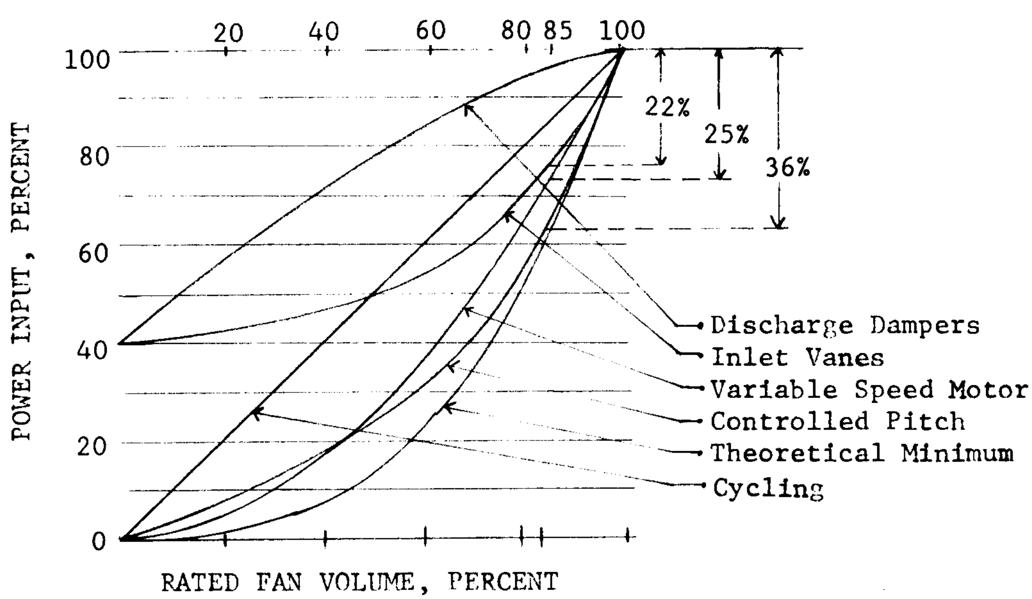


Figure A-14. Power Savings for Four Methods of Capacity Control.

Discharge Air Dampers

Discharge air dampers are installed to add resistance at the fan. The purpose of the discharge air damper arrangement is to create an excess pressure drop near the fan, thus permitting smaller pressure drops at the terminal units. The amount of air delivered to the terminal units depends on the pressure built up at the fan. When air delivery has to be reduced at the terminal unit, the terminal unit air dampers will throttle it down. It is impractical to have the terminal units serve as the sole means of throttling air supply. For example, terminal units should not have to throttle down 3 in. w.g. when all they were designed to throttle is 1 in. w.g. So, the pressure over and above what is actually needed at the terminal units is throttled down by the discharge air dampers before it even enters the rest of the system. Because of the initial throttling, there is less noise at the terminal units. With an initial pressure drop at the fan, there is more ductwork to aid in sound attenuation prior to discharge.

Sizing of discharge air dampers should be done with great care. There are many rules-of-thumb, but the recommended procedure is to size the discharge air dampers for a wide open pressure drop of from 7 to 10 percent of the system pressure.

Because discharge air dampers waste horsepower, they should not be used to control VAV systems if operational efficiency is desired. Figure A-11 shows that their efficiency is not adequate to warrant their use for economic operation purposes.

Variable Inlet Vanes

The most commonly used method of controlling fan capacity on VAV systems is variable inlet vanes. Inlet vanes, often referred to as pre-rotation vanes, cause the air to swirl before it encounters the fan wheel. The fan wheel cannot "grip" the air as well and consequently, capacity is reduced more efficiently than with discharge damper control. Excess pressure is not created and wasted. Figure A-15 shows examples of an inlet vane type system. The fan inlet vanes are positioned by an actuator in response to a signal received from the system static pressure receiver-controller.

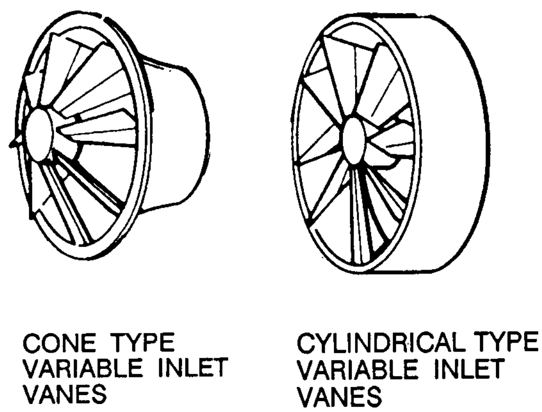


Figure A-15. Examples of Inlet Vane System.

Modification from AMCA, Publication 201-90. Used with permission.

The static pressure transmitter, through the receiver-controller and actuator, repositions the inlet vanes to maintain a relatively constant duct pressure at the point of sensing. As the terminal unit dampers throttle, the characteristic curve (resistance curve) shifts. The static pressure transmitter senses this shift, and throttles the inlet vanes accordingly. The fan curve is shifted, and a new operating point is established. The new operating point will depend on where the terminal units that are being throttled are located in the system. Fan discharge pressure will not remain constant since the location of the pressure transmitter is at the end of the system.

Care must be exercised in selecting the fan. It is important for the fan to be able to be throttled to the near minimum flow required without becoming unstable. Systems that can be throttled to near shut-off must often be equipped with a fan bypass to permit a minimum flow through the fan at all times.

Variable Pitch Blades

Variable pitch axial-flow fans deliver an amount of air in accordance with the pitch of the fan blades. As more or less air is needed in the system, an actuator positions the pitch of the fan blades accordingly. The positioning of the blades is similar to the positioning of the inlet vanes. However, the fan is always spinning while inlet vanes remain stationary. The degree to which the blades are pitched determines how much air can be “gripped” and passed on into the system.

Variable Speed Drives

Various ways to control fan speed include variable speed motors, magnetic couplers, and fluid drive systems. Fluid drive units use hydraulic fluid for transmitting power. Magnetic coupling models use interacting magnetic fields to transmit power. Another method is to use exhaust steam, when readily available, to drive the fan with a steam turbine. The most common method, however, is motor speed control.

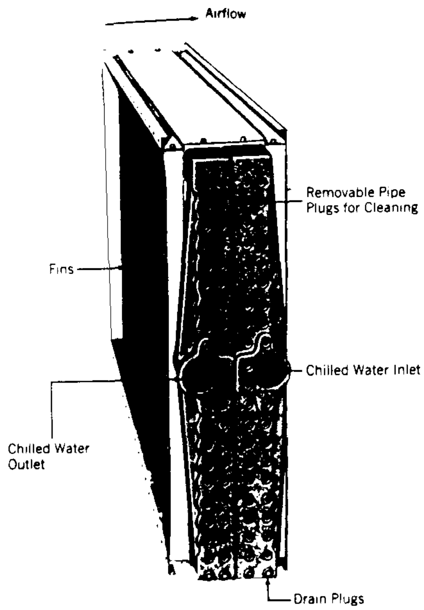
Adjustable speed drives (ASDs) are devices that vary the speed of a motor to match the load being put on the motor. Many types of ASDs are available, including mechanical (eddy current drives, variable-ratio pulley, and hydraulic drives), direct current (DC motors), and electronic. Although mechanical drives and DC motors have been applied extensively in industrial settings, they are seldom used in commercial buildings for economic or technical reasons. The mechanical variable-ratio pulley is applicable to commercial buildings (from 5 to 125 horsepower), but space requirements and mechanical problems usually make commercial applications impractical. DC motors comprise a mature technology, but they are expensive and have a reputation for high maintenance costs. The electronic load-commutated inverter has also been used in industry, but it is not an energy-conscious choice for commercial buildings.

Frequency operated adjustable speed drives are most commonly used for variable fan speed control today. Fan motor speed control is accomplished by mechanically, electrically, or hydraulically varying fan rpm in response to the signal from the pressure transmitter in the system. The transmitter/receiver-controller arrangement varies fan speed to maintain a constant duct pressure at the transmitter.

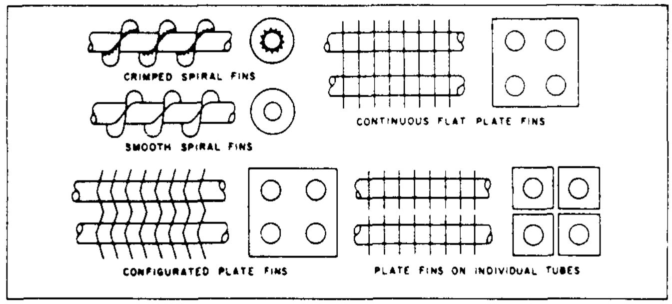
Heating and Cooling Coils

Heating and cooling coils are simply heat exchangers between a heating or cooling medium and the air stream. Heating mediums available for heating coils are steam, hot water, or electricity. Steam and hot water coils consist of banks of copper tubing surrounded by sheets of corrugated fins that guide the air toward the tubing to maximize the heat transfer surface in contact with the air. Figure A-16 is a four-row cooling coil with double-tube serpentine circuiting.

A boiler is required to produce the steam or hot water for these types of coils, which in turn requires piping from the boiler to the AHU. Thus, the steam or hot water coils are economical only for medium and large-size installations, and become a more and more attractive option as the number of AHUs served increases.



Four-row cooling coil with double-tube serpentine circuiting. *(Courtesy The Trane Company. LaCrosse. Wis.)*



TYPES OF FIN-COIL ARRANGEMENTS

Figure A-16. Four-row Cooling Coil With Double-tube Serpentine Circuiting.

A preheat coil is used to raise the outside air temperature to 55 °F before it gets to the AHU when the outside air is below 32 °F. Another special heating coil, known as a reheat coil, is sometimes placed downstream from the cooling coil for applications where humidity control is critical, such as in hospitals, laboratories, and some industries. The cooling coil dehumidifies the air to a precise point, and then the reheat coil warms it back up to the necessary temperature.

Cooling coils may carry either chilled water or refrigerant gas. The arrangement could consist of a single coil section or a number of individual coil sections built up into banks. The coil assembly will usually include an air cleaning means to protect the coil from accumulation of dirt, and to keep dust and foreign matter out of the conditioned space. Cooling coils for water or for volatile refrigerants most frequently have aluminum fins and copper tubes, although copper fins on copper tubes, and more rarely, aluminum fins on aluminum tubes are also used.* The diameter of the tubes can vary from ¼ to 1 in. The fin spacing should be chosen for the duty to be performed, with special attention being paid to air

* Approximately 90 percent of common HVAC coils are copper tube with aluminum fins due to cost, weight, and environment. There is always a difference in heat transfer between metals, but it is an insignificant amount.

friction, possibility of lint accumulation, and (especially at lower temperatures) the consideration of frost accumulation. The fins are generally spaced 3 per inch up to 14 per inch.

Coil capacity can be controlled without using a control valve. In Figure A-17, the face and bypass damper is actually two dampers linked together. When full heating (or cooling) is required, the damper section in front of the coil face is full open and the damper section in the bypass is shut. All the air passes through the coil. As the room demand for the coil capacity diminishes, a room thermostat signals a motor to move the face dampers toward the closed position while moving the bypass dampers to a more open position.

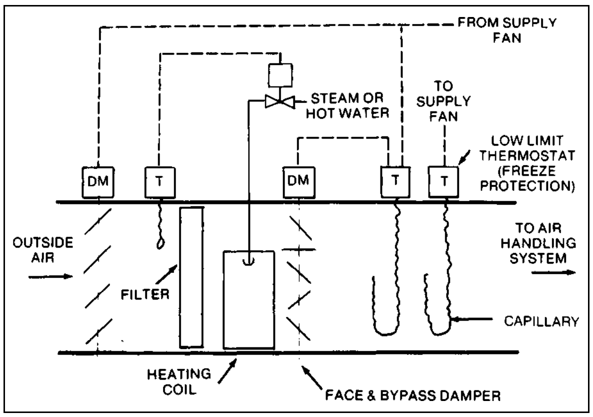


Figure A-17. Coil Capacity Controlled by Two Dampers Linked Together.

Reprinted with permission from the 1995 *ASHRAE Applications Handbook.*

Direct Expansion Coil Circuiting for Variable Air Volume Systems (DX/VAV)

Direct expansion (DX) cooling coils are thermodynamically complex. Sensible and latent heat exchange occur on the inside and outside surfaces of the coils. Mixed or all outside air flowing across the coils is both sensibly and latently cooled; this causes moisture to condense on the coil surface. Inside the refrigerant tube (cooling coil), sensible and latent heating occur as refrigerant is evaporated, and superheated. Superheated vapor (refrigerant vapor) is characterized by the actual pressure of the vapor being lower than the saturation pressure at a given temperature, and the actual temperature of the vapor being higher than the saturation temperature of the vapor. Superheating occurs beyond the saturated vapor phase, and it is very important to keep the refrigerant in this superheated phase until it gets to the compressor. If a mix between saturated vapor and saturated liquid exists in the line prior to entering the compressor, liquid refrigerant will dilute compressor oil, robbing the compressor of vital lubrication.

If left in the compressor, liquid refrigerant can cause oil foaming when the compressor restarts, and foaming hampers delivery of oil to critical crankshaft and journal bearings.

Sensible and latent heat do not occur in a linear fashion along the outside of the coil surface, so the coil may not be uniformly wetted. On the inside surface, as liquid and vapor refrigerant (mix) are forced through the tube, a pressure drop results and lowers the refrigerant boiling point.

Chilled water gets warmer as it goes down the tubes, but the refrigerant actually cools in this process. Only after all the refrigerant is completely evaporated can superheating begin to warm the vapor. The point where superheating begins also affects the coil capacity and performance because it affects the pressure drop which is not always uniform.

Even though excellent refrigerant piping practices are followed in most installations of DX/VAV split systems, some systems become very unstable, especially at part load conditions. Some of the common problems that have occurred include erratic thermal expansion valves, continued compressor cycling, coil frosting, poor temperature control, and the return of liquid refrigerant to compressors. In some severe cases, compressors can be destroyed. Considering all of these problems, suppose a system was designed with identical equipment, employing similar controls and prudent piping and installation practices, and problems still occur, what separates good and poor DX/VAV systems? The difference could be the internal circuiting of the DX cooling coil.

A distributor is the device that uniformly transfers or distributes refrigerant from the thermal expansion valve to each circuit. There is only one distributor for each expansion valve. Since the expansion valve bulb senses the degree of superheat for all the circuits on that distributor, it is unaware of any differences between circuits. Therefore, the refrigerant must be distributed uniformly to all circuits.

When the distributor's maximum MBh/circuit is insufficient to meet design load, or minimum compressor loading is less than the distributor's minimum MBh/

circuit, the coil must be divided into separate sections or "splits."* Each section is fed by one distributor.

The coil can be split in three ways to satisfy the needs of a specific design. This is where the engineer must use a knowledge of the environment in which the system will operate and select accordingly. Following is a brief description of each kind of split system and where they are best used. The three systems are horizontal or face split coils, vertical or row split coils, and intertwined coil circuiting (Figure A-18).

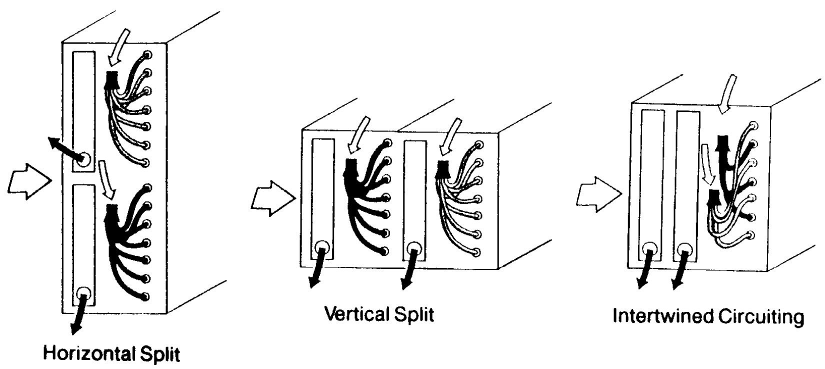


Figure A-18. Split Systems.

Source: *ASHRAE HVAC Systems and Equipment Handbook, 1992.* Used with permission.

Horizontal split/face split coils. Face splits are preferred for VAV applications as adequate superheat is assured for each coil section. This is true even at part load because all coil sections receive unconditioned air.

When the system changes to handle part load conditions, the mass flow rates decrease, and a solenoid valve upstream of one of the thermal expansion valves is closed. All the refrigerant now flows through the remaining distributor. With the horizontal split coils, the mass flow at the open distributor doubles, and it singly maintains loading above the minimum MBh/circuit. The inactive coil section bypasses unconditioned air that may cause problems for systems that use 100 percent outside air at part load. Face split coils are not recommended for 100 percent outside air applications in humid climates. At part load, this type of

* The minimum mass flow rate at which the distributor can provide uniform distribution to all circuits is expressed as MBh/circuit, and is a function of suction pressure. The maximum MBh/circuit is a function of distributor geometry and suction pressure. Stable performance is assured only when full load and part load MBh/circuit remains within this allowable range.

split coil can pose a potential problem because the system controllers try to keep the coil leaving air temperature constant. With a major part of the coil inactive, the active coil section must cool air to very low temperatures to maintain 55 °F leaving air. For example, if entering air is at 80 °F and half the coil is inactive, the active coil section must cool to 30 °F to achieve an average discharge of 55 °F. These conditions are conducive to coil frosting, and liquid refrigerant being returned to the compressor.

Vertical/row split coils. This type of split coils is not recommended for VAV applications. Compressor cycling can occur in row split coils. This is not bad in itself because reciprocating compressors can be used and will tolerate the cycling. However, compressor cycling upsets superheat control. Excessive superheat hampers compressor motor and discharge valve cooling. In the absence of superheat, liquid refrigerant is returned to the compressor. Reciprocating compressors are designed to tolerate brief periods of liquid in the suction line, but the combination of reduced airflow and humidity, and sustained initial temperature difference causes the upstream two row coil to produce a colder than anticipated leaving air temperature. The colder temperature leaving the upstream coil can hamper the ability of the downstream coil to provide adequate superheat. This superheat loss can occur for an extended period of time. If the loss of superheat lasts longer than the cycle rate of the VAV discharge air temperature controller, the compressor will likely fail.

Intertwined coil circuiting. The pitfalls of row and face split coils can be avoided by using intertwined coil circuiting. It provides more active fin surface at part load, and improved superheat capabilities at all load conditions.

At part load, the coil behaves like a coil with substantially greater fin surface, but without the penalty of higher airside pressure drop. By increasing the active fin surface at part load, the potential for coil frosting is reduced while maintaining excellent dehumidification. Superheat is not lost at part load conditions, and stabilizes quickly after a change in compressor or capacity.

Intertwined coil circuiting may require additional distributors and thermal expansion valves in some circumstances, but the DX/VAV stability at part load is worth the additions. Intertwined coils are best for almost all DX/VAV split system applications. They have been used extensively in packaged unitary equipment including rooftop and self-contained air conditioners in VAV applications.

If intertwined coils are not suitable, face split coils are acceptable if used with some type of supply air reset at part load. The row split should be avoided in VAV applications, but they are preferred in 100 percent outside air applications.

Filters

Filters are important for providing a comfortable and healthy air supply to the occupants, reducing dust deposits on room surfaces, and keeping interiors of HVAC system components clean. Filters and other air cleaning devices are available in four general types for four general purposes: (1) typical commercial filters to remove visible particles of dust, dirt, lint, and soot, (2) electrostatic filters to remove microscopic particles such as smoke and haze, (3) activated charcoal to destroy odors, and (4) ultraviolet lamps or chemicals to kill bacteria.

Both throwaway and cleanable filters are available. Throwaway filters are generally standard on smaller AHUs (less than 10,000 cfm). The standard commercial grade filters remove about 75 to 85 percent of the particles in the air. In hospitals and laboratories where a high degree of cleanliness is called for, high-efficiency filters are used.

Three different physical arrangements for filters in air handlers are flat, offset, and V-bank. The latter two provide more filter face area and, therefore, a lower face velocity across the filter. The maximum allowable face velocity for throwaway filters is 300 fpm versus a maximum of 500 fpm for cooling coils and 800 fpm for heating coils.

Filter banks may contain many throwaway filters that slide into the filter section channels on the top and bottom of each row of filters. The easiest way to change filters in large systems is to open access doors on each end of the filter bank. New filters are pushed into one end, while the used filters fall onto the floor at the other end. When the filter bank is accessible from only one end, a strip is used in the bottom channel. As the strip is pulled out, the farthest filter from the access door is pulled, pushing all the other filters in that line ahead of it (Figure A-19).

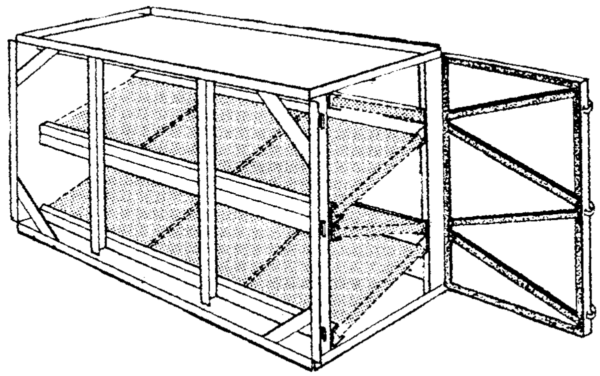


Figure A-19. Changing Filters.

Source: NAFA Guide to Air Filtration, 1993. Used with permission.

When dust loadings are expected to be quite heavy, the labor cost to continually replace filters can become prohibitive. Automatic filter changing can be done by using a roll filter (Figure A-20). The filter is advanced a few inches at a time, exposing new filter media at one end and rolling up dirty media at the other end. The advance of the filter is based on either a timer or a pressure-drop reading across the media. The latter is better because it exposes new media based on how much dirt the existing media has collected rather than on how long it has been in place.

For very critical jobs, a bag filter (Figure A-21) provides an extremely high cloth area, allowing the air to move through the filtering media very slowly. These are sometimes referred to as HEPA filters, which stands for high-efficiency particulate arrest. They are expensive to replace, and should be used with a less expensive throwaway filter upstream to filter out the larger size particulates.

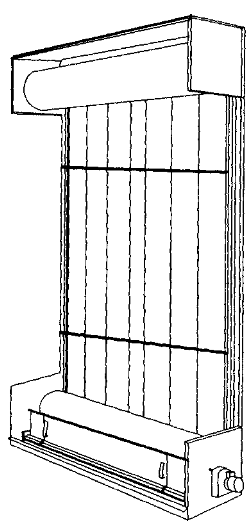


Figure A-20. Roll Filter Exposes Clean Media While Rolling Up the Dirty Media.

Source: NAFA Guide to Air Filtration, 1993. Used with permission.

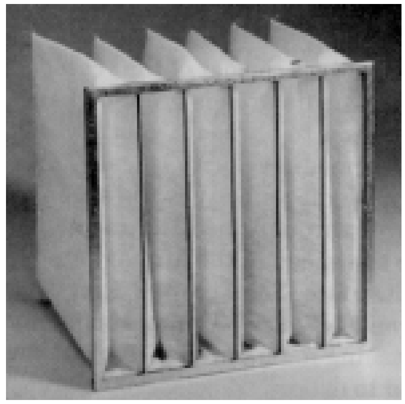


Figure A-21. Bag Filter Provides High Efficiency by Using Low Velocity Through the Filter Media.

Source: NAFA Guide to Air Filtration, 1993. Used with permission.

Mixing Box

A mixing box section (Figure A-22) is a convenient way to bring return air and outside air into the air handler.

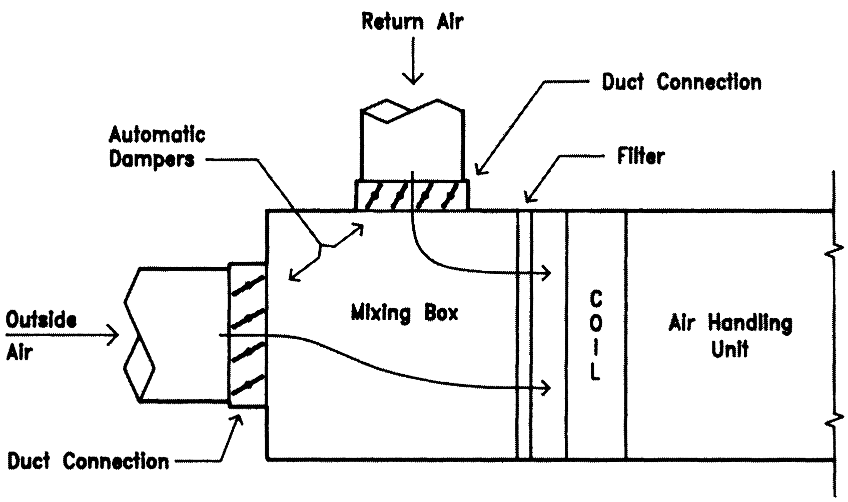


Figure A-22. Mixing Box for Return Air and Outside Air.

Source: Colen 1990. Used with permission of R.S. Means Company, Inc.

A damper is provided for each air stream to allow the controls technician to balance the percentage of outside air versus return air. The dampers may be either parallel blade or opposed blade (Figure A-23). Parallel blade damper sections are less expensive. Opposed blade damper sections provide better control characteristics.

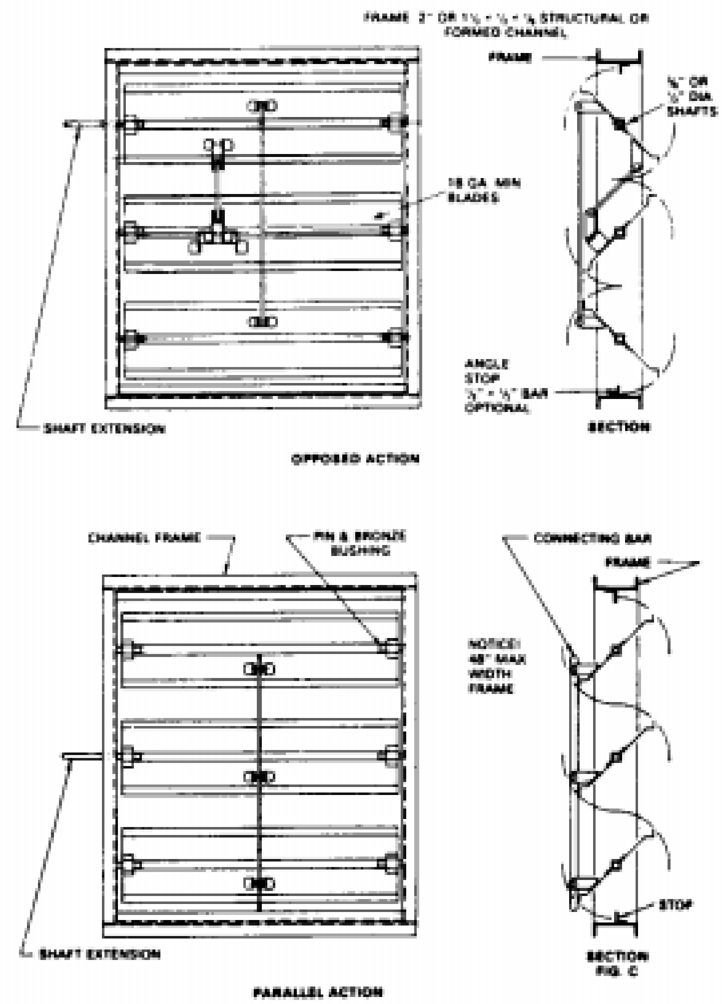


Figure A-23. Opposed and Parallel Blade Dampers.

SMACNA *HVAC Duct Construction Standards - Metal and Flexible*, 2nd Ed., 1995. Used with permission.

Humidification

Air leaving the central system of a VAV system is usually at 55 °F. Placing a humidifier in the air handling unit just before the cooling coils in a draw through setup would defeat its purposes for humidification. The steam would condense and drip, causing puddles in the AHU. If it were a rooftop system, this could lead to more serious problems.

Most humidifiers should be placed at least 10 ft from the AHU, if even this close. Since the VAV system has cool air and uses VAV boxes that are obstructions, it is best to place a humidifier in the duct after these boxes.

Humidification can be accomplished by direct injection of steam into the air stream, vaporizing water from a pan by heating it, passing air through a moist porous pad, or by spraying water from a nozzle into the air stream.

An example of a humidifier is the single-tube or Mini-Bank* multi-tube humidifier. These are specifically designed for application in hospital surgery rooms, intensive care units, delivery rooms, clean rooms, and where rapid steam absorption (in cool air) is required. If large ducts are used, the Maxi-Bank* may be used as it has an instantaneous total absorption within three feet of the tube bank, in any air temperature, and up to 50 percent relative humidity.

For a more detailed discussion on humidification, please refer to the Appendix A Annex.

* A commercial product of the "DRI STEEM" Humidifier Company, Hopkins, Minnesota.