

Customer Name **Sample Application**  
 Location **Anywhere USA**  
 Baghouse Reference



### Calculator for Profit From Retained Product

acfm 207,110  
 temp (deg F) 250  
 Dew point (deg F) 65  
 Humidity Ratio {0 for no moisture}  
 density factor 0.75  
 airflow (dscfm) 131,654

Measured outlet emissions 0.0034 gr/dscf  
 3.83 lb/hr

\$ Profit per pound \$0.10

Process Load		Cyclone Efficiency	Baghouse Load		Fabric efficiencies		Retained Product (lbs/hr)		Profit per month
(tons/hr)	(lbs/hr)		(lbs/hr)	gr/dscf	Current	BHA-TEX	Current	BHA-TEX	
12	24000	75.00%	6000	5.31697	99.94%	100.00%	5996.17	5999.7	\$254.16
		77.50%	5400	4.78527	99.93%	100.00%	5396.17	5399.73	\$256.32
		80.00%	4800	4.25357	99.92%	100.00%	4796.17	4799.76	\$258.48
		82.50%	4200	3.72188	99.91%	100.00%	4196.17	4199.79	\$260.64
		85.00%	3600	3.19018	99.89%	100.00%	3596.17	3599.82	\$262.80
		87.50%	3000	2.65848	99.87%	100.00%	2996.17	2999.85	\$264.96
		90.00%	2400	2.12679	99.84%	100.00%	2396.17	2399.88	\$267.12
		92.50%	1800	1.59509	99.79%	100.00%	1796.17	1799.91	\$269.28
		95.00%	1200	1.06339	99.68%	100.00%	1196.17	1199.94	\$271.44

**Average** \$262.80

**NOTES:**

1. If customer does not have a cyclone, enter 0% efficiency.

CFM		OV	20"SP		22"SP		24"SP		25"SP		26"SP		27"SP		28"SP		29"SP		30"SP		31"SP		32"SP			
			RPM	BHP	RPM	BHP																				
9280 9860 10440 11020		3200	1467	43.5	1535	48.3	1600	53.1	1634	55.7	1666	58.3	1696	60.7	1727	63.3	1756	65.8	1787	68.5	1814	70.9	1843	73.5	1843	73.5
		3400	1471	45.7	1538	50.6	1605	55.8	1636	58.3	1666	60.8	1697	63.4	1726	65.9	1756	68.6	1784	71.1	1813	73.8	1843	76.6	1843	76.6
		3600	1477	47.9	1543	53.0	1607	58.2	1639	60.9	1669	63.5	1698	66.0	1728	68.8	1759	71.6	1785	74.1	1815	77.0	1843	79.8	1843	79.8
		3800	1483	50.3	1549	55.5	1612	60.8	1642	63.5	1673	66.2	1703	69.0	1731	71.6	1760	74.4	1790	77.4	1819	80.2	1845	82.9	1845	82.9
11600 12180 12760 13340		4000	1490	52.6	1554	57.9	1618	63.5	1648	66.3	1677	69.0	1708	71.9	1737	74.8	1764	77.5	1793	80.4	1819	83.2	1847	86.1	1847	86.1
		4200	1496	54.9	1561	60.5	1623	66.2	1654	69.1	1682	71.8	1713	74.9	1740	77.7	1769	80.6	1798	83.7	1826	86.7	1852	89.6	1852	89.6
		4400	1505	57.5	1568	63.1	1631	69.1	1660	72.0	1689	74.9	1718	77.9	1746	80.9	1776	84.0	1801	86.8	1830	90.0	1857	93.1	1857	93.1
		4600	1515	60.2	1577	66.0	1637	71.9	1667	74.9	1696	77.9	1726	81.1	1753	84.1	1780	87.2	1809	90.4	1834	93.3	1862	96.6	1862	96.6
13920 14500 15080 15660		4800	1525	62.9	1586	68.8	1646	75.0	1675	78.0	1704	81.2	1731	84.2	1761	87.5	1787	90.6	1814	93.8	1840	96.9	1867	100	1867	100
		5000	1533	65.5	1597	71.9	1654	78.0	1683	81.2	1711	84.3	1741	87.7	1767	90.8	1794	94.0	1822	97.4	1847	100	1875	104	1875	104
		5200	1544	68.5	1605	74.8	1664	81.3	1694	84.6	1720	87.7	1748	91.0	1775	94.2	1803	97.7	1830	101	1856	104	1882	108	1882	108
		5400	1555	71.4	1616	78.0	1673	84.5	1702	87.8	1731	91.3	1758	94.6	1786	98.1	1812	101	1838	105	1864	108	1889	112	1889	112
16240 17400 18560 19720 20880		5600	1566	74.4	1628	81.3	1685	88.1	1713	91.4	1741	94.9	1767	98.2	1794	102	1821	105	1846	109	1871	112	1898	116	1898	116
		6000	1589	80.8	1649	88.0	1706	95.0	1734	98.7	1761	102	1788	106	1812	109	1841	113	1864	116	1891	120				
		6400	1614	87.6	1673	95.1	1729	102	1755	106	1782	110	1810	114	1837	118	1860	121	1885	125						
		6800	1640	94.9	1698	103	1753	110	1780	114	1806	118	1832	122	1857	126	1882	130								
7200	1668	103	1725	111	1779	119	1804	123	1831	127	1856	131	1881	135												

CFM		OV	20"SP		22"SP		24"SP		25"SP		26"SP		27"SP		28"SP		29"SP		30"SP		31"SP		32"SP			
			RPM	BHP	RPM	BHP																				
11808 12546 13284 14022		3200	1300	55.3	1362	61.5	1418	67.5	1448	70.8	1476	74.0	1503	77.1	1531	80.4	1557	83.6	1583	86.9	1608	90.1	1633	93.4	1633	93.4
		3400	1304	57.9	1364	64.2	1422	70.8	1450	74.0	1476	77.1	1504	80.5	1530	83.7	1556	87.1	1584	90.7	1607	93.7	1633	97.3	1633	97.3
		3600	1309	60.9	1367	67.2	1424	73.9	1453	77.3	1480	80.6	1505	83.9	1532	87.3	1559	91.0	1585	94.5	1609	97.8	1634	101	1634	101
		3800	1315	63.8	1373	70.5	1429	77.3	1455	80.6	1483	84.1	1509	87.6	1534	91.0	1560	94.5	1587	98.3	1612	102	1635	105	1635	105
14760 15498 16236 16974		4000	1320	66.8	1377	73.6	1434	80.7	1461	84.2	1487	87.7	1514	91.4	1539	95.0	1564	98.5	1589	102	1615	106	1639	110	1639	110
		4200	1326	69.8	1384	76.9	1439	84.1	1466	87.8	1493	91.5	1518	95.1	1542	98.7	1568	102	1594	106	1618	110	1641	114	1641	114
		4400	1334	73.0	1391	80.4	1446	87.8	1472	91.5	1497	95.1	1523	98.9	1548	103	1574	107	1596	110	1622	114	1646	118	1646	118
		4600	1343	76.4	1398	83.8	1452	91.5	1477	95.1	1503	99.0	1529	103	1553	107	1578	111	1603	115	1625	119	1650	123	1650	123
17712 18450 19188 19926		4800	1352	79.9	1405	87.4	1459	95.3	1484	99.1	1511	103	1534	107	1561	111	1584	115	1608	119	1631	123	1655	127	1655	127
		5000	1359	83.3	1415	91.3	1466	99.1	1492	103	1517	107	1543	111	1566	115	1590	119	1615	124	1637	128	1662	132	1662	132
		5200	1369	87.0	1423	95.0	1475	103	1501	107	1524	111	1549	116	1573	120	1598	124	1622	128	1645	133	1668	137	1668	137
		5400	1378	90.7	1433	99.1	1483	107	1508	112	1534	116	1558	120	1583	125	1606	129	1629	133	1652	138	1675	142	1675	142
20664 22140 23616 25092 26568		5600	1388	94.5	1443	103	1494	112	1518	116	1543	121	1566	125	1590	129	1614	134	1636	138	1659	142	1682	147	1682	147
		6000	1409	103	1462	112	1512	121	1537	125	1560	130	1584	134	1606	139	1632	144	1652	148	1676	153				
		6400	1433	112	1483	121	1532	130	1555	135	1580	140	1605	145	1628	150	1649	154	1671	159						
		6800	1454	121	1505	131	1554	140	1578	145	1601	150	1624	155	1646	160	1668	165								
7200	1478	131	1529	141	1576	151	1599	156	1623	161	1645	166	1668	172												

CFM		OV	20"SP		22"SP		24"SP		25"SP		26"SP		27"SP		28"SP		29"SP		30"SP		31"SP		32"SP			
			RPM	BHP	RPM	BHP	RPM	BHP	RPM	BHP	RPM	BHP	RPM	BHP												
14624 15538 16452 17366		3200	1161	68.1	1215	75.6	1268	83.4	1292	87.2	1318	91.2	1342	95.1	1367	99.3	1391	103	1415	107	1437	111	1460	116	1460	116
		3400	1165	71.6	1218	79.2	1269	87.1	1294	91.1	1320	95.4	1342	99.2	1368	104	1390	107	1415	112	1436	116	1460	120	1460	120
		3600	1168	74.9	1222	83.2	1272	91.2	1298	95.5	1322	99.7	1345	104	1370	108	1392	112	1416	117	1437	121	1460	125	1460	125
		3800	1173	78.5	1226	86.8	1276	95.3	1300	99.5	1325	104	1349	108	1371	113	1395	117	1417	121	1439	126	1460	130	1460	130
18280 19194 20108 21022		4000	1179	82.3	1231	90.8	1280	99.5	1304	104	1328	108	1350	113	1374	117	1398	122	1418	126	1442	131	1464	136	1464	136
		4200	1184	86.0	1236	94.9	1284	104	1309	108	1331	113	1354	117	1378	122	1401	127	1422	131	1444	136	1465	141	1465	141
		4400	1190	89.9	1241	99.0	1290	108	1313	113	1336	117	1360	122	1382	127	1404	132	1426	136	1447	141	1469	146	1469	146
		4600	1198	94.1	1248	103	1296	113	1318	117	1341	122	1364	127	1387	132	1409	137	1430	142	1452	147	1473	151	1473	151
21936 22850 23764 24678		4800	1206	98.4	1254	108	1302	117	1324	122	1348	127	1370	132	1392	137	1414	142	1435	147	1457	152	1477	157	1477	157
		5000	1212	102	1260	112	1309	122	1332	127	1353	132	1377	137	1398	142	142									

## SYSTEM CALCULATION

### INTRODUCTION

A fan system is any combination of ductwork, hoods, filters, louvers, collectors, etc., that relies upon a fan to produce airflow. When the air moves past each of these components, resistance is created which must be considered in system calculations. It is also important to remember that fans are rated independently of a system and that fan performance will vary depending upon the accuracy of the system calculations. This Engineering Letter will explain some of the basic fundamentals of system design and calculation.

### SYSTEM DESIGN

The purpose of the system will dictate the design criteria to be used. Generally they will fall into one of the following two categories:

**Velocity** is typically the primary consideration in dust collection, dilute pneumatic conveying, fume removal, and contaminant applications. In these applications, a capture velocity is required to redirect the flow of airborne materials into the duct system. In addition, a minimum conveying velocity is necessary to maintain the flow of the materials within the system.

Given these velocity requirements, system components can be selected to maintain the appropriate air volume and required velocity through the system.

**Air Mass** is the primary consideration in many drying, combustion process, and ventilating applications. These applications generally require a certain amount of air mass, usually measured in pounds of air, to support the application. Because fan manufacturers publish fan capacities in actual cubic feet per minute (ACFM), the mass of air required must be converted from standard cubic feet per minute (SCFM) to ACFM.

The velocity through a system can be determined once the ACFM is known. The relationship between velocity and airflow is defined by the equation:

$$Q = VA$$

where: Q = ACFM

V = velocity in lineal feet per minute

A = cross-sectional area in square feet

To determine the airflow requirement, the cross-sectional area is multiplied by the required velocity.

System design is really a matter of defining the required work in terms of volume or velocity and then sizing and selecting the necessary system components to accomplish that work. Of course, this must be done within the economic and space constraints of the installation.

### DETERMINING SYSTEM RESISTANCE

System resistance is the sum of the resistance through each component within the system. The system depicted in Figure 1 may appear complex, but dealing with each component separately provides an orderly process for determining the overall resistance.

### HOOD LOSS

To determine hood or entrance losses, resistance calculations must be made for both the acceleration loss and the entry loss.

Since the air or atmosphere surrounding the hood must be accelerated from a state of rest, energy will be required to set the air in motion. This energy is equal to the velocity pressure at the entrance to the duct. Assuming the hood in this example empties into a 7" diameter duct, the required 1165 ACFM results in a velocity of 4363 FPM:

$$V = Q \div A$$

where: Q = 1165 CFM

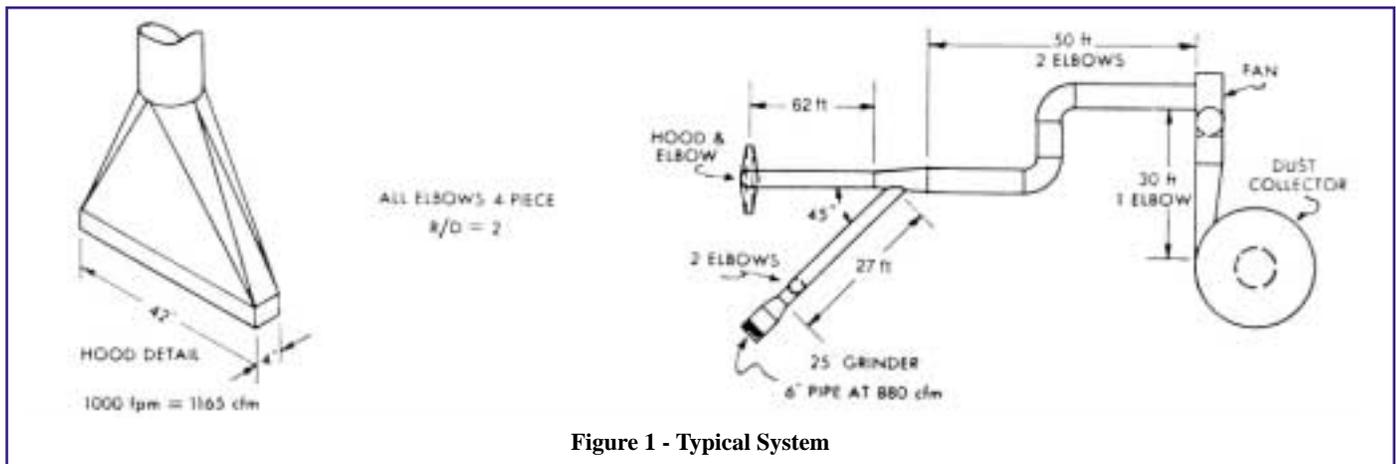


Figure 1 - Typical System

$$A = \frac{(3.5 \text{ in. radius})^2 \times 3.1416}{144 \text{ in.}^2 / \text{ft.}^2} = .267 \text{ ft.}^2$$

therefore:  $V = 1165 \text{ CFM} \div .267 \text{ ft.}^2 = 4363 \text{ FPM}$

The velocity pressure (VP) at 4363 FPM is calculated by:

$$VP = \left( \frac{\text{Velocity}}{4005} \right)^2$$

therefore: Acceleration Loss =  $\left( \frac{4363}{4005} \right)^2 = 1.19'' \text{ W.G.}$

The same result can be obtained by interpolating from the data in Figure 2.

The entry loss of a hood is a function of its efficiency. The efficiencies of several common entry conditions are shown in Figure 3. The relative efficiencies are expressed as losses in percentage of the duct velocity pressure. Consequently, the lowest percentage is actually the most efficient.

Outlet Velocity	Velocity Pressure	Outlet Velocity	Velocity Pressure	Outlet Velocity	Velocity Pressure
800	.040	2800	.489	4600	1.32
1000	.063	3000	.560	4800	1.44
1200	.090	3200	.638	5000	1.56
1400	.122	3400	.721	5200	1.69
1600	.160	3600	.808	5400	1.82
1800	.202	3800	.900	5600	1.95
2000	.250	4000	.998	5800	2.10
2200	.302	4200	1.10	6000	2.24
2400	.360	4400	1.21	6200	2.40
2600	.422				

Figure 2 -Entrance Loss Percentage

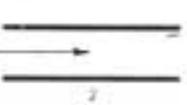
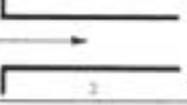
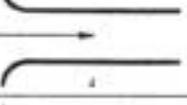
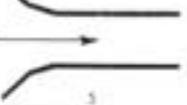
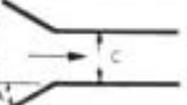
ILLUSTRATION 3 -- PIPE ENTRANCE LOSSES	
ENTRY	LOSS IN % OF PIPE V.P.
	GRINDER HOOD 60%
	UNFLANGED PIPE 90%
	FLANGED PIPE 30%
	SMOOTH WELL ROUNDED 3%
	FABRICATED WELL SHAPED 5%
	"A" LESS THAN 45" % LOSS = 100% "B" V.P. 25% "C" V.P.

Figure 3 - Entrance Losses

The hood in this example is most similar to item 2 in Figure 3. Therefore, the entry loss from atmosphere into the hood is .90 x the entering air velocity pressure at 1000 feet per minute or:

$$\text{Entry Loss} = .90 \times \left( \frac{1000}{4005} \right)^2 = .06'' \text{ W.G.}$$

This loss could have been reduced to .5 VP by simply adding a flange to the bottom edge of the hood as indicated by item 3 in Figure 3.

The total hood loss in the example is the acceleration loss added to the entry loss:

$$\text{Hood loss} = .06'' + 1.19'' = 1.25'' \text{ W.G.}$$

**PRIMARY BRANCH**

The duct loss from the hood to the branch junction can be determined by using the equivalent length method. This run of duct includes 62' of 7" diameter duct and one 4 piece 90° elbow of R/D = 2. According to Figure 4, the elbow has a loss equal to 12 diameters of 7" duct, or 7'. Thus, the total equivalent length of straight duct is 69'.

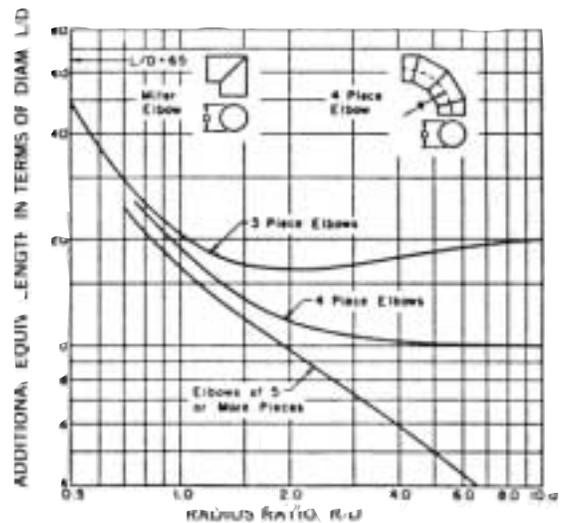


Figure 4 - Loss in 90° elbows of round cross-section

Chart I on page 4 indicates a 4.0" loss for every 100' of 7" diameter duct handling 1165 CFM. The loss for this run can be determined as:

$$\text{Duct Loss} = \left( \frac{69}{100} \right) \times 4.0 = 2.76'' \text{ W.G.}$$

Therefore, the total resistance of the hood branch to the junction is:

$$\text{Branch Loss} = 1.25'' + 2.76'' = 4.01'' \text{ W.G.}$$

**SECONDARY BRANCH**

A secondary branch is calculated in the same manner as the main branch. For example, a grinder hood handling 880 CFM through a 6" pipe results in a velocity of 4500 FPM, which has a 1.26" VP.

According to item 1 in Figure 3, a grinder hood has a .6 VP loss, so the total hood loss will be:

$$\text{Hood Loss} = 1.26'' + (.60 \times 1.26'') = 2.02'' \text{ W.G.}$$

The duct branch from the grinder hood to the junction consists of 27' of 6" pipe and (2) 4 piece 90° elbows of R/D = 2. With an equivalent length of 39' (27' + 6' + 6') the duct loss for this run is:

$$\text{Duct Loss} = \left( \frac{39}{100} \right) \times 5.2 = 2.03" \text{ W.G.}$$

The total resistance of the grinder branch to the junction is:

$$\text{Branch Loss} = 2.02" + 2.03" = 4.05" \text{ W.G.}$$

Note that the resistance in both branches is nearly equal. This is because the pressures in converging branches must be equal during operation or the system will automatically equalize by adjusting the flow different than the design point. If the variation in resistance between any two converging branches exceeds 5%, further design is required to balance the loss in both branches. Where necessary, balancing can be accomplished by altering duct lengths, duct diameters, or air volumes.

### MAIN DUCT

The main duct resistance calculations begin with the selection of the appropriate duct diameter. Assuming a minimum conveying velocity of 4500 FPM and an airflow requirement of 2045 ACFM (880 + 1165) in the main, a 9" diameter duct will suffice with a resulting velocity of 4630 FPM.

The junction itself represents a loss due to the mixing effect of the converging branches. The ratio of the CFM in the branch (1165 ÷ 880 = 1.3) can be used to determine the loss in percent of VP in the main. Interpolating from the data in Figure 5 results in:

$$\text{Junction Loss} = .19 \left( \frac{4630}{4005} \right)^2 = .25" \text{ W.G.}$$

LOSS IN MAIN AT JUNCTION WITH BRANCH. (BASED ON 45° TEE & EQUAL MAIN & BRANCH VELOCITIES.)	
CFM in Upstream Main ÷ CFM in Branch	Loss in Main in % of Main V.P.
1	.20
2	.17
3	.15
4	.14
5	.13
6	.12
7	.11
8	.10
9	.10
10	.10
CORRECTION FACTORS FOR OTHER THAN 45° TEE.	
Tee Angle	45° Loss X Factor
0	0
15	0.1
30	0.5
45	1.0
60	1.7
75	2.5
90	3.4

Figure 5

Chart II on page 4 indicates a resistance of 3.3" for every 100' of 9" diameter duct handling 2045 CFM. According to Figure 4 the two elbows are equal to another 18' of duct, so the total equivalent length is 68' between the junction and the fan.

$$\text{Duct Loss} = \left( \frac{68}{100} \right) \times 3.3 = 2.24" \text{ W.G.}$$

Note that all the losses to this point, up to the fan inlet, are expressed as negative pressure. Also that only the branch with the greatest loss is used in determining the total. Therefore:

$$\text{SP inlet} = (-4.05") + (-.25") + (-2.24") = -6.54" \text{ W.G.}$$

Assuming the same size duct from the fan to the collector, the 30' of duct and the one elbow will have a loss equivalent to the following:

$$\text{Duct Loss} = \left( \frac{39}{100} \right) \times 3.3 + 1.29" \text{ W.G.}$$

The pressure drop across the dust collector, like coils or filters, must be obtained from the manufacturer of the device. Assuming a 2.0" loss for this example, the resistance at the fan outlet is:

$$\text{SP outlet} = 1.29" + 2.0" = 3.29" \text{ W.G.}$$

### FAN SELECTION

At this point enough information is known about the system to begin fan selection. Because fans are rated independent of a system, their ratings include one VP to account for acceleration. Since the system resistance calculations also consider acceleration, fan static pressure can be accurately determined as follows:

$$\text{Fan SP} = \text{SP outlet} - \text{SP inlet} - \text{VP inlet}$$

In this example with 4630 FPM at the fan inlet:

$$\text{Fan SP} = 3.29" - (-6.54") - 1.33" = 8.5" \text{ W.G.}$$

For this example, a fan should be selected for 2045 ACFM at 8.5" SP and have an outlet velocity of at least 4500 FPM to prevent material settling. This presumes a standard airstream density of .075 lbs./ft.<sup>3</sup> If the density were other than standard, the system-resistance calculations would have been the same but the resulting fan SP would have been corrected. Refer to Engineering Letter 4 for density correction procedures.

This example also assumes that the fan inlet and outlet connections are aerodynamically designed. Fans are sensitive to abrupt changes in airflow directly adjacent to the fan inlet or outlet. The effects of abrupt changes and other "system effect" problems are discussed in Engineering Letter 5.

### CONCLUSION

It is the responsibility of the system designer to ensure that there are adequate air flows and velocities in the system and that the selection of duct components and fan equipment has been optimized. While computer programs do the bulk of system calculations today, this Engineering Letter should help to provide a common set of methods and terminology to assist in that effort.

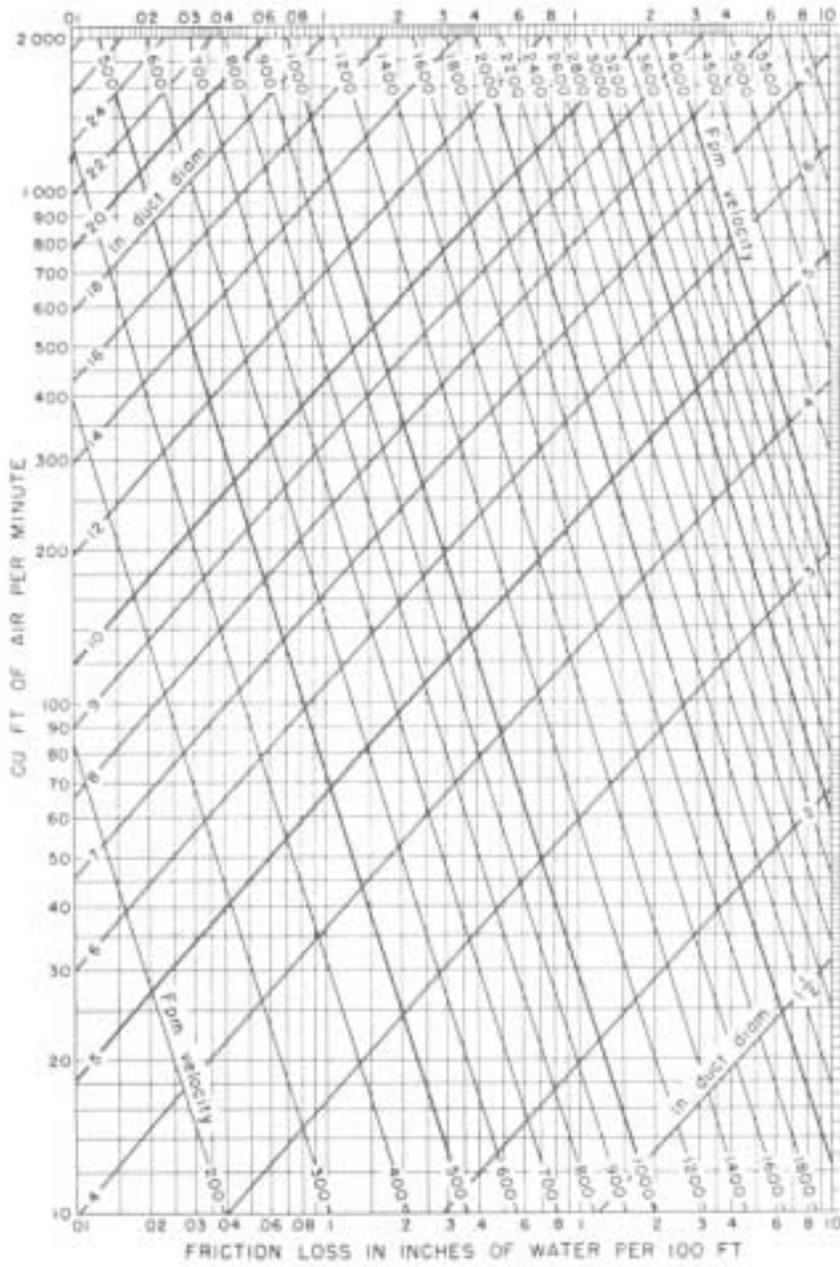
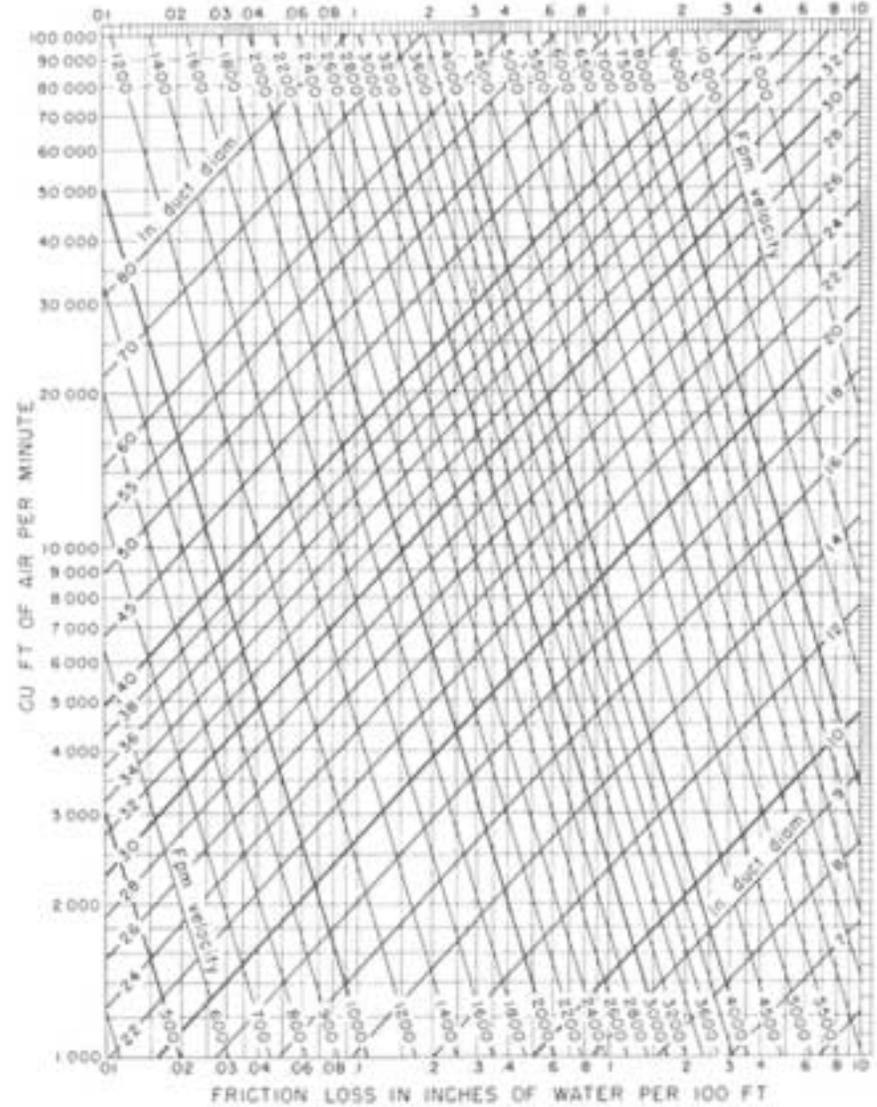


Chart I



**FRICTION OF AIR IN STRAIGHT DUCTS**  
**Heating Ventilating Air Conditioning Guide 1950**  
**Copyright 1950**

Based on standard air of 0.075 lb per cu. ft. density flowing through average, clean, round, galvanized metal ducts having approximately 40 joints per 100 ft.

**No safety factor included. Caution: Do not extrapolate below chart.**

Chart II

## FAN LAWS AND SYSTEM CURVES

### INTRODUCTION

The purpose of this Engineering Letter is to explain the basis and application of the rules used to predict fan performance in a given system. With a basic understanding of these rules, the performance of a fan can be quickly calculated for various conditions.

### SYSTEM REQUIREMENTS

The three fundamental rules governing fan performance are commonly called the “fan laws.” These rules are only valid within a fixed system with no change in the aerodynamics or airflow characteristics of the system. For the purpose of this discussion, a *system* is the combination of ductwork, hoods, filters, grills, collectors, etc., through which air is distributed. Therefore, these rules can also be referred to as “system laws.”

### VOLUME AND PRESSURE

The motion of any mass causes friction with its surroundings. The movement of air through a system causes friction between the air molecules and their surroundings (duct walls, filter media, etc.) and any other air molecules. Energy is required to overcome this friction, or resistance. The faster the air moves the greater the resistance to flow and the more energy is required to push or pull the air through the system.

This energy is stated in terms of pressure. The portion of the pressure that results in air velocity is described as velocity pressure (VP). The portion necessary to overcome friction in the air and in the system is described as static pressure (SP). The sum of the two is described as total pressure (TP).

The law of physics, for motion, is expressed algebraically as:

$$V = \sqrt{2gh} \quad \text{or} \quad V^2 = 2gh$$

where V = velocity of flow  
 g = force of gravity  
 h = pressure causing flow

As can be seen from the equation, the pressure necessary to cause flow is proportional to the square of the velocity. In a system, this means that SP will vary as the square of the change in velocity or volume expressed in cubic feet per minute (CFM). This makes it possible to predict all possible combinations of SP at the corresponding CFM given any one such calculated relationship of SP and CFM for a fixed system.

For example, a system is calculated to require a static pressure equal to 2" water gauge at an airflow rate of 1000 CFM. If it is desired to increase the flow to 1500 CFM without any physical change in the system, the required SP would be:

$$(1500 \div 1000)^2 \times 2" = 4.5" \text{ SP}$$

$$\left(\frac{\text{CFM new}}{\text{CFM old}}\right)^2 = \frac{\text{SP new}}{\text{SP old}}$$

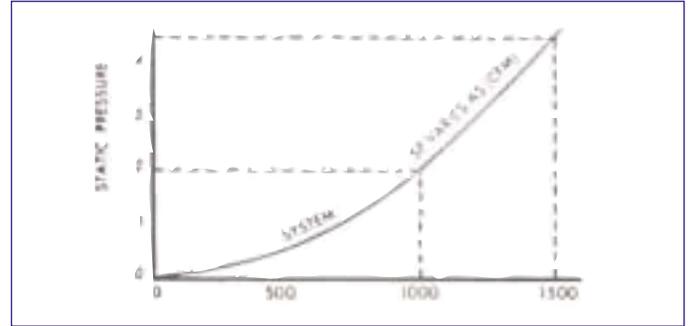


Figure 1 - System Curve

The same calculation using any number of varying CFM ratings would result in a plotted curve as shown in Figure 1.

Regardless of fan type, fan size, or volume of flow through a system, the relationship of CFM to SP will not change unless the system itself is altered in some way. SP always varies as the square of the change in CFM. The only exception to this rule is found in a laminar flow characteristic where VP is of far greater importance than SP. Such circumstances are not typical of fan systems.

### FAN LAWS

In air movement systems, it is the fan wheel that does the work. In a sense, the fan wheel acts like a shovel. As it revolves, it discharges the same volume of air with each revolution. Working within a fixed system, a fan will discharge the same volume of air regardless of air density, (disregarding the effects of compression at high pressures).

If the fan RPM is increased, the fan will discharge a greater volume of air in exact proportion to the change in speed. This is the first “fan law.”

1. CFM varies in direct proportion to change in RPM

$$\text{CFM (new)} = \frac{\text{RPM (new)}}{\text{RPM (old)}} \times \text{CFM (old)}$$

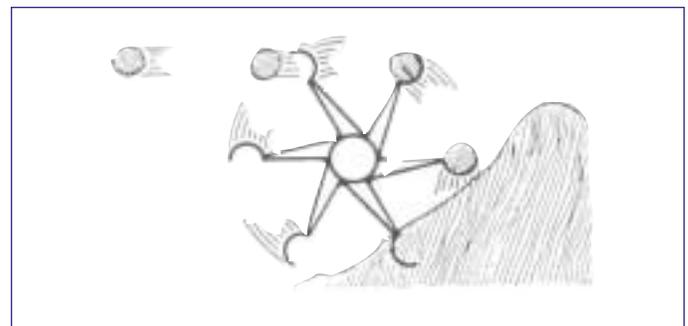


Figure 2 - A fan wheel is a constant volume device.

As shown earlier, in a system, the SP varies as the square of the change in CFM. Since CFM varies directly with RPM, RPM can be substituted for CFM in the system equation. Therefore, SP varies as the square of the change in RPM. This is the second “fan law.”

2. SP varies in proportion to the change in (RPM)<sup>2</sup>

$$SP \text{ (new)} = \left( \frac{RPM \text{ (new)}}{RPM \text{ (old)}} \right)^2 \times SP \text{ (old)}$$

The efficiency of a fan is a function of its aerodynamic design and point of operation on its SP/CFM curve (see Figure 3). As the fan speed changes, this relative point of operation remains unchanged as long as the system remains unchanged. Thus, the fan brake horsepower varies proportionally as the cube of the change in RPM. This is the third “fan law.”

3. BHP varies in proportion to the change in (RPM)<sup>3</sup>

$$BHP \text{ (new)} = \left( \frac{RPM \text{ (new)}}{RPM \text{ (old)}} \right)^3 \times BHP \text{ (old)}$$

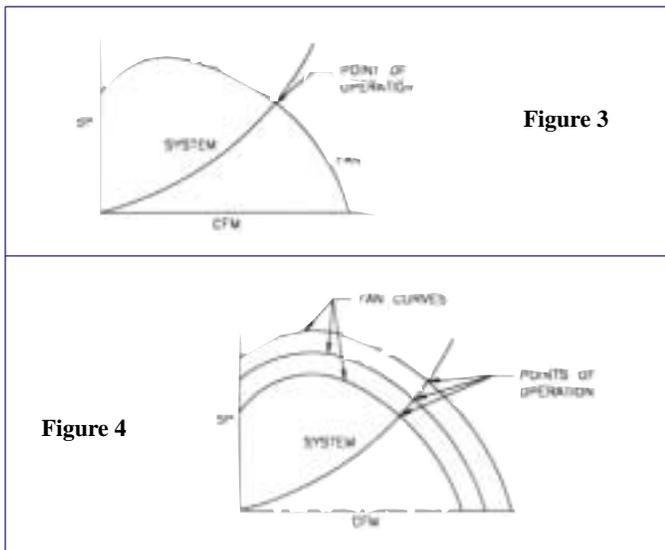
It is important to remember that each of these “fan law” relationships takes place simultaneously and cannot be considered independently.

### FAN CURVE AND SYSTEM CURVE

As stated previously, a system curve can be plotted to show all possible combinations of SP and CFM for a given fixed system. Any fan used on that system must operate somewhere on that system curve.

Fan performance is determined by laboratory testing and is presented graphically in the form of fan curves. Unless it is physically altered in some way, a fan must operate somewhere on its SP/CFM curve. The relative shape of that curve will not change, regardless of fan speed.

Because the fan and system can each only operate somewhere on their own respective curves, a fan used on a fixed system can only have one point of operation. The point of operation, as shown in Figure 3, is the intersection of the system curve and the fan SP CFM curve.



If the fan speed is increased or decreased, the point of operation will move up or down the existing system curve. This is shown in Figure 4.

The following are examples of how the fan curve can be used to calculate changes to flow and pressure requirements.

Example 1: A fan has been selected to deliver 35,530 CFM at 8" SP. The fan runs at 1230 RPM and requires 61.0 BHP.

After installation, it is desired to increase the output 20%. At what RPM must the fan run? What SP will be developed? What BHP is required?

1. CFM varies as RPM  
(1230) (1.20) = 1476 RPM
2. SP varies as (RPM)<sup>2</sup>  
(1476/1230)<sup>2</sup> (8) = 11.52" SP
3. BHP varies as (RPM)<sup>3</sup>  
(1476/1230)<sup>3</sup> (61.0) = 105.4 BHP

Example 2: A fan was originally installed to deliver 10,300 CFM at 2 1/4" SP and to run at 877 RPM, requiring 5.20 BHP.

After installation, it is found that the system only delivers 9,150 CFM at 2 1/2" SP and uses 4.70 BHP. This indicates the original calculations were in error, or that the system was not installed according to plan. What fan RPM and BHP will be necessary to develop the desired 10,300 CFM? What SP should have been figured?

1. CFM varies as RPM  
(10,300/9,150) (877) = 987 RPM
2. SP varies as (RPM)<sup>2</sup>  
(987/877)<sup>2</sup> (2.50) = 3.17" SP
3. BHP varies as (RPM)<sup>3</sup>  
(987/877)<sup>3</sup> (4.70) = 6.70 BHP

### CONCLUSION

Use of the “fan laws” is based on a fixed system and a non-modified fan. Adding or deleting system components such as dampers, or incurring density changes, will create completely new system curves. Changing fan accessories such as inlet boxes, evases, or inlet dampers will alter the fan’s performance curve from standard. These variables must be considered before the fan laws can be applied.

During the process of system design, the fan laws can be helpful in determining alternate performance criteria or in developing a minimum/maximum range. If “safety factors” are applied to system calculations, it should be recognized that a 10% factor on volume will result in an increase in horsepower of 33% according to the third fan law. An evaluation should be made weighing the necessity of the safety factor versus the cost penalty incurred.

## UNDERSTANDING FAN PERFORMANCE CURVES

### INTRODUCTION

One of the most important documents customers request from fan manufacturers is performance curves. In addition to graphically depicting the basic fan performance data of CFM, RPM, and SP (on the static pressure curve) and BHP (on the brake horsepower curve), these curves also illustrate the performance characteristics of various fan types, like areas of instability, or the rate of change between flow and pressure. With some basic knowledge of performance curves, decisions can be made concerning fan selection, fan and system alterations, or the advisability of using a fan in a modulating system, for example.

Except for very large fans, performance curve information is generated by connecting the fan to a laboratory test chamber. Very specific test procedures are followed as prescribed in the Air Movement and Control Association's Standard 210 to assure uniform and accurate readings. Data points are collected at a given RPM while the flow is slowly modulated from full closed to full open. The information gathered is then used to develop computer selection programs and published capacity tables for use by system designers and end users.

### STATIC PRESSURE CURVE

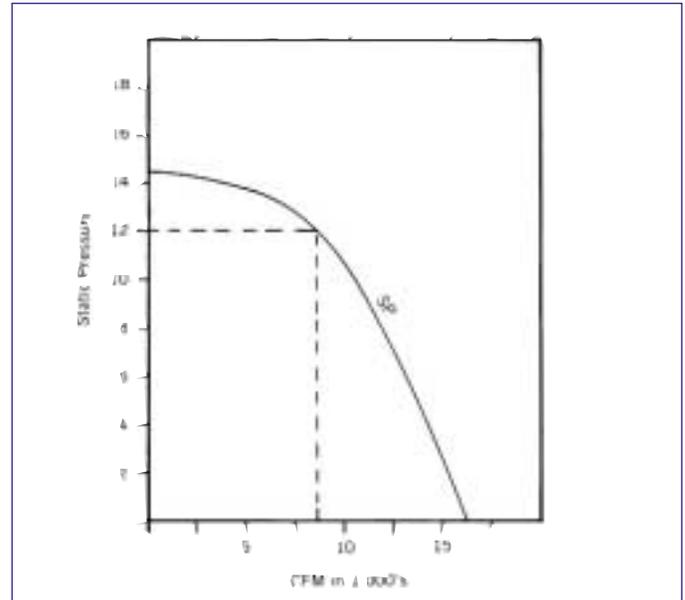
The static pressure curve provides the basis for all flow and pressure calculations. This curve is constructed by plotting a series of static pressure points versus specific flow rates at a given test speed. While the static pressure curve depicts a fan's performance at a given speed, it can be used to determine the fan's pressure capability at any volume.

In addition, it is also possible to approximate the fan's performance at other speeds by applying the following fan laws:

1. CFM varies as RPM
2. SP varies as (RPM)<sup>2</sup>
3. BHP varies as (RPM)<sup>3</sup>

To locate a fan's point of operation, first locate the required static pressure on the SP scale at the left of the curve. Then draw a horizontal line to the right, to the point of intersection with the SP curve. Next, draw a vertical line from the point of operation to the CFM scale on the bottom to determine the fan's flow capability for that SP at the given speed.

As shown in Figure 1, the performance for this fan is 8750 CFM and 12" SP at 1750 RPM.



**Figure 1 - Static Pressure Curve**

Assuming this same fan was intended to operate at 1200 RPM, the fan laws can be applied to obtain performance at this lower speed.

1. CFM varies as RPM

$$\frac{\text{CFM (new)}}{\text{CFM (old)}} = \frac{\text{RPM (new)}}{\text{RPM (old)}}$$

Therefore:

$$\text{CFM (new)} = \frac{1200}{1750} (8750) = 6000 \text{ CFM}$$

2. SP varies as (RPM)<sup>2</sup>

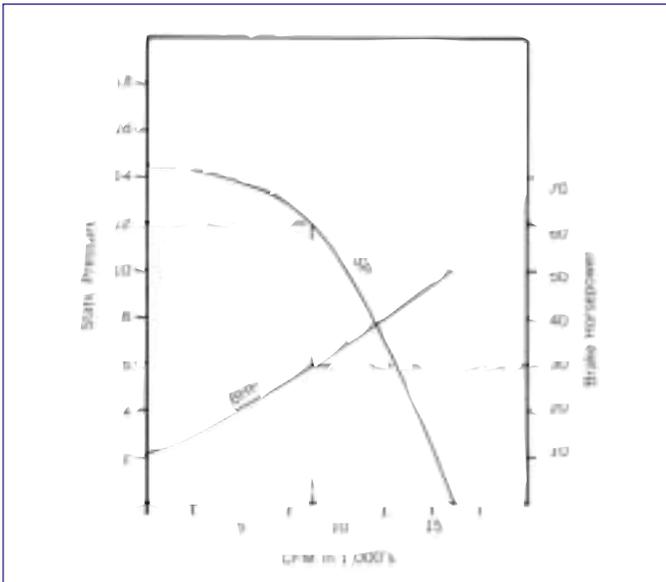
$$\frac{\text{SP (new)}}{\text{SP (old)}} = \left( \frac{\text{RPM (new)}}{\text{RPM (old)}} \right)^2$$

Therefore:

$$\text{SP (new)} = \left( \frac{1200}{1750} \right)^2 (12) = 5.6" \text{ SP}$$

### BRAKE HORSEPOWER CURVE

Once the CFM and SP have been determined, a BHP rating can be established. An accurate BHP rating is necessary to properly size the motor or to determine the operating efficiency of one fan as compared to another. Performance curves contain a BHP curve from which the BHP rating can be determined for specific capacities. To determine BHP at a specific point of operation, a horizontal line is drawn to the right from the point of intersection of the vertical CFM line and the BHP



**Figure 2 - Performance Curve**

curve. As shown in Figure 2, the fan operating at 8750 CFM and 12" SP at 1750 RPM is rated at 30 BHP. By employing the third fan law, the BHP rating can be determined for operation at 1200 RPM.

3. BHP varies as  $(\text{RPM})^3$

$$\frac{\text{BHP (new)}}{\text{BHP (old)}} = \left( \frac{\text{RPM (new)}}{\text{RPM (old)}} \right)^3$$

Therefore:

$$\text{BHP (new)} = \left( \frac{1200}{1750} \right)^3 (30) = 9.67 \text{ BHP}$$

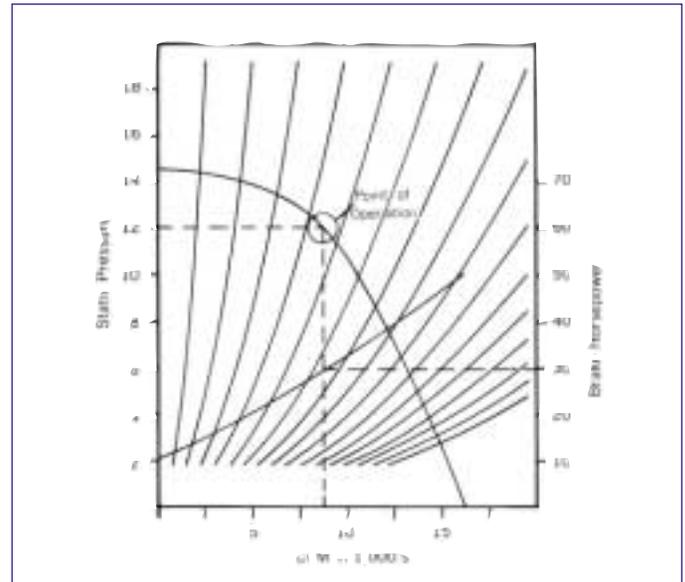
### SYSTEM LINES

Since fans are tested and rated independently from any type of system, a means of determining the fan's capabilities within a given system must be provided. The fan laws apply equally to any system; therefore, CFM and SP variations within the system are predictable. This enables system lines to be superimposed on performance curves to simplify performance calculations. The system line is nothing more than the sum of all possible CFM and SP combinations within the given system. Any combination of fan and system must operate somewhere along that system line.

Because a fan must operate somewhere along its SP curve and since the system has a known system line, their intersection is the point of operation. If the fan speed is changed, the point of operation must move up or down the system line. If the system is changed in some way, the point of operation must move up or down the SP curve. In practice, these principles can be used to check the accuracy of fan performance and system design.

### USING PERFORMANCE CURVES

Figure 3 illustrates the point of operation of a fan selected for 8750 CFM and 12" SP operating at 1750 RPM. Included in Figure 3 are a number of different system lines. If the system does not operate properly upon start-up, measurements can be taken and compared against the available performance curve.



**Figure 3 - Performance Curve with System Lines**

Let's assume that a tachometer reading indicates the fan is running at 1200 RPM instead of 1750 RPM, because of mistakes in motor speed or drive selection, and an airflow check indicates only 6000 CFM. These readings confirm that the system was calculated correctly and that the fan speed must be corrected to 1750 RPM as originally specified to achieve the desired 8750 CFM. If the tachometer reading indicates the proper speed but the airflow reading is down, additional system resistance beyond that originally calculated is indicated. This additional resistance could be caused by partially closed louvers/dampers, changes in duct sizing from the original design, system effect losses, or just an error in the system-resistance calculations. The deficiency can usually be corrected by either altering the system or increasing the fan speed.

Often, performance curves for one speed must be used to determine the performance of a fan for use on systems requiring more air or higher pressures. A performance curve such as Figure 4 can be used to determine fan performance beyond the SP scale shown by using the fan laws to obtain a reference point of operation on the system line. This can be accomplished by applying some suitable factor to the required CFM and the square of that factor to the required SP.

For example, the performance curve shown in Figure 4 can be used to determine fan performance requirements for a system calculated at 15,000 CFM and 23.5" SP, even though that point is beyond the curve. By determining a suitable reference capacity using the fan laws, that falls within the curve data, fan performance requirements can be obtained at the curve speed and then projected up to the system requirements using the fan laws once again.

The required 15,000 CFM and 23.5" SP is on the same system line as 10,000 CFM at 10.4" SP which intersects the fan's SP curve drawn for 1750 RPM and has a corresponding BHP of 33.0 at 1750 RPM. Therefore:

$$\text{RPM (new)} = \frac{15000}{10000} (1750) = 2625 \text{ RPM}$$

$$\text{BHP (new)} = \left( \frac{15000}{10000} \right)^3 (33.0) = 111 \text{ BHP}$$

## FAN PERFORMANCE CHARACTERISTICS

The performance characteristics of a fan can be determined from the performance curve at a glance. These characteristics include such things as stability, increasing or non-overloading BHP, and acceptable points of operation.

Fan performance is based on certain flow characteristics as the air passes over the fan wheel blades. These flow characteristics are different for each generic fan or wheel type, (i.e. radial, forward-curved, backwardly-inclined, radial-tip, and axial). Thus, the performance characteristics will be different for each of these general fan types. Further, these performance characteristics may vary from one manufacturer to the next depending upon the particular design. The characteristics described in this letter are based on **nyb** fan equipment.

The performance curves presented in Figures 1 through 4 are typical of fans with radial-blade wheels. The SP curve is smooth and stable from wide open to closed off. The BHP curve clearly indicates that the BHP increases steadily with the volume of air being handled as shown in Figure 4.

Fans with forward-curved wheels, such as shown in Figure 5, also have a BHP curve that increases with the volume of air being handled. The SP curve differs significantly from the radial since it exhibits a sharp “dip” to the left of the static pressure peak. This sharp dip (shaded area) indicates unpredictable flow characteristics. Though not technically accurate, this region is often referred to as the “stall” region. It indicates that at these combinations of pressure and relatively low volumes, the airflow characteristics across the wheel blades change or break away so that the fan performance point is no longer stable. Any fan with this characteristic SP curve should not be selected for operation in the unstable area.

As shown in Figure 6, the SP curve for a backwardly-inclined fan has a sharp dip to the left of the static pressure peak. This indicates an area of instability. However, the backwardly-inclined SP curve is generally steeper than that of the forward-curved wheel indicating its desirability for use in higher pressure systems. Therefore, variations in system resistance will result in smaller changes in volume for the BI Fan when compared to the FC Fan.

Even though New York Blower centrifugal fans with AcoustaFoil® wheels are stable in the area left of the peak, the majority of fans with backwardly-inclined wheels exhibit an SP curve similar in appearance to that of the forward-curved fan. The SP curve shown (in Figure 7) for fans using AcoustaFoil (air-foil, backwardly-inclined) wheels exhibits a much smoother depression to the left of the static pressure peak. This indicates that the overall fan design is such that internal flow characteristics remain desirable or predictable even in the region left of peak and that performance in this region is stable.

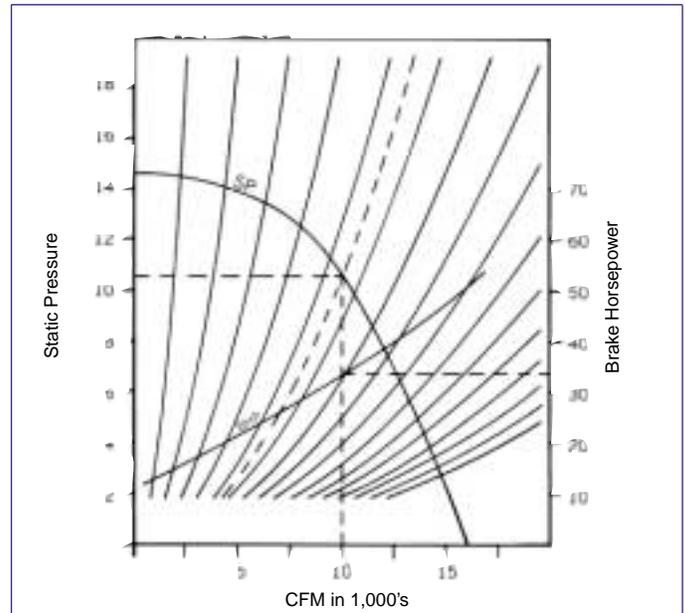


Figure 4 - Typical Radial-Blade Fan Performance Curve

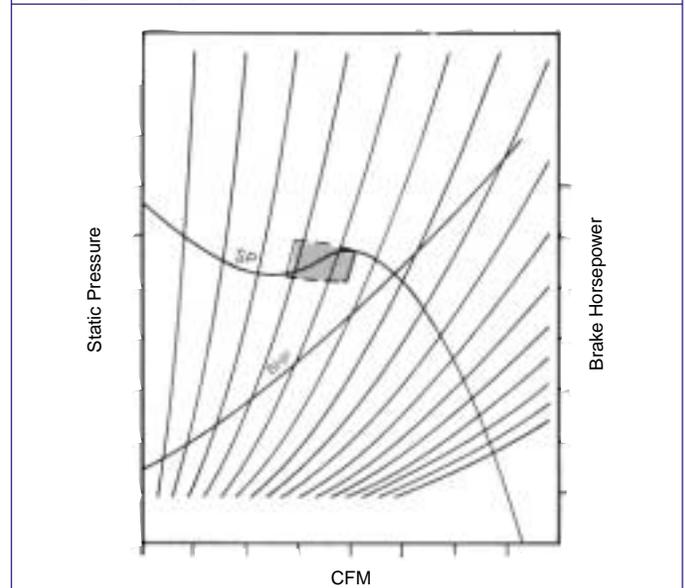


Figure 5 - Typical FC Fan Performance Curve

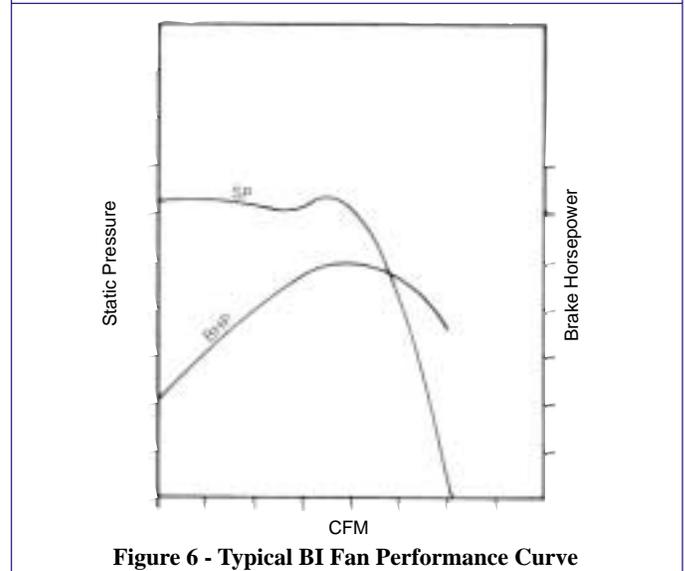


Figure 6 - Typical BI Fan Performance Curve

The BHP curve for all backwardly-inclined fans is the major difference between them and all other fan types. As shown in Figures 6 and 7, the BHP curve for backwardly-inclined fans reaches a peak and then drops off as the fan's volume increases. With this "non-overloading" BHP characteristic, it is possible to establish a maximum BHP for a given fan speed and select a motor that can not be overloaded despite any changes or errors in system design. Because BHP varies as  $(RPM)^3$ , this non-overloading characteristic does not apply to increases in fan speed, but it is very useful for motor protection against errors or changes in system calculations and installation.

Figures 5 and 6 indicate certain unacceptable selection areas on the SP curve. Although stability or performance may not be a problem, a point of operation down to the far right on the SP curve should be avoided. Selecting a fan that operates far down to the right, eliminates the flexibility to compensate for future system changes. Also, the fan is less efficient in this area as compared to a larger fan operating at a slower speed. Figure 7 shows the best selection area on the SP curve and the area in which the majority of capacity tables are published.

As is evident in Figure 8, the radial-tip fan design combines the backwardly-inclined SP curve characteristics with the radial fan's BHP curve. The radial tip is often more efficient than radial fans and typically best applied in high-pressure applications. As a result of its efficiency and dust-handling capabilities, the radial-tip fan can also be applied to lower pressure material conveying systems.

The term axial fan is used to describe various propeller, vaneaxial, tubeaxial, and duct fans. The performance curves of these fans are characterized by the ability to deliver large volumes of air in relatively low pressure applications. As can be seen in Figure 9, the axial flow fan is distinguished by a drooping BHP curve that has maximum horsepower at no flow or closed-off conditions. The axial fan SP curve exhibits an area of extreme instability to the left of the "hump" in the middle of the curve. Depending upon the severity, axial fans are normally only selected to the right of this area.

## CONCLUSION

A good working knowledge of performance curves is necessary to understand the performance characteristics and capabilities of different fan equipment. Use of performance curves in the selection of fan types and sizing will assure stable and efficient operation as well as future flexibility.

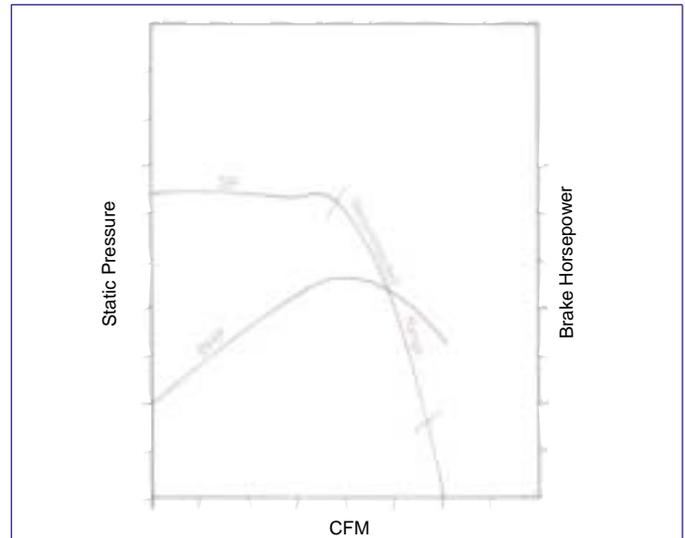


Figure 7 - Typical AcoustaFoil Fan Performance Curve

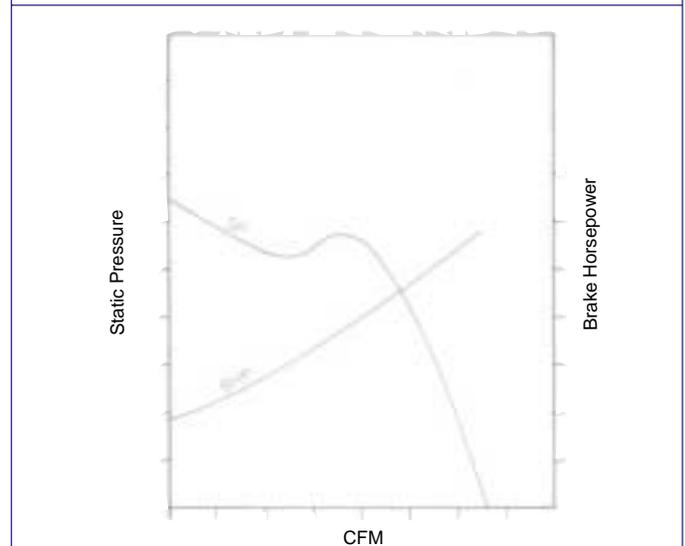


Figure 8 - Typical Radial-Tip Fan Performance Curve

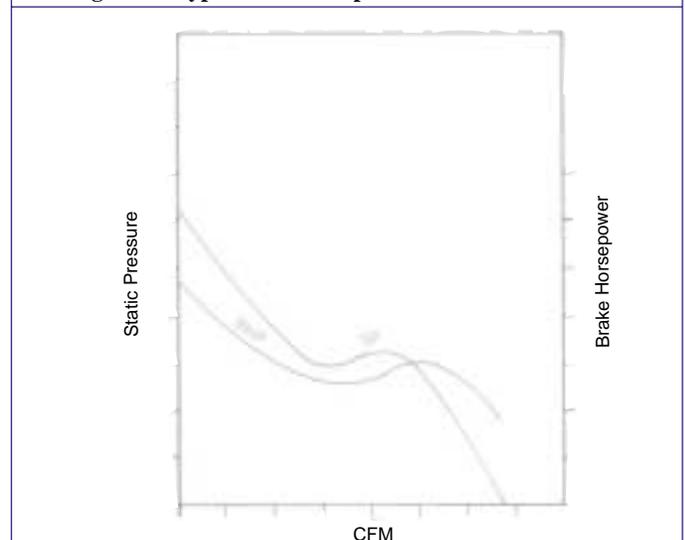


Figure 9 - Typical Axial Fan Performance Curve

## TEMPERATURE AND ALTITUDE AFFECT FAN SELECTION

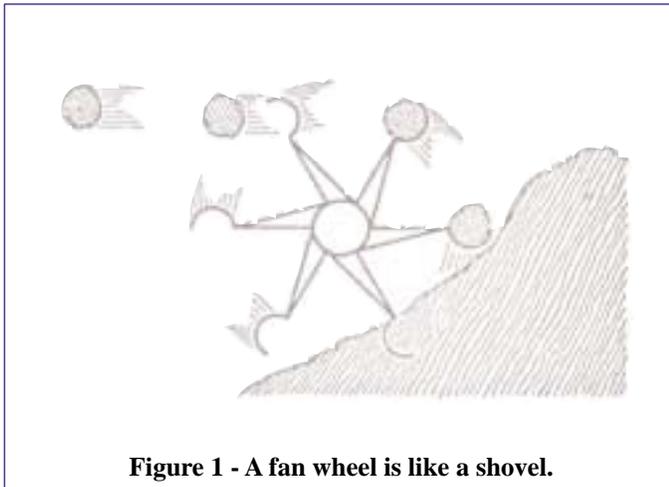
### INTRODUCTION

Fan performance changes with the density of the gas being handled. Therefore, all fans are cataloged at a standard condition defined as: 70°F. air, at sea level, with a gas density of .075 lb./ft.<sup>3</sup> at a barometric pressure of 29.92" Hg. At any other condition, the fan's horsepower requirement and its ability to develop pressure will vary. Therefore, when the density of the gas stream is other than the standard .075 lb./ft.<sup>3</sup>, correction factors must be applied to the catalog ratings in order to select the correct fan, motor, and drive.

In addition, the maximum safe speed of a wheel, shaft, or bearing can change due to an alloy becoming too brittle or too pliable at temperatures other than 70°F. Temperature derate factors must be applied to the fan's catalog maximum safe speed to ensure against overspeed situations.

### HOW TO CALCULATE ACTUAL FAN PERFORMANCE AT OTHER THAN 70 DEGREES FAHRENHEIT

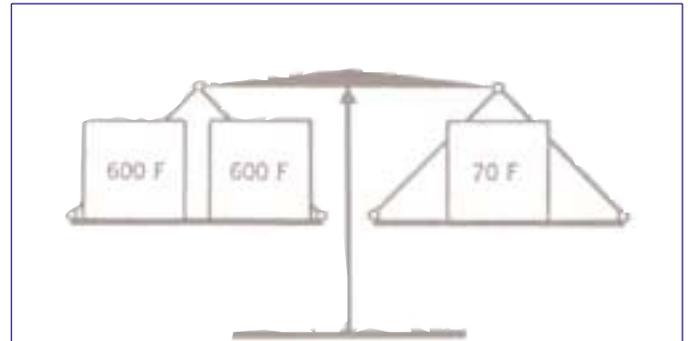
As illustrated in Figure 1, a fan wheel is similar to a shovel. In a given system, it will move the same volume of air regardless of the air's weight. If a fan moves 1000 CFM at 70°F., it will also move 1000 CFM at 600°F.



**Figure 1 - A fan wheel is like a shovel.**

However, air at 600°F. weighs half as much as it does at 70°F. Therefore, the fan requires just half the horsepower. (See Figure 2.) Likewise, since the air weighs half as much, it will create only half the static and velocity pressures. The reduction in static pressure is proportional to the reduction in horsepower, thus the overall fan efficiency will remain unchanged.

$$\text{Total Efficiency} = \frac{\text{CFM} \times \text{Total Pressure}}{6356 \times \text{Brake Horsepower}}$$



**Figure 2 - With hot gas, there is less weight to shovel.**

**Example 1.** A fan handling standard density, 70°F. air, delivers 12,400 CFM against 6" SP (static pressure) requiring 14.6 BHP (brake horsepower). If the system and fan RPM are not changed, but the inlet airstream temperature is increased to 600°F., how will the fan perform?

The fan will still deliver 12,400 CFM, but since the air at 600°F. weighs half as much as the air at 70°F., static pressure and horsepower will be cut in half. The fan will generate only 3" SP and require only 7.3 BHP.

A typical fan specification based on hot operating conditions is illustrated in Example 2.

**Example 2.** Required: 11,000 CFM and 6" SP at 600°F. (This means the actual, measurable static pressure of the fan at 600°F. will be 6 inches of water.)

The fan's catalog performance tables are based on 70°F. air at .075 density. The specified SP must be corrected by the ratio of the standard density to operating density. Since densities are inversely proportional to absolute temperature (degrees F. + 460):

$$6" \left( \frac{460 + 600}{460 + 70} \right) = 6" \left( \frac{1060}{530} \right) = 12"$$

The fan must be selected from the rating tables for 11,000 CFM at 12" SP. The BHP obtained from the table should be multiplied by the ratio of operating density to standard density in order to obtain the BHP at 600°F. If the rating table showed 30.0 BHP, the operating BHP would be 30.0 (530/1060) = 15.0 BHP.

In most "hot" systems, the fan is required to handle cold air until the system reaches temperature. A good example is in oven exhaust systems.

If Example 2 were such a case, the fan would require 30.0 BHP when operating at 70°F., and 15.0 BHP when the oven had warmed to 600°F. Very often a damper is furnished with the fan so that, during the warming-up period, the fan can be dampered to reduce the horsepower. Without the damper, a 30 HP motor would be needed.

Confusion can be avoided if the SP is specified at the temperature it was calculated. In Example 2, the specifications should read either:

- 11,000 CFM and 6" SP at 600°F., or
- 11,000 CFM for operation at 600°F. and 12" SP at 70°F.

Table 1 gives correction factors used to convert from a non-standard density to a standard density of 70°F. air. These factors are merely the ratios of absolute temperatures. Multiply the actual static pressure by the specific temperature/altitude factor so standard catalog rating tables can be used. Divide the brake horsepower from the catalog rating table by the temperature/altitude factor to get BHP at conditions.

**Table 1 - Corrections for Temperature**

Air Temperature °F.	Factor	Air Temperature °F.	Factor
-50	0.77	275	1.39
-25	0.82	300	1.43
0	0.87	325	1.48
+20	0.91	350	1.53
40	0.94	375	1.58
60	0.98	400	1.62
70	1.00	450	1.72
80	1.02	500	1.81
100	1.06	550	1.91
120	1.09	600	2.00
140	1.13	650	2.09
160	1.17	700	2.19
180	1.21	750	2.28
200	1.25	800	2.38
225	1.29	900	2.56
250	1.34	1000	2.76

**Table 2 - Corrections for Altitude**

Altitude Feet Above Sea Level	Factor	Altitude Feet Above Sea Level	Factor
0	1.00	5000	1.20
500	1.02	5500	1.22
1000	1.04	6000	1.25
1500	1.06	6500	1.27
2000	1.08	7000	1.30
2500	1.10	7500	1.32
3000	1.12	8000	1.35
3500	1.14	8500	1.37
4000	1.16	9000	1.40
4500	1.18	10000	1.45

**HOW TO CALCULATE ACTUAL FAN PERFORMANCE AT OTHER THAN SEA LEVEL**

A fan operating at an altitude above sea level is similar to a fan operating at air temperatures higher than 70°F.; it handles air less dense than standard. Table 2 gives the ratio of standard air density at sea level to densities of 70°F. air at other altitudes.

**Example 3.** Required: 5800 CFM at 6" SP at 5000 ft. altitude. 70°F. air at sea level weighs 1.20 times as much as 70°F. air at 5000 Ft. Therefore, at sea level, the SP is 1.2 x 6 = 7.20" SP. The fan would need to be selected for 5800 CFM at 7.2" SP at 70°F. .075 density.

When both heat and altitude are combined, the density of the air is modified by each, independently, so that the correction factors can be multiplied together.

**Example 4.** Required: 5800 CFM at 6" SP at 5000 ft. altitude at 600°F. Air at 70°F. at sea level weighs 2.00 x 1.20 = 2.40 times as much as 600°F. air at 5000 ft. altitude. At sea level and 70°F., SP = 2.40 x 6 = 14.4" SP. Select a fan for 5800 CFM at 14.4" SP. Divide the brake horsepower in the rating table by 2.40 to obtain horsepower at 600°F. and 5000 ft. If the fan is to start cold, it will still be at 5000 ft. altitude. Therefore, to get the "cold" horsepower requirement, divide by 1.20, the altitude factor only.

**DENSITY CHANGES FROM OTHER THAN HEAT AND ALTITUDE**

Fan densities may vary from standard for other reasons than heat and altitude. Moisture, gas, or mixtures of gases (other than air) are a few possibilities. In these cases, it is necessary to obtain the actual density of the airstream gas by some other reference material. A similar factor, as shown in Table 1, is then created using the standard density of air .075 lb. per cubic foot divided by the new density.

$$\text{Factor} = \frac{.075 \text{ lb./ft.}^3}{\text{special gas density}}$$

**ACFM-SCFM DEFINITION**

The terms ACFM and SCFM are often used in design work and cannot be used interchangeably.

SCFM is Standard Cubic Feet per Minute corrected to standard density conditions. To determine the SCFM of the volume used in Example 2, which was 11,000 CFM at 600°F., we would multiply the CFM by the density ratios.

$$11000 \times \frac{.037}{.075} = 5500 \text{ SCFM}$$

This indicates that if the weight of air at 600°F. were corrected to standard conditions its volume would be reduced to 5500 CFM.

ACFM stands for Actual Cubic Feet per Minute. It is the volume of gas flowing through a system and is not dependent upon density.

The terms ACFM and SCFM are often used in system design work where both quantities need to be known. It should be remembered, however, that since a fan handles the same volume of air at any density, ACFM should be used when specifying and selecting a fan.

## FAN SAFE SPEED AND TEMPERATURE

Whenever a fan is used to move air at temperatures substantially above or below 70°F., care must be taken to ensure that the safe speeds of wheel and shaft are not exceeded, and that bearing temperature and lubrication are satisfactory.

The maximum safe speed of a particular fan must be determined by calculations or actual tests. Safe speed depends entirely upon the wheel and shaft assembly's ability to withstand the centrifugal forces created by its own weight. Higher temperatures can affect the wheel and shaft assembly's ability to withstand these forces and therefore must be considered.

Most metals become weaker at higher temperatures. This weakness is measurable in terms of yield and creep strength. It can be translated into formulas that accurately determine the safe speed of a wheel and shaft assembly in relation to its tested maximum speed at standard conditions. Manufacturers provide safe speed reductions in their catalogs based on the alloy that was used to manufacture the wheel and/or shaft.

Some metals withstand heat better than others. Certain grades of stainless steel can be substituted to increase temperature limits. On the other hand, **fan wheels constructed of aluminum should never be operated above 200°F.**

For information regarding fiberglass reinforced plastic fan equipment, consult the appropriate product bulletin.

Table 3 gives an indication of the speed derate factors for several different alloys. These are listed for **reference purposes only**. For a specific fan, consult the appropriate product bulletin.

**Table 3 - RPM Derate Factors By Material**

Temperature °F.	Mild Steel	Aluminum	Stainless Steel		
			304L	316L	347
70	1.0	1.0	1.0	1.0	1.0
200	.97	.97	.88	.95	.95
300	.95	--	.82	.92	.93
400	.94	--	.78	.89	.90
500	.93	--	.75	.86	.90
600	.92	--	.73	.84	.90
800	.80	--	--	.79	.86
1000	--	--	--	.75	.83

The limiting temperature on any fan is the highest temperature that any component of the fan assembly will reach during any operating cycle. A fan in a process oven application may handle air several hundred degrees above the highest temperature the oven reaches, especially during start-up. On such applications, a temperature indicator should be located in the fan inlet to control the heat source and to keep the fan within its maximum safe temperature. This is particularly true where burners are located on the inlet side of the fan. In all cases, the fan should remain in operation until the air is cooled to 180°F. or less to prevent "heat soaking" of the fan shaft which could cause sagging.

Bearings must be kept cool; otherwise standard lubricants lose their effectiveness and bearing failures are likely. For axial fans, where the bearings are located in the airstream, care must be taken to ensure proper lubrication. Special fan and bearing designs, as well as high temperature lubricants, are available to extend the operating range to higher temperatures.

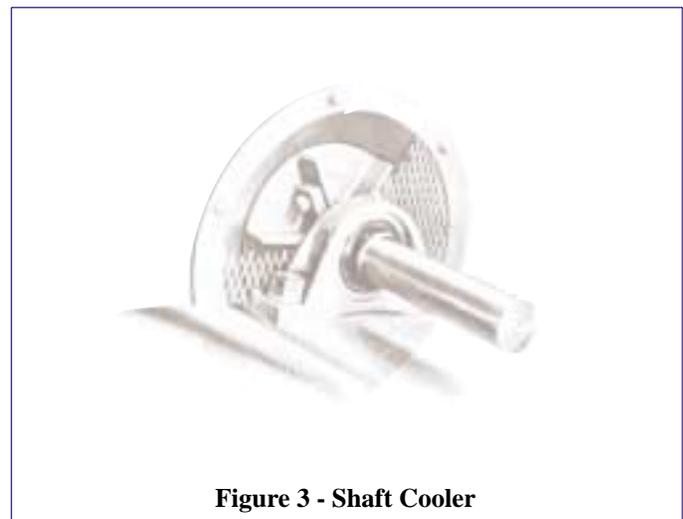
Arrangement 4 centrifugal fans, where the fan wheel is mounted on the motor shaft, should not be used above 180°F., unless special provisions are made (i.e., a shaft cooler or heat shield) to keep heat radiated from the housing from increasing motor bearing and winding temperatures.

When fan bearings are located outside of the airstream, as in Arrangement 1, 8, and 9 centrifugal fans, higher airstream temperatures are possible. Table 4 lists some typical maximum recommended operating temperatures for fans using ball or roller bearings.

A conventional fan using standard bearings and standard lubricant can normally be operated to a maximum of approximately 300°F. With the addition of a shaft cooler (Figure 3), this temperature limitation can be extended to 650°F. The shaft cooler has the effect of absorbing and dissipating heat from the shaft while circulating air over the inboard bearing.

**Table 4 - Maximum Fan Inlet Temperatures**

<b>Arrangement 1 and 8 (Overhung Wheel)</b>	
Standard Construction	300°F.
With Shaft Cooler	650°F.
With Shaft Cooler and Heat Gap	800°F.
With Shaft Cooler, Heat Gap, Stainless Wheel, and Alloy Shaft	1000°F.
<b>Arrangement 3 (Wheel Suspended Between Bearings)</b>	
Standard Construction	200°F.
<b>Arrangement 4 (Wheel on Motor Shaft)</b>	
Standard Construction	180°F.
<b>Enclosed Bearing Fans (Axial Fans)</b>	
Arrangement 4	105°F.
Arrangement 9	120°F.
With Special V-Belts with 2.0 S.F.	200°F.
Arrangement 9 Duct Fan With Heat-Fan Construction	600°F.
<b>Plenum Fans</b>	
Arrangement 3	105°F.
Arrangement 4	105°F.



**Figure 3 - Shaft Cooler**

With the addition of a heat gap (Figure 4) the temperature limitation can be extended to 800°F. since the fan pedestal is isolated from the hot fan housing. For specific applications, consult the appropriate product bulletin. Also, recognize that these limitations apply only to bearings and that wheel and shaft limitations must be treated independently.

All of the foregoing is based on the use of standard lubricants. When high-temperature lubricants are required, the type of lubricant and the frequency of relubrication are normally much more critical.

When the fan shaft is heated to the point that it expands more than the structure to which it is attached, one expansion bearing and one fixed bearing should be furnished. The fixed bearing is located on the drive end of the fan while the floating bearing is located next to the fan. This arrangement, however, is not critical and may vary by manufacturer.

When the fan is handling air below 70°F, there is the possibility of other problems. Below -30 to -50°F, ordinary steel is too brittle. Aluminum wheels or wheels of steel containing at least 5% nickel must be used, and shafts must be made of

nickel-bearing steel. In addition, lubricants become stiff, or even solid in these low-temperature applications. Exact operating conditions should be given to the fan manufacturer to relay to the bearing supplier for proper selection.

### CALCULATING “HOT” RESISTANCE FOR SYSTEMS

Figure 5 shows a system that operates at the same temperature throughout. If the inlet temperature is known, the fan may be selected from the fan capacity tables and the rated horsepower and static pressure corrected by the temperature correction factor from Table 1. However, what happens to the system that the fan was operating against? If a fixed system, which originally was calculated for standard air, was subjected to the same temperature increase as the fan, then system static pressure will change and be identical to the fan static pressure change. The result is that if a fan and system operate together the flow will remain unchanged. (See Figure 6.) Unfortunately, this example assumes that the entire system is being subjected to the same temperature change, which is not always the case.

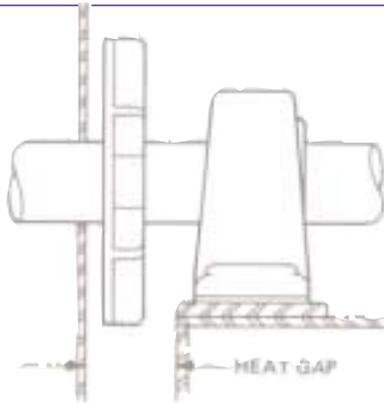


Figure 4 - “Heat Gap” between fan and bearing.

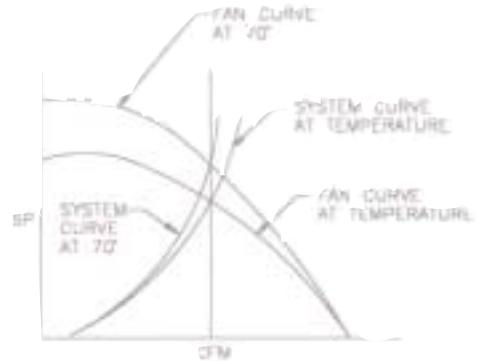


Figure 6 - Fan-system curve relationship with fan and system at the same temperature.

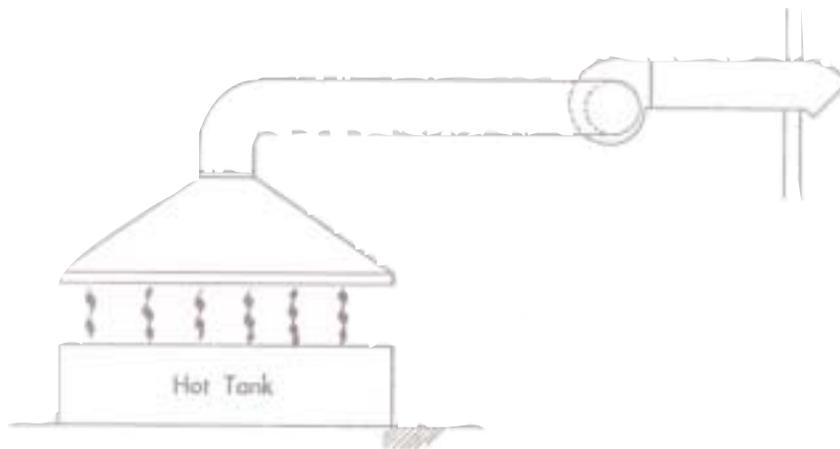


Figure 5 - A system with the same temperature throughout.

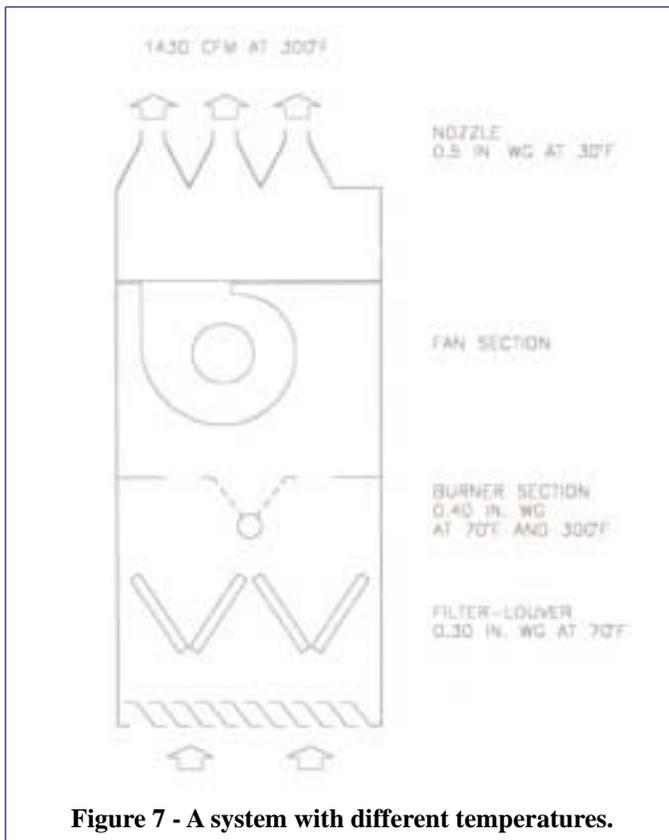
Figure 7 shows a system in which different temperatures are involved. The fan will not handle the same volume of air when operating hot as it does when cold. If the burner is on, the fan will handle 1430 ACFM against an actual static pressure of 1.2 inches. This is arrived at by adding the filter, burner, and nozzle resistance, neglecting for the sake of simplicity any external resistance from additional ductwork. The fan would be selected from the capacity tables on the basis of 1430 CFM at 1.72 inches static pressure (300°F. correction factor times 1.2 inches).

If the burner is turned off while the fan continues to operate at the same RPM, it is necessary to determine the system characteristic curve and plot its intersection with the fan to determine how much air the fan would move and at what static pressure. To accomplish this we must assume an arbitrary capacity, such as 1000 CFM at 70°F. The filter louver resistance would be the same, cold or hot, at .3 inches 70°F. The burner resistance would remain unchanged with temperature

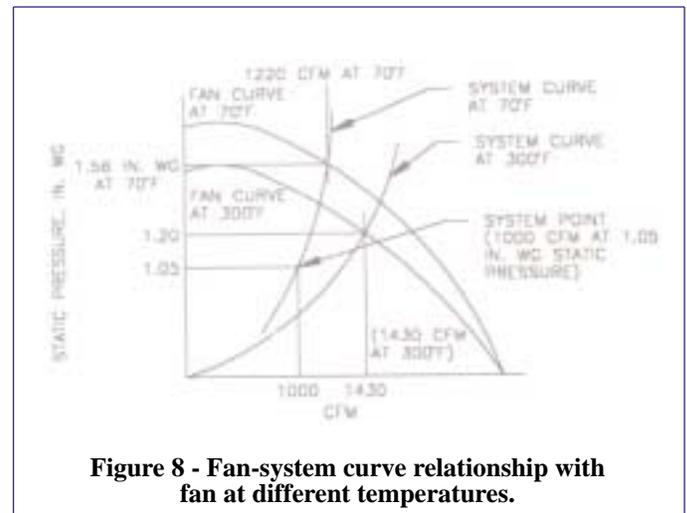
since it must be assumed that air expansion takes place after the high velocity section of the burner. The nozzles will vary in resistance directly as the density changes and inversely as the square of the flow. The nozzle would then have a resistance cold at 1000 CFM of:

$$.5" \times \left(\frac{1000}{1430}\right)^2 \times 1.43 = .35"$$

Summing these resistances yields the cold resistance at 1000 CFM of 1.05"SP. This new system point and corresponding curve are then plotted against a fan curve at standard conditions such that the resulting intersection will be the final operating point of the cold system. Using an actual fan as an example, the resulting flow would be 1220 CFM at 1.5 inches static pressure. (See Figure 8.)



**Figure 7 - A system with different temperatures.**



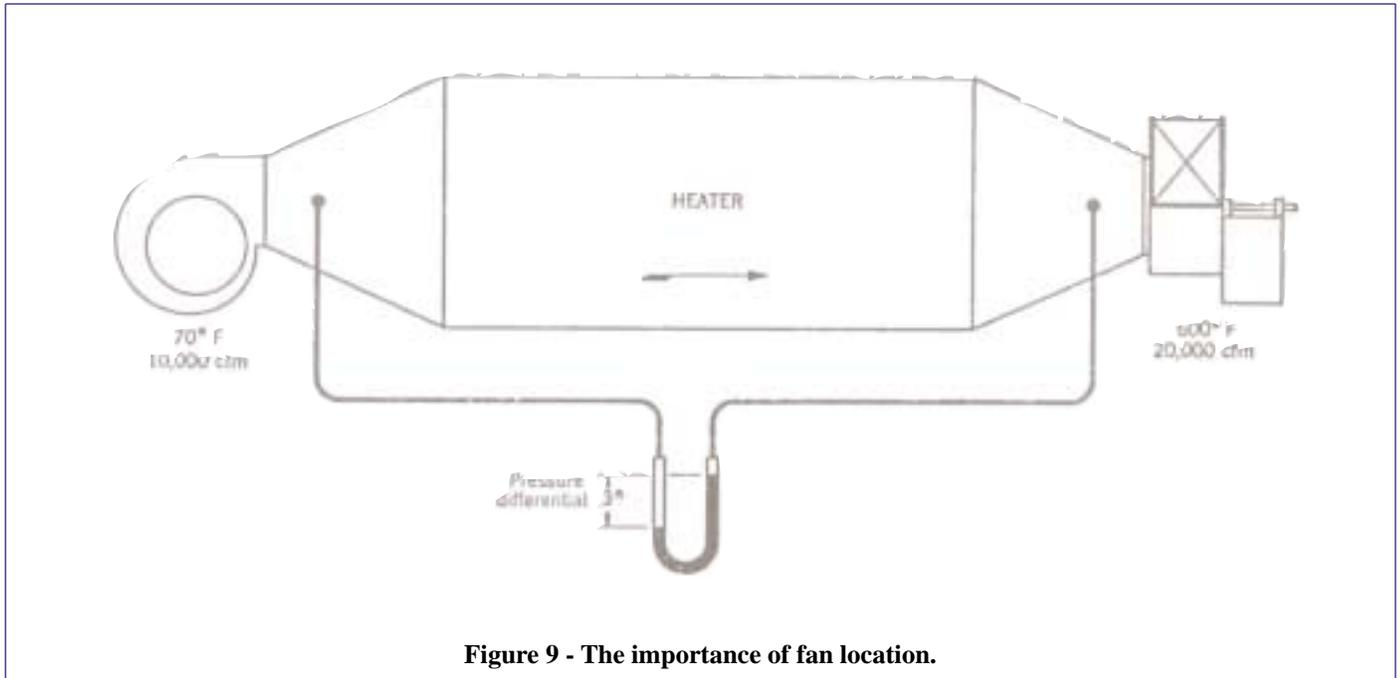
**Figure 8 - Fan-system curve relationship with fan at different temperatures.**

## FAN LOCATION IN HOT PROCESS SYSTEMS

Figure 9 shows how a fan may be located more economically in one part of a system, as contrasted to another. Suppose 10,000 CFM is to be heated from 70°F. to 600°F. Obviously, the heater will require the same 3-inch pressure differential whether the fan is to push the air into, or pull the air out of, the heater.

A fan pushing air into the heater would be specified to handle 10,000 CFM at 70°F. against 3 inches of static pressure at 70°F. One possible selection is a fan with a 27-inch wheel diameter, Class I design utilizing a 7½ HP motor.

The alternative fan, pulling air from the heater, would be specified to handle 20,000 ACFM at 600°F. against 3" SP at 600°F. It would be selected from the capacity tables for 20,000 CFM at 6" SP. One suitable choice is a fan with a 36½-inch wheel diameter, Class II design utilizing a 15 HP motor. (Note: 26 HP, from the tables, at 70°F., divided by temperature correction factor, is 13 HP at 600°F.) This example illustrates why it is usually more economical to locate the fan at the coolest part of the system. In this case, the "push" fan might cost half as much as the "pull" fan.



**Figure 9 - The importance of fan location.**

## FIELD TESTING OF FAN SYSTEMS

### INTRODUCTION

A fan system may require field testing when the system is thought to be malfunctioning, needs modification, or requires balancing of its volume and pressure characteristics.

When it has been determined that a field test is required, the test can provide a complete check on fan performance. This includes determination of air volume, fan static pressure, and fan brake horsepower.

This Engineering Letter details the steps involved in performing a field air test. A field test sheet, which simplifies the recording of test data and the calculation of test results, is provided. A list of safety precautions to be observed while conducting the test is also included.

### INSTRUMENTS REQUIRED

1. The best method of measuring both air velocity and static pressure in the field is with a Pitot tube and manometer. The absence of moving parts combined with fundamental simplicity make this set of instruments accurate and nearly foolproof. Both instruments may be used in nearly any atmosphere and require no adjustments except for zeroing the manometer prior to testing. Figure 1 shows a Pitot tube cross-section. Figure 2 demonstrates how it is connected to the manometer to indicate pressures by measuring the difference in heights of water columns in the "U" tubes.

Most manometers, such as shown in Figure 3, read directly in inches of water column. Some manometers may have velocity graduations marked directly in feet per minute for use where barometric pressure and temperature corrections are normal (i.e., test conditions assumed to be 70°F. and 29.92 inches of mercury).

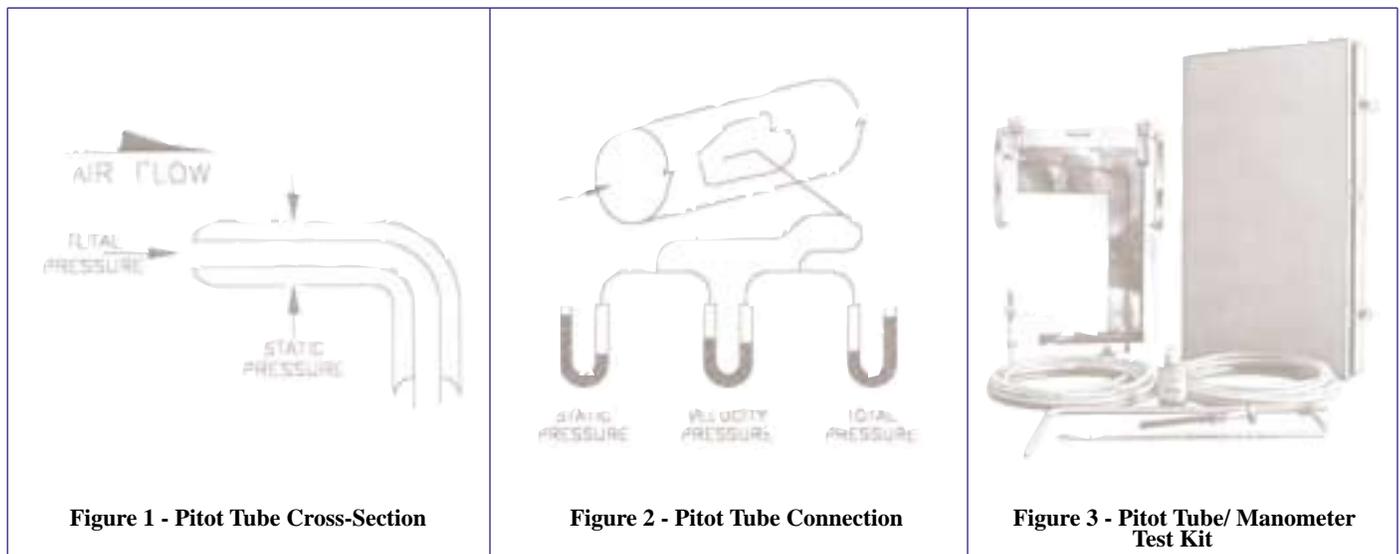
For greater convenience, a more compact Magnehelic pressure gauge may be used with a Pitot tube as a substitute for the manometer mentioned earlier. These gauges, illustrated in Figure 4, are available in a variety of pressure ranges.

2. A clip-on ammeter/voltmeter is used to obtain a reasonable estimate of fan motor horsepower.
3. A calibrated hand tachometer is used to determine the fan RPM.
4. An accurate temperature probe is used to measure temperature at each test location where volume or static pressure readings are taken.

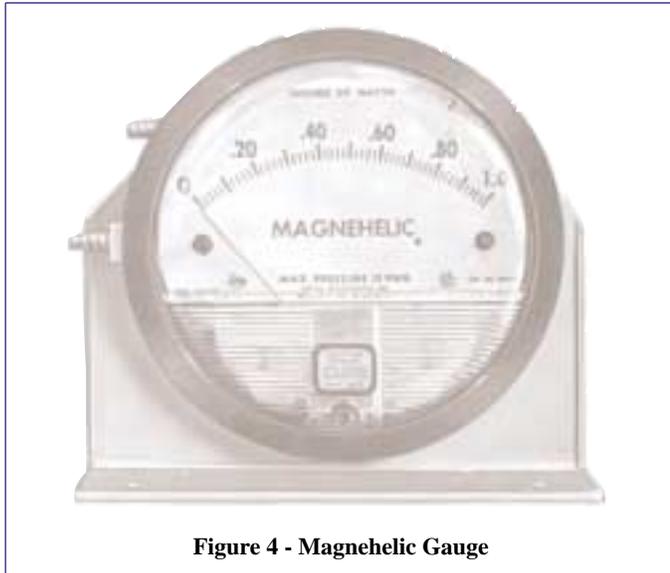
Sometimes there are no accessible test duct locations suitable for use with the Pitot tube. In this case, the air volume can be determined at the system entrance or exit, or through a grille or coil by using an anemometer or velometer. This method, however, is not as accurate and readings should only be taken by experienced service personnel familiar with this type of testing.

### PERFORMING A PITOT TUBE/MANOMETER TEST:

1. Make a sketch of the system as a record and as a guide for selecting locations for taking test readings. Often this will call attention to poor system-design features. Include dimensions, such as duct diameters or areas, duct length, motor size, motor speed, and sheave diameters on belt drive fans.



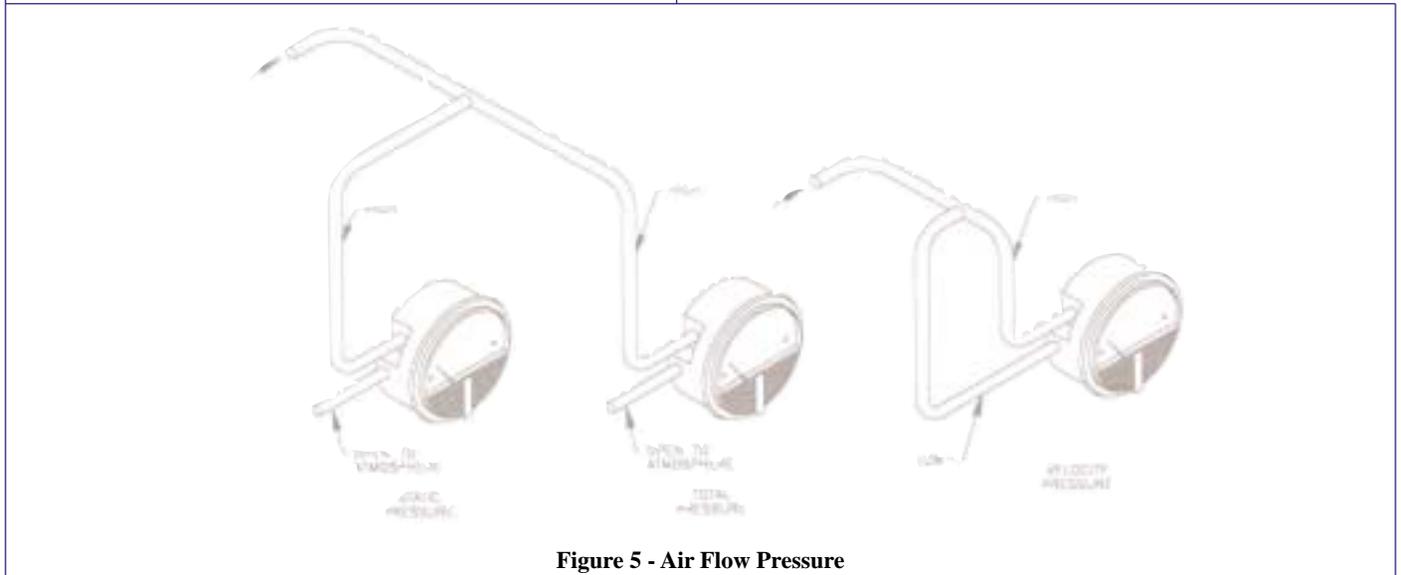
- Determine the best possible location for obtaining the air volume readings via a Pitot tube traverse (set of readings). The traverse location should not be directly after any turns, transitions, or junctions. The traverse should be after a minimum of 2 1/2 duct diameters of straight duct. To obtain the correct air volume, the Pitot tube and manometer or gauge should be connected to display velocity pressures, not velocities (see Figure 5). The location of the test points within each traverse is shown on the field test sheet included with this letter.



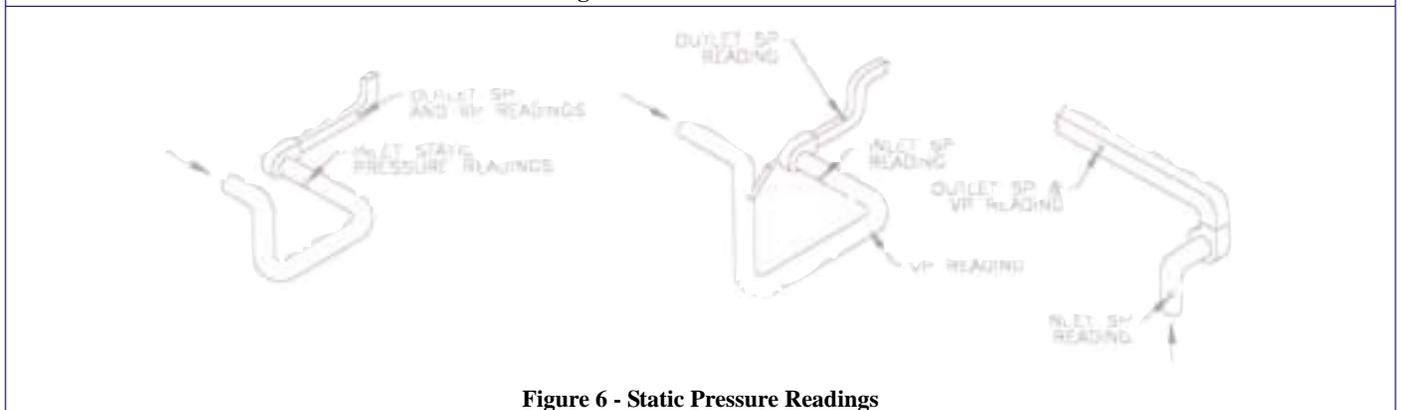
**Figure 4 - Magnehelic Gauge**

- Take static pressure readings several duct diameters from the fan inlet and outlet to avoid turbulence, (see Figure 6). If the fan has either an open inlet or outlet, assume the static pressure to be zero at the opening. Record the airstream temperatures at each static pressure location.
- Record the fan speed after measuring it with the tachometer. If a tachometer is unavailable, make sure you record the motor nameplate RPM and sheave diameters from which the fan speed can be calculated.
- Read the voltage and amperes supplied to the motor and record the values for calculation of fan motor horsepower.
- Measure the barometric pressure at the fan site with a portable barometer or obtain the pressure from the nearest weather station or airport. Be sure the barometric pressure is correct for your altitude and that it has not been corrected to sea level reference.
- Determine whether the air being handled contains quantities of moisture, particulates, and/or gases other than clean air. If so, obtain the concentrations and densities of the gases or mixture for use in making density corrections.

The attached test sheet is used to calculate flow through a fan. For additional information on conducting field tests of fan systems, AMCA Publication 203, Field Performance Measurements of Fan Systems, is recommended.



**Figure 5 - Air Flow Pressure**



**Figure 6 - Static Pressure Readings**

## CALCULATING FAN PERFORMANCE

The following steps explain how to calculate density, CFM, SP, and BHP using the acquired test data.

1. Determine the density of the airflow through the fan during the test by using the dry-bulb temperature at the fan inlet and the barometric pressure. Density in pounds per cubic foot is determined by:

$$\text{Density}_{\text{inlet}} = 0.075 \left( \frac{530}{460 + ^\circ\text{F.}} \right) \left( \frac{\text{Barometric Pressure}}{29.92} \right)$$

2. Determine the density of the airflow at the CFM test location (if different from inlet density) by:

$$\text{Density}_{\text{CFM}} = 0.075 \left( \frac{530}{460 + } \right) \left( \frac{\text{Barometric Pressure}}{29.92} \right)$$

3. Calculate fan inlet air volume in CFM as measured with the Pitot tube and manometer/gauge as follows: First, take the square roots of the individual velocity pressures and compute the average of the square roots. Then:

$$\text{CFM}_{\text{inlet}} = [1096 \times \text{test duct area (ft}^2\text{)}] \times \left( \sqrt{\frac{\text{Avg. of Sum of } \sqrt{\text{VP's}}}{\text{Density}_{\text{CFM test}}}} \right) \times \left( \frac{\text{Density}_{\text{CFM test}}}{\text{Density}_{\text{inlet}}} \right)$$

The above calculation gives air volume in actual cubic feet per minute (ACFM) which is the conventional catalog rating unit for fans. If standard cubic feet per minute is desired, it may be calculated as follows:

$$\text{SCFM} = \text{ACFM} \times \left( \frac{\text{Actual Inlet Density}}{\text{Standard Density}} \right)$$

4. Determine the fan static pressure (SP) by the following formula:

$$\text{SP}_{\text{fan}} = \text{SP}_{\text{outlet}} - \text{SP}_{\text{inlet}} - \text{VP}_{\text{inlet}}$$

$$\text{Where: } \text{VP}_{\text{inlet}} = \left( \frac{\text{CFM}_{\text{inlet}}}{1096 \times \text{inlet area in sq. ft.}} \right)^2 \times \text{Density}_{\text{inlet}}$$

NOTE: Correct inlet and outlet static pressure to standard values by the following formula before summing.

$$\text{SP}_{\text{standard}} = \text{SP}_{\text{actual}} \left( \frac{\text{Actual Density}}{\text{Standard Density}} \right)$$

5. Fan motor horsepower may be determined in several ways. The best is to read the volts and amperes supplied to the motor and apply the formula:

For single phase motors:

$$\text{Fan BHP} = \frac{\text{Volts} \times \text{Amps} \times \text{Power Factor} \times \text{Motor Eff.}}{746}$$

For three phase motors:

$$\text{Fan BHP} = \frac{\text{Volts} \times \text{Amps} \times \text{Power Factor} \times \text{Motor Eff.} \times \sqrt{3}}{746}$$

This method requires power factor and motor efficiency data, which may be difficult to obtain.

Another method is to draw an amps versus horsepower curve, (see Figure 7). This is done by plotting a rough horsepower versus amps curve for the motor as follows:

- a. Establish no-load amps by running the motor disconnected from the fan (point a).
- b. Draw a dotted line through one-half no-load amps, at zero HP, and nameplate amps, at nameplate HP (points b).
- c. At one-half nameplate HP, mark a point on this line (point c).
- d. Draw a smooth curve through the three points (a, c, b).
- e. Determine running HP by plotting running amps.

Multiply fan horsepower by the "K" density correction factor to determine HP at standard conditions.

6. Locate volume, static pressure, and horsepower on a performance curve drawn at the fan RPM. Curves can be generated using manufacturer's fan-selection software at specific densities, temperature, and altitude.

The test plot values will probably not fall exactly on the curve. If the fan system has been designed and installed properly, the difference should be small, reflecting test accuracy. If the difference is great, the system should be analyzed as described in the next section. Figure 8 shows a typical fan curve and field test points which fall on the curve.

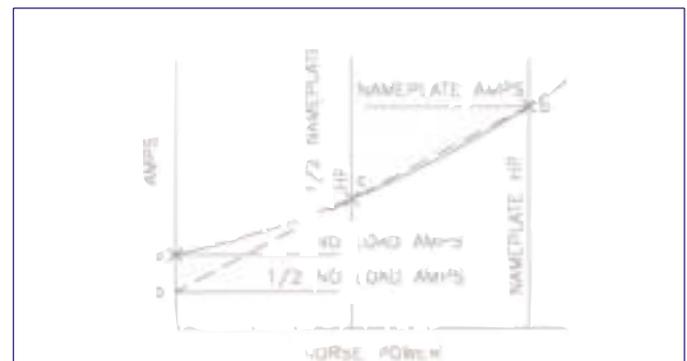


Figure 7 - Amperes versus Horsepower

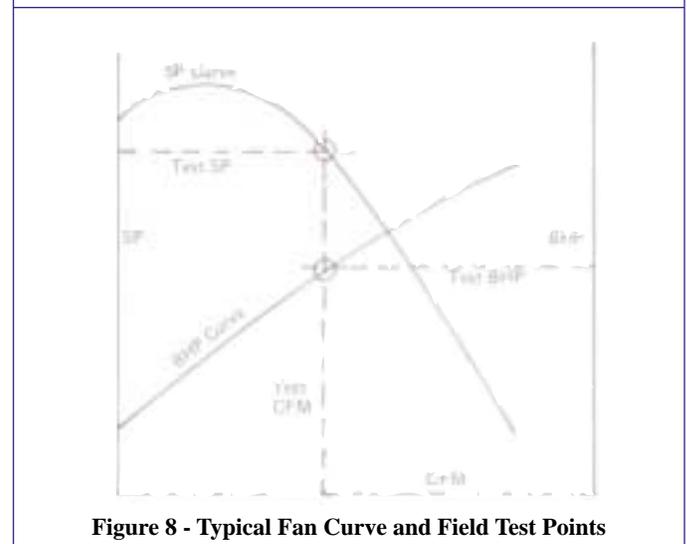


Figure 8 - Typical Fan Curve and Field Test Points

## POOR PERFORMANCE TEST RESULTS

If the test results indicate poor fan performance, a number of simple steps can be taken that could improve performance.

Be sure that any dampers at the fan inlet or outlet are set to the correct position, and that no other system dampers such as fire dampers, smoke dampers, or balancing dampers have been inadvertently closed.

A frequent cause of poor fan performance is the presence of poor inlet connections. Sharp elbows, inlet boxes without turning vanes, and duct configurations causing the air to spin upon entering the fan are examples of undesirable inlet connections.

Fan performance is also impacted by poor outlet conditions. Examine the outlet connection, keeping in mind that sharp elbows, rapid expansions, reductions, or the absence of an outlet connection all together can reduce fan performance.

By connecting the Pitot tube and manometer/gauge to read velocity pressure and inserting the Pitot tube through a hole at the inlet connection (as illustrated in Figure 9), pre-spin can be

determined. Once inserted, slowly twist the tube. The angle at which air is entering the fan can be determined by observing the angle of the tube generating the highest gauge reading. If the angle deviates noticeably from being parallel to the fan shaft, the air entering the fan inlet may be spinning and therefore reducing fan performance.

Another reason for poor performance could be stratification of the air entering the fan. By taking four temperature readings ninety degrees apart in the inlet duct near the fan, the possibility of stratification can be determined. A temperature difference of 10 degrees or more in the readings indicates stratification exists. An illustration of stratification is shown in Figure 10.

Refer to Engineering Letters 5 and 6 for more detailed explanations of system effect and improving fan performance.

## SAFETY PRECAUTIONS

The included list of safety precautions should be observed whenever testing or servicing fan equipment.

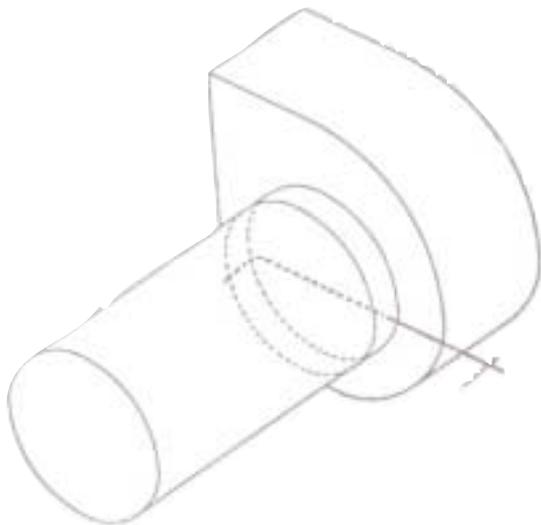


Figure 9 - Testing Fan Inlet for Spinning Airflow

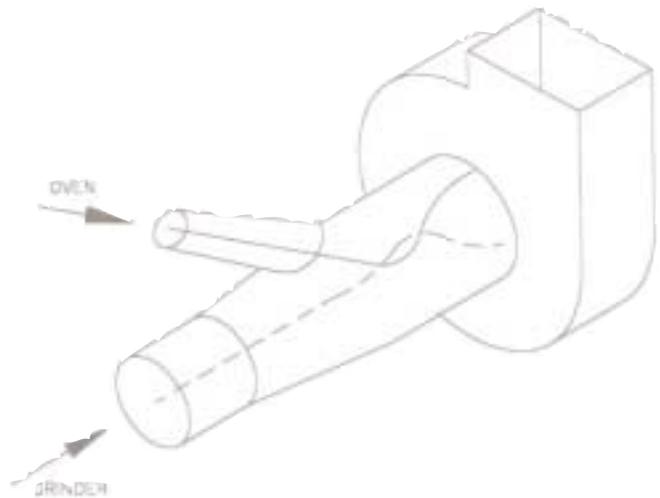
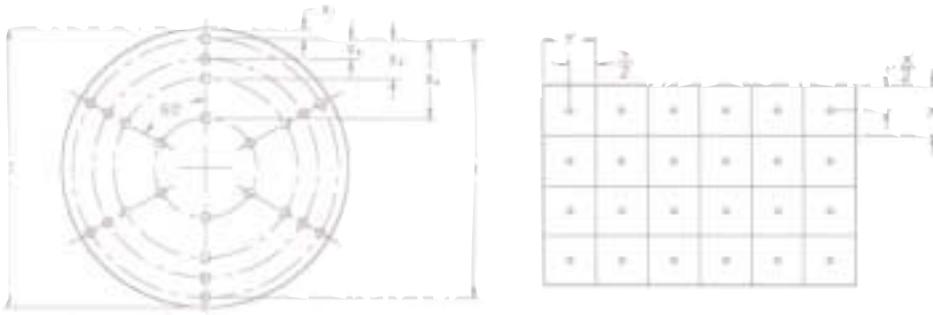


Figure 10 - Condition Causing Stratification

# FIELD TEST SHEET

Fan Owner \_\_\_\_\_  
 Fan Location \_\_\_\_\_  
 Fan Nameplate Data \_\_\_\_\_  
 Fan RPM \_\_\_\_\_ Motor Nameplate Data \_\_\_\_\_  
 Motor Test Current \_\_\_\_\_ Voltage \_\_\_\_\_  
 Date \_\_\_\_\_ Tested by \_\_\_\_\_



**Traverse Points for Round Duct**

**Traverse Points for Rectangular Duct**

Barometric Pressure	Average SP Inlet	Temperature °F. Inlet	Average Static Pressure Outlet	Temperature °F. Outlet

Test * Points	SP	VP	√VP
1			
2			
3			
4			
5			
6			
7			
8			
9			
10			
11			
12			
13			
14			
15			
16			
17			
18			
19			
20			
21			
22			
23			
24			
Sum of √VP's			
Avg. of Sum of √VP's			

$$\text{Density}_{\text{inlet}} = \frac{0.075 \text{ lb.}}{\text{ft.}^3} \times \left( \frac{530}{460 + ^\circ\text{F.}} \right) \times \left( \frac{\text{Barometric Pressure}}{29.92} \right) = \text{_____ lbs./ft.}^3$$

$$\text{Density}_{\text{outlet}} = \frac{0.075 \text{ lb.}}{\text{ft.}^3} \times \left( \frac{530}{460 + ^\circ\text{F.}} \right) \times \left( \frac{\text{Barometric Pressure}}{29.92} \right) = \text{_____ lbs./ft.}^3$$

$$\text{Density}_{\text{CFM test}} = \frac{0.075 \text{ lb.}}{\text{ft.}^3} \times \left( \frac{530}{460 + ^\circ\text{F.}} \right) \times \left( \frac{\text{Barometric Pressure}}{29.92} \right) = \text{_____ lbs./ft.}^3$$

$$\text{CFM}_{\text{inlet}} = 1096 \times \text{Duct Area} \times \left( \frac{\text{Avg. of Sum of } \sqrt{\text{VP's}}}{\sqrt{\text{Density}_{\text{CFM test}}}} \right) \times \left( \frac{\text{Density}_{\text{CFM test}}}{\text{Density}_{\text{inlet}}} \right) = \text{_____ CFM}$$

$$\text{VP}_{\text{inlet}} = \left( \frac{\text{CFM}_{\text{inlet}}}{1096 \times \text{Inlet Area}} \right)^2 \times 0.075 = \text{_____ " W.G.}$$

$$\text{SP}_{\text{fan}} = \text{SP}_{\text{outlet}} \times \left( \frac{0.075}{\text{Density}_{\text{outlet}}} \right) - \text{SP}_{\text{inlet}} \times \left( \frac{0.075}{\text{Density}_{\text{inlet}}} \right) - \text{VP}_{\text{inlet}} = \text{_____ " W.G.}$$

$$\text{Single Phase BHP}_{\text{fan}} = \frac{\text{Amps} \times \text{Volts} \times \text{Power Factor} \times \text{Motor Efficiency}}{746} = \text{_____ BHP}$$

$$\text{Three Phase BHP}_{\text{fan}} = \frac{\text{Amps} \times \text{Volts} \times \text{Power Factor} \times \text{Motor Efficiency} \times \sqrt{3}}{746} = \text{_____ BHP}$$

\* A minimum of 24 test points is recommended for round ducts less than 8 feet in diameter and rectangular ducts with areas 24 square feet and less. For larger ducts, more test points are required.

# SAFETY PRECAUTIONS

## A WORD ABOUT SAFETY

Testing, adjusting, and maintaining fan equipment exposes personnel to potential safety hazards. Only experienced mechanics, who are aware of the safety hazards created by moving or rotating parts, should be authorized to work on fan equipment. The proper precautions must be followed to prevent injury from moving parts.

# CAUTION

**THIS MACHINE HAS MOVING PARTS THAT CAN CAUSE SERIOUS BODILY INJURY. BEFORE OPERATING OR PERFORMING MAINTENANCE, THE FOLLOWING PRECAUTIONS MUST BE TAKEN.**

- 1. MAKE SURE ALL MOVING PARTS ARE SHIELDED FROM PERSONNEL AND FALLING OBJECTS.**
- 2. READ THE INSTALLATION AND MAINTENANCE INSTRUCTIONS AS WELL AS THE RECOMMENDED SAFETY PRACTICES MANUAL FURNISHED WITH THIS UNIT.**
- 3. DO NOT OPERATE AT SPEEDS OR TEMPERATURES HIGHER THAN PUBLISHED FOR THE SPECIFIC OPERATING CONDITIONS FOR WHICH THE MACHINE WAS PURCHASED.**

**FAILURE TO TAKE THESE PRECAUTIONS COULD RESULT IN SERIOUS BODILY INJURY AND PROPERTY DAMAGE.**

**98-0250**

The above **CAUTION** decal appears on all **nyb** fans. Air moving equipment involves electrical wiring, moving parts, and air velocity or pressure which can create safety hazards if the equipment is not properly installed, operated, and maintained. To minimize this danger, follow these instructions as well as the additional instructions and warnings on the equipment itself.

All installers, operators, and maintenance personnel should study AMCA Publication 410 - Recommended Safety Practices for Air Moving Devices, which is included as part of every shipment. Additional copies can be obtained by writing to The New York Blower Company, 7660 Quincy Street, Willowbrook, IL 60521-5596.

## ELECTRICAL DISCONNECTS

Every motor-driven fan should have an independent disconnect switch to isolate the unit from the electrical supply. It should be near the fan and must be capable of being locked by maintenance personnel while servicing the unit, in accordance with OSHA procedures. **Do not attempt any maintenance on a fan unless the electrical supply has been completely disconnected and locked.**

## MOVING PARTS

**All moving parts must have guards to protect personnel.** Safety requirements vary, so the number and type of guards needed to meet company, local, and OSHA standards must be determined and specified by the user. Never start a fan without having all safety guards installed. Check regularly for damaged or missing guards and do not operate any fan with guards removed. Fans can also become dangerous because of potential "windmilling", even though all electrical power is disconnected. Always block the rotating assembly before working on any moving parts.

## AIR PRESSURE AND SUCTION

In addition to the normal dangers of rotating machinery, fans present another hazard from the suction created at the fan inlet. This suction can draw materials into the fan where they become high velocity projectiles at the outlet. It can also be extremely dangerous to persons in close proximity to the inlet as the forces involved can overcome the strength of most individuals. **Inlets and outlets that are not ducted should be screened to prevent entry and discharge of solid objects.**

## ACCESS DOORS

# DANGER

**DO NOT OPEN UNTIL THE POWER SUPPLY HAS BEEN LOCKED OFF AND THE SHAFT HAS STOPPED ROTATING.**

**FAILURE TO DO THIS CAN RESULT IN SERIOUS BODILY INJURY.**

**98-0249**

The above **DANGER** decal is placed on all **nyb** cleanout doors. **These doors, as well as access doors to the duct system, should never be opened while the fan is in operation.** Serious injury could result from the effects of air pressure or suction.

Quick-opening doors must have the door-handle bolts securely tightened to prevent accidental or unauthorized opening. Bolted doors must be tightened for the same reason.

## MAXIMUM SAFE SPEED

Safe operating speed is a function of system temperature and wheel design. **Do not, under any circumstances, exceed the maximum safe fan speed published in the nyb bulletin,** which is available from your **nyb** field sales representative.

## SELECTION CRITERIA FOR FAN DAMPERS

### INTRODUCTION

Dampers are the most common volume control device used in fan systems. Low in cost, dampers require little maintenance, easily adjust airflow during operation, and need little space. For these reasons, they are often selected over more complex control systems such as variable frequency drives.

To select the best damper for a particular application, it is necessary to understand the requirements of the application as well as the capabilities of different damper systems.

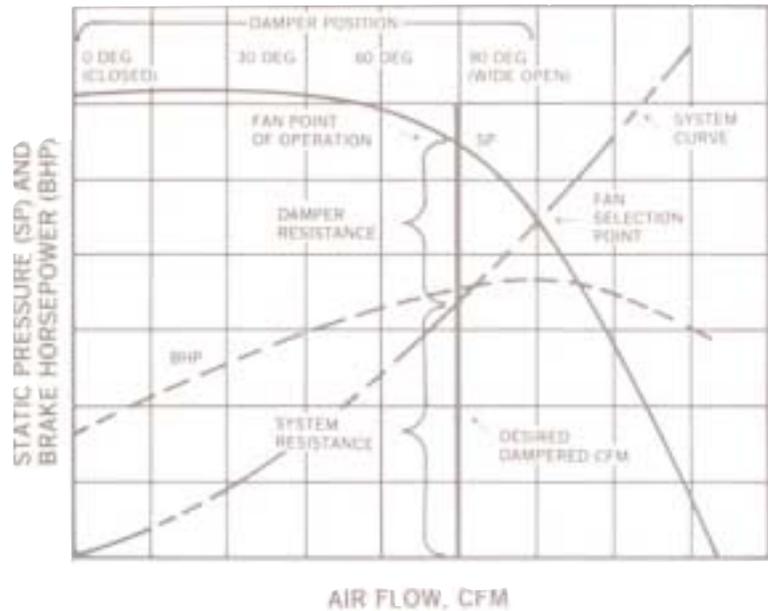
Since dampers may be placed on either side of the fan, they are classified as either inlet or outlet. Both reduce airflow in predictable amounts, but by different means.

Outlet dampers control the air after it has passed through the fan by changing the resistance the fan is working against. Figure 1 shows the effects of various outlet damper settings on a backwardly-inclined fan. It illustrates how the damper controls CFM, static pressure, and its impact on fan BHP.

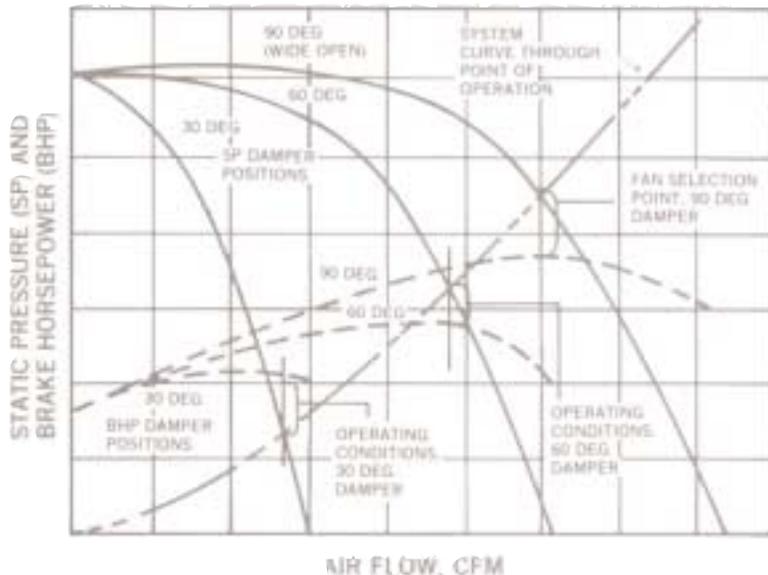
As the outlet damper is closed, the point of operation moves to the left of the selection point along the fan's static pressure curve. Adding resistance with the outlet damper also moves the fan horsepower to the left on its curve. With radial-blade and forward curved-fans, the dampered horsepower will be less than the wide open horsepower as the fan moves to the left on the BHP curves. With backwardly inclined fans, the dampered horsepower may be less, the same, or more than its wide open horsepower, depending on the original point of operation. For more information see Engineering Letter 3.

Inlet dampers affect the air before it enters the fan. External, internal, or inlet box inlet dampers cause the entering air to spin in the same direction as the fan rotation. Because of this, the fan wheel can not develop full output. This results in lower volume and reduced BHP. When a backwardly inclined fan has an inlet damper, it reacts as shown in Figure 2 as the damper vane angle is changed. For each new damper vane position, new SP and BHP curves are generated. The new point of operation is defined by the system in which the fan is installed. The end result is similar to the change that occurs when slowing down an undampened fan.

The horsepower and electrical power savings of this damper make it attractive for systems required to operate at reduced flow rates for extended periods, such as in variable-air-volume systems. While Figure 2 illustrates an inlet damper's effects on a backwardly inclined fan, the same general results are achieved using inlet dampers on any type of centrifugal fan.



**Figure 1** - Static pressure and brake horsepower curves for backwardly-inclined fan with outlet damper. As the damper closes, the point of operation - brake horsepower and static pressure - moves to the left of the original fan selection point to the 90-degrees (wide open) damper setting.



**Figure 2** - Effect of applying inlet dampers to the fan in Figure 1. Separate SP and BHP curves are developed for each vane setting. Fan operating points at these settings are determined by system resistance (points where system curve intersects SP and BHP fan curves).

## TYPES OF OUTLET DAMPERS

The parallel blade arrangement shown in Figure 3 is the simplest, most economical, and most popular type of outlet damper. The cross-sectional area of a wide-open damper is not greatly reduced until the blades have been moved to the 30 degree open position. Consequently, the outlet damper control arm swings through a relatively large arc to reduce fan capacity a small amount. This makes the parallel-blade outlet damper particularly useful when installed on a continuous process system where sensitive control of air volume between wide open and 70% or 80% of wide-open is desired. The large control arm swing also allows predetermined settings of airflow to be repeated accurately. This damper, being the least expensive of the various designs, also makes it the usual selection for systems that require two position damper operation (either full-open or full-closed). Another common application involves cold starts on a "hot" system requiring a reduction in airflow to reduce BHP until the system reaches temperature.

Opposed-blade outlet dampers, as pictured in Figure 4, are used when a straight line relationship between fan volume and control arm swing is desired. In this design, alternate blades turn in opposite directions. Therefore, the change in volume, with respect to the damper position, is proportional to control arm swing.

The opposed-blade damper is usually selected when it is necessary to maintain an even distribution of air immediately downstream from the damper. Figure 5 illustrates the downstream air pattern of an opposed-blade versus a parallel-blade damper. Opposed-blade dampers cost more than parallel-blade models of the same size due to the increased complexity of the linkage required to provide the opposed-blade motion.



Figure 3 - Parallel-Blade Outlet Damper



Figure 4 - Opposed-Blade Outlet Damper

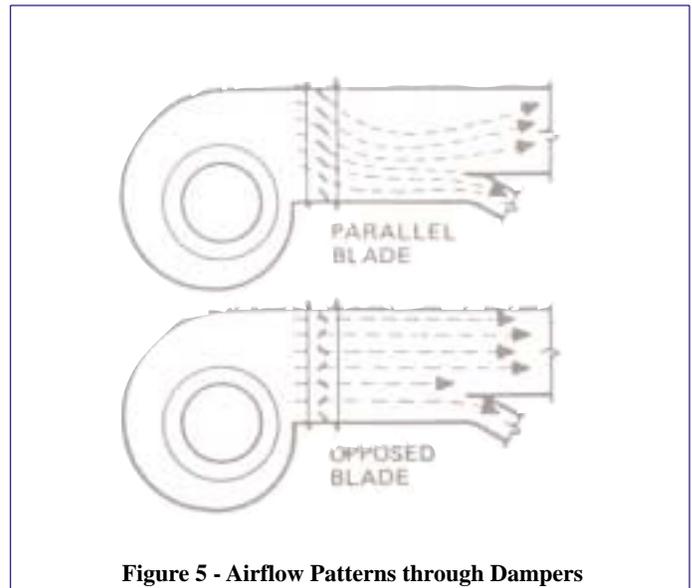


Figure 5 - Airflow Patterns through Dampers

## TYPES OF INLET DAMPERS

Inlet dampers can provide a substantial horsepower savings for fans that are operated at reduced capacity for extended periods of time. Concerns for energy conservation and reduced operating expense make this feature desirable and often mandatory when designing a system.

A good example of how inlet dampers are used to accomplish energy savings can be seen in a typical variable volume heating-cooling ventilation system. In this application much less air is needed for winter heating than for summer cooling. In addition, during summer operation, less air is needed for cooling during the nighttime hours than during the peak daytime hours. Yet, the fan system must be selected for the worst condition/highest air flow. The inlet damper offers the greatest long term savings in VAV applications due to reduced horsepower requirements at reduced volumes.

External inlet dampers, as shown in Figure 6, are mounted external of the fan structure. The configuration is circular with the damper vanes connected to a central hub through pivot bearings. The control linkage is also circular and exposed for easy inspection and maintenance.

Generally, this is the most expensive damper configuration. It is also capable of handling higher velocities and pressures than the internal inlet damper.



Figure 6 - External Inlet Damper



**Figure 7 - Internal Inlet Damper**

The internal inlet damper, pictured in Figure 7, is similar to the external inlet damper with respect to controlling fan performance. The most significant difference is that the internal damper is a self-contained unit furnished as an integral part of the fan inlet cone. This provides considerable space savings and eases installation. The internal inlet-damper design, however, may tend to create some resistance at wide-open, due to the control vanes being in the high velocity region of the fan inlet. Therefore, appropriate airflow reduction factors, as listed in a separate engineering supplement, must be used when sizing a fan with this type of damper. In addition, the damper control linkage is in the airstream on the inside of the fan housing and must be serviced through a cleanout door in the housing.

Inlet-box dampers (Figure 8) are parallel-blade rectangular dampers mounted on an inlet box in such a way that the airflow from the damper produces a vortex at the fan inlet. Inlet-box dampers are generally preferable on fans equipped with inlet boxes and have the same general control requirements as standard inlet dampers. Because the bearings are not in the airstream, inlet-box dampers are often used in airstreams that contain some particulate. Predicting the exact flow reduction with damper angle varies with damper types and products. Normally this is not a requirement since flow should be established using manual reference or feedback from automatic control systems. For all inlet-vane dampers, vane angle versus flow relationship will change when dampers are applied to wheels that have been narrowed to establish specific capacities at direct drive speeds.



**Figure 8 - Inlet-Box Damper Mounted To Inlet Box**

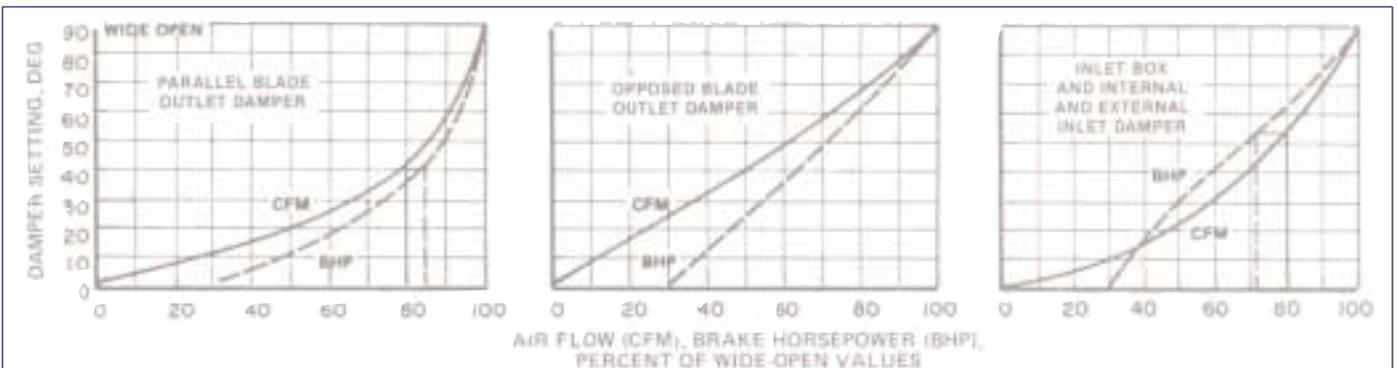
Inlet dampers typically improve the stability of most products because they control the flow through the fan inlet. At extreme dampering, about 30° open, inlet dampers can no longer create a vortex and become essentially a blocking damper. This causes the fan to operate far to the left on its curve. When this happens, a fan is subject to the same problems of instability as if the point of rating was established by an outlet damper or other system changes.

**COMBINED INLET AND OUTLET DAMPERS**

Occasionally it is desirable to save more power at reduced capacity while maintaining very sensitive control. In this case, the fan may be equipped with both inlet and parallel-blade outlet dampers. With the outlet damper set at wide-open, the inlet damper is set to give just slightly more air than needed. Exact air delivery is obtained by adjusting the outlet damper. Because the outlet damper vanes require a lot of movement to achieve a slight change in air delivery, sensitive control is achieved.

**PERFORMANCE COMPARISON**

Figure 9 shows the effects of damper settings on airflow and brake horsepower for parallel and opposed-blade outlet dampers, and inlet and inlet-box dampers. These plots represent generalizations of damper effect on fan performance and can be used to compare one type to another.



**Figure 9**

Effect of vane setting on airflow and power for various damper types. When a parallel-blade outlet damper is set for 80 percent of wide-open capacity, the damper setting is 40 degrees, and the fan operates at 85 percent of wide-open horsepower. However, with an inlet damper, operation at 80 percent of wide-open requires a 53 degree damper setting and 72 percent of wide-open horsepower. Note: These curves are representative, not precise. See text.

	<b>Parallel-Blade Outlet Damper</b>	<b>Opposed-Blade Outlet Damper</b>	<b>External and Internal Inlet Dampers</b>	<b>Inlet-Box Damper</b>
1. Cost	Least costly.	1.1 to 1.2 times as much as parallel blade.	Internal - 1.5 to 2.5 times as much as parallel-blade. External - 3 to 4 times as much as parallel-blade.	1.3 to 1.4 times as much as parallel-blade; combined with inlet box 3 to 4 times as much as parallel-blade.
2. Control	Best for full-open or closed requirements or for fine control between 80% to 100% full-flow.	Best for systems where air volume is changed over a wide range and a straight line relationship of volume to control arm swing is desired.	Same as opposed-blade outlet damper.	Used on fan inlet box. Can be used with some particulate in airstream.
3. Horsepower	<b>Depends upon characteristic BHP curve; Backwardly inclined - same, more, or less than wide-open, FC and Radial - less than wide-open.</b>		Power consumption at reduced air volumes is less than with outlet dampers.	Same as inlet damper
4. Air flow after fan	Throws air to one side.	Distributes air evenly.	No effect.	No effect.

**Figure 10 - Comparison of Inlet and Outlet Dampers**

## SUMMARY

Each system has its own requirements with respect to the control of air volume. System designers must be aware of not only first cost considerations but, more importantly, of the long term operating savings that can be achieved by a properly engineered system. Each system also imposes limits on which

dampers can be used with respect to fumes, control sensitivity, and temperature. No one damper design is best for all applications. Figure 10 provides a comparison to help the designer recognize some of the factors to be considered in damper selection.

# **Upgrade to ePTFE Membrane Technology to Improve Processes, Lower Operating Costs, Reduce Emissions, and Extend Filter Bag Life**

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Key words: Steel, ePTFE membrane, Emissions, Cleaning, Cost savings, Filter life

## **INTRODUCTION**

Throughout their existence, steel mills have been continually tasked to produce more product using less resources and funds. This has become more pronounced with the increased cost of scrap in recent months. Steel producers are also facing ever-increasing pressure from public and government entities to reduce emissions and other means of environmental impact. With the pending MACT regulations, steel producers are now faced with the challenge of producing high quality steel at competitive prices, while doing so with the least detrimental effects to the environment.

The development of advanced filtration media technology in the early 1990's, and recent advances in its application, is providing steel producers with an opportunity to improve several aspects of their plants' dust collection operations by upgrading to fine filtration.

## **ADVANCED TECHNOLOGY**

Expanded Polytetrafluoroethylene (ePTFE or Teflon®) membrane filter bag technology involves expanding PTFE resin into a membrane composed of millions of microscopic pores in a three-dimensional web-like structure. The membrane layer is then laminated to traditional filter fabrics (woven or felted) through a highly controlled process of heat and pressure. The resulting media provides substantial benefits over traditional fabrics including near-zero emissions, a reduction in operating and cleaning costs, an increase in production, improved spark resistance, and longer bag life.

Upgrading to ePTFE membrane technology involves an increase in filter bag costs. However, the benefits realized from this technology, individually or as a whole, often provide economic justification for its implementation.

## **REDUCED EMISSIONS**

Traditional filter fabrics have a relatively open weave structure inherent by their manufacture, and therefore require an initial layer of dust for proper filtration. The differential pressure across the media is a good indicator of the characteristics of this layer. For typical applications in a steel production plant, the minimum recommended differential pressure for traditional fabrics that represents a suitable dust cake layer is 4 inches water column (w.c.).

Emissions are often observed with traditional media when the bags are new or have just been cleaned, as the initial layer has not yet formed. Because the resistance is low at these conditions, the air volume moved by the fans increases, causing the media to operate at a higher air-to-cloth ratio (filtration velocity) than at the design air volume. In these conditions, the dust can be driven into or through the media. During startup of a system with new bags, a dust cloud can often be seen at the discharge of a fan or stack until a suitable layer of dust forms.

Unlike a traditional fabric filter bag (felted or woven), an ePTFE membrane filter bag does not require a layer of dust cake to achieve its maximum efficiency. All of the dust collection occurs on the membrane surface, without penetration into the fabric. This is especially beneficial during start-up.

Because all of the dust remains on the surface, ePTFE membrane filtration is often referred to as “surface filtration”. Conversely, filtration with traditional fabrics is often referred to as “depth filtration” because of the need for a dustcake and the fact that dust often embeds into the fiber matrix. Figures 1 and 2 illustrate the difference between depth filtration and surface filtration.



Figure 1: Depth Filtration

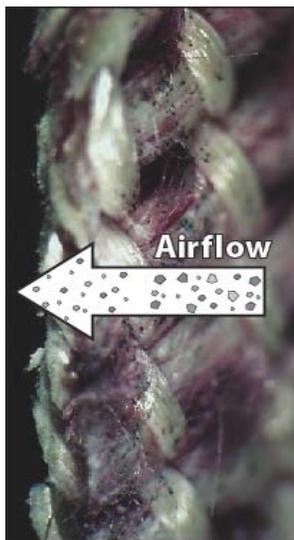


Figure 2: Surface Filtration

Membrane filtration technology is the most efficient media available for dust collection applications. Some manufacturers’ offerings can provide near 100% efficiency to 0.5 microns.

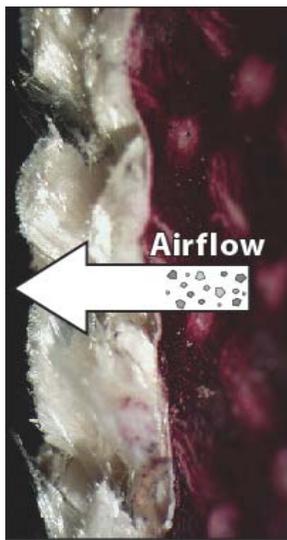
This enhanced efficiency is provided by the 3-dimensional microporous structure of the membrane. These pores are small enough to capture virtually any particulate, yet large enough to allow the passage of airflow with limited resistance.

Figures 3 and 4 show the results of a smoke bomb test where filtration medias were exposed to submicron smoke particles in a controlled air stream. The smoke particles easily penetrate and bleed through the traditional media, but are retained at the surface of the media laminated with ePTFE membrane.



Angled Top View

Figure 3 – Smoke particles penetrate traditional filter media.



Angled Top View

Figure 4 – Smoke particles collect on surface of ePTFE media.

Several steel facilities are switching to membrane technology to prepare for the pending MACT requirements. The cost of this environmentally sound alternative is offset by the financial and production related benefits described in the following sections.

## REDUCED OPERATING AND CLEANING COSTS

Production facilities are often evaluated or rated (internally) on the cost per ton produced. These costs may include electrical energy for ancillary systems, including ventilation systems, as well as downtime for system trouble. Membrane technology provides several advantages against the cost of operation, including electrical costs for the main ventilation fans and the electrical cost for cleaning the media. It also provides more stable system operation, resulting in less downtime and longer production runs.

A membrane filter bag's microporous, highly efficient structure prevents dust from penetrating into the base fabric, a phenomena that inherently occurs with traditional fabrics. Embedded dust blocks airflow, thereby increasing the resistance through the media. This becomes significantly more pronounced with time in operation. In addition, sticky and cohesive dusts do not readily build up on ePTFE-derived membrane filter media, further reducing the potential pressure drop.

Membrane filter media is especially beneficial in baghouses for systems with substantial moisture issues. High levels of moisture can mix with the collected dust to form a cohesive dustcake or "mud" that cannot be readily removed from traditional media. Baghouses with these issues typically run at high differential pressures (8" w.c. or greater) and experience short bag life from aggressive cleaning.

With membrane filter media, these issues are nearly eliminated as the collected material is easily discharged due to the inherent non-stick properties of the ePTFE-derived surface.

Compared with traditional media, membrane technology often runs at a significantly lower differential pressure, sometimes 2 to 4 inches water column. This can significantly lower the electrical costs associated with running the baghouse.

### Comparative Cost Example

A 2,392-bag reverse-air dust collector with 11.5-inch diameter polyester bags, each 362.5 inches long, vents an EAF's 4<sup>th</sup> hole and the canopy at a northern steel plant. There are (13) compartments of (184) bags to filter a total design air volume of 639,000 ACFM. The baghouse is a positive pressure unit.

The system consists of (3) main ventilation fans, (1) DEC fan, and (2) reverse air fans.

#### Main system fans

The design operating point for each fan is 213,000 acfm @ 11" SP, requiring 580 BHP. Measured amperage on the main system fans matched with the fan curve provide the following parameters:

Fan ID	AMPS	BHP	CFM	SP
<i>North</i>	104	475	163,000	13.2"
<i>East</i>	126	576	210,000	11.1"
<i>South</i>	150	685	272,000	7.2"

The average baghouse differential pressure was 6.5 inches w.c.

Based on an electrical cost of \$0.04 per kWh, the monthly operating cost for the three fans at design is \$37,383.

If the differential pressure in the baghouse were reduced by 1 inch w.c., the monthly cost would decrease to \$33,990; a 2-inch decrease would result in a reduced cost of \$30,600.

#### Reverse air fans

The reverse-air fans should each pull about 25,000 acfm (representing 1.5 times the compartment cloth area), at a static pressure 1" greater than the compartmental differential pressure when online. These numbers are based on common industry recommendations.

The monthly reverse-air cost associated with cleaning the current media (@ 7.5" SP) is \$1,016 when assuming the reverse-air fans operate 50% of the time.

The energy required for cleaning membrane bags is also substantially less than that required for traditional fabric bags. Typically, membrane filtration provides reductions in both the cleaning pressure and the frequency at which the bags must be cleaned to maintain the system flow.

Assuming that the effective operating time for the reverse-air fan reduces to 30% when cleaning membrane bags, the monthly cost to clean the baghouse reduces to \$528 if the baghouse differential pressure reduces 1" w.c., and \$447 if the baghouse differential pressure reduces 2" w.c.

This value may be further reduced if the bags are cleaned based on differential pressure instead of continuous cycles.

Table 1 documents the comparison cost of operation (main air fans and reverse-air fans) between traditional fabric technology and membrane technology. Economic studies based on these principles are often used to justify the expenditures for upgrading the media.

<b>Table 1: Comparison of Monthly Operating Cost, Traditional Fabrics vs. Membrane Technology</b>		
Parameter	Traditional Fabric	Membrane Technology (2" reduction)
Main Fan Energy (monthly)	\$37,383	\$30,600
Cleaning Energy (monthly)	<u>\$1,016</u>	<u>\$447</u>
Total Monthly Cost	\$38,399.00	\$31,047.00
Total Annual Cost	\$460,788.00	\$372,564.00

It is important to note that the fan must be controlled to match the reduced operating requirements in order to achieve cost savings. This can be accomplished by implementing speed control (using a variable frequency drive), or with an inlet damper.

### **PRODUCTION IMPROVEMENTS**

Many users of membrane technology take advantage of the reduced differential pressure in the baghouse by allowing the fan to move more air volumes. This increased air volume allows for better dust capture at the furnace and other pickup points, but the significant benefit is often recognized in shorter tap-to-tap times because of better dust capture in the melt shop.

One customer reported to BHA that with a partial switch to membrane technology, they were able to reduce the tap-to-tap time from 40 to 50 minutes down to 35 minutes. For a 180-ton furnace, this allowed for at least 4 more heats per day, or 720 more tons of steel.

Membrane technology also provides more stable flow and less downtime, allowing for more production time. Part of this reduced downtime is provided by longer bag life, which is discussed in a later section of this paper.

### **HIGHER TEMPERATURE FABRICS AND MEMBRANE TECHNOLOGY**

Several facilities that have traditionally utilized polyester media (typically knit bags without seams) are switching to higher temperature fabrics laminated with ePTFE membrane in order to obtain several benefits. The benefits include less total air volume handled by the fan, more air flow from the process, less cooling water usage (where applicable), and less aggressive operation needed in the baghouse.

Polyester media has been considered the "workhorse" for the industry, providing reasonable filtration efficiency at the lowest fabric cost. However, the two biggest drawbacks to polyester media are the lower temperature rating (275° F) and the potential for damage from moist heat (hydrolysis).

Facilities often bleed in ambient air to match the maximum allowable temperature for the media they are using. For low temperature fabrics, the ambient air is a significant portion of the total volume going to the baghouse.

Table 2 shows the percentages of make up air (ambient plant air at 100 degrees F) and process air required for polyester, aramid (or Nomex®), and fiberglass media for a range of operating temperatures.

**Table 2: Comparison of Dilution Air and Process Air**

Process Temperature (degrees F)	Polyester T <sub>max</sub> = 275 deg F		Aramid T <sub>max</sub> = 400 deg F		Fiberglass T <sub>max</sub> = 500 deg F	
	Process Air	Dilution Air	Process Air	Dilution Air	Process Air	Dilution Air
	300	88%	12%			
350	70%	30%				
400	58%	42%	100%			
450	50%	50%	86%	14%		
500	44%	56%	75%	25%	100%	
550	39%	61%	67%	33%	89%	11%
600	35%	65%	60%	40%	80%	20%
650	32%	68%	55%	45%	73%	27%
700	29%	71%	50%	50%	67%	33%
750	27%	73%	46%	54%	62%	38%
800	25%	75%	43%	57%	57%	43%

As illustrated in Table 2, higher temperature fabrics allow for a larger percentage of ventilation air from the furnace than traditional fabrics, which can greatly improve production. Conversely, higher temperature fabrics require less total flow in the system for the same volume from the process. Some plants may wish to run with more air from the process, while still operating with less total flow.

Facilities using water-jacked duct or in-duct spray systems can reduce the amount of water used to cool the gas stream. Using less water reduces the potential for problems with excess moisture including high differential pressure across the media and corrosion in the baghouse and the system components.

Finally, the production department can run with higher melt temperatures in the furnace, resulting in shorter tap-to-tap times and increased production.

These various strategies often show significant justification for higher-temperature membrane technology, including the fact that the required fan energy can decrease because a lower volume of higher temperature air is moved through the system.

It is important to note that the higher temperature woven fabrics (aramid, fiberglass) are not as efficient as their polyester alternatives. For this reason, facilities that are considering switching to these fabrics should also consider membrane technology to prevent emission issues.

### **LONGER BAG LIFE**

Membrane filter bags can endure a longer operating life than traditional fabric bags. Any filtration fabric will undergo wear during cycles of operation and cleaning, but the fact that membrane bags require reduced cleaning energy and less cleaning cycles to maintain the draft in the system typically extends their life past traditional fabrics.

Another key aspect to longer bag life is the fact that collected particulate remains on the surface of the membrane filter bags and does not penetrate the fiber matrix. Embedded particulate often requires increasing cleaning frequency and energy over time, and naturally causes internal bag abrasion. The cumulative effect is weakened fibers that lead to bag failures.

Finally, the membrane surface is more resilient to sparks than traditional fabric, as the sparks often extinguish on the surface rather than enter the fiber matrix and form holes. Of course, membrane technology is not a cure for every spark problem; effort must be made to contain them at the point of generation.

When comparing fabric technologies, bag life is an important component of the financial analysis. The cost of a typical bag change out can be significant when considering production downtime and labor costs. By investing in a filter media that will last longer, the plant is able to produce more with less maintenance activities in the baghouse.

## **SUMMARY**

As with any technology upgrade, ePTFE membrane filter bags must be applied correctly and are not ideal for every situation. With proper application and installation, membrane technology can improve many situations to provide more stable operation with decreased operating costs and increased production, while limiting the environmental impact of your operations.

Furthermore, as with any fabric filter purchase, it is very important to qualify vendors and their product supplier to ensure the best return on investment.

## Renfert, David A (GE Energy)

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**From:** Hamilton, Tom [thamilton@nyb.com]  
**Sent:** Monday, October 31, 2005 9:28 AM  
**To:** Renfert, David A (GE Energy)  
**Subject:** RE: Requested permission

Dear David,

No problem. You have authorization to use any published nyb materials that you deem fit.

Tom

-----Original Message-----

**From:** Renfert, David A (GE Energy) [mailto:David.Renfert@ge.com]  
**Sent:** Monday, October 31, 2005 8:54 AM  
**To:** Hamilton, Tom  
**Subject:** RE: Requested permission

No apologies necessary - I've been really swamped myself. Would you also mind if I used portions of the Engineering Letters in my Appendix? All of our sales offices around the world will have a copy with appropriate portions highlighted.

Thank you.

David Renfert, PE

-----Original Message-----

**From:** Hamilton, Tom [mailto:thamilton@nyb.com]  
**Sent:** Wednesday, October 26, 2005 8:41 AM  
**To:** Renfert, David A (GE Energy)  
**Cc:** Baty Sr., Bob  
**Subject:** RE: Requested permission

Dear Mr. Renfert,

I am embarrassed to say that I misplaced your original e-mail for over one month. I apologize for the delay.

We would be honored to have you use the nyb selection software output in your thesis.

Best wishes in your studies.

Tom

-----Original Message-----

**From:** Renfert, David A (GE Energy) [mailto:David.Renfert@ge.com]  
**Sent:** Monday, September 19, 2005 7:16 PM  
**To:** Hamilton, Tom  
**Cc:** Baty Sr., Bob  
**Subject:** Requested permission

Good Evening,

I am an engineer with GE Energy in Kansas City, MO, and have worked with Bob Baty, Sr. at Aircorp (NYB local rep). He suggested I contact you with a request.

I am finishing my project for my Masters program. The project will result in a cost analysis procedure and guidebook for our offices around the world. This book will have substantial background on fan selection, etc. To that end, NYB's fan sizing program can provide some excellent graphs and charts for the project.

I would like to use the output of these graphs and import them into my literature. I would also like to put portions of the Engineering Letters for reference. Proper acknowledge will be included in the documentation, including a supportive statement regarding the invaluable help of your company and local representative.

I would appreciate your permission to use the aforementioned tools. Please contact me by e-mail if you have any questions, as I will be traveling all week,

Regards,

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