

Fan Applications & System Guide

Ventilation Calculation Methods

Airflow for general ventilation can be calculated by;

- Area Method
- Air Change Method
- Occupancy Method

- **Area Method** – Derives the ventilation rate from the area of the space (in square feet) to be ventilated multiplied by ventilation rate per square foot.

Example: For residential bathrooms up to 100 sq. ft. in area, HVI recommends an exhaust rate of 1 CFM per square foot. A bathroom is 6' x 10' or 60 sq. ft.

Multiply 1 cfm x 60 sq. ft. for a flow rate of 60 cfm.

- **Air Change Method** - Derives the ventilation rate from the volume of the space (in cubic feet) to be ventilated multiplied by the number of total air changes in one hour.

Example: For an auditorium the suggested air change rate is 4 to 15 air changes per hour. An auditorium is 80' x 90' with 20' ceiling or 144,000 cu. ft. Use 10 air changes per hour.

$$\text{CFM} = \frac{144,000 \text{ cu. ft.} \times 10 \text{ AC p/ hr}}{60} \quad \text{CFM} = 24,000$$

- **Occupancy Method** - Derives the ventilation rate from the number of people that will occupy the space at any given time.

Example: For an office, the recommended ventilation rate is 20 cfm per person. The occupancy of a general office is one person per 80 to 150 sq. ft. An office is 40' x 60' or 2,400 sq. ft.

Occupancy = 2,400 sq. ft./150 sq. ft p/ person

$$\text{CFM} = 16 \text{ people} \times 20 \text{ cfm per person} \quad \text{CFM} = 320$$

Heat Removal Method

When the temperature of a space is higher than the ambient outdoor temperature general ventilation can be used to provide cooling. What is needed to calculate CFM is the amount of heat to be removed in BTU/hr, the desired indoor temperature and design outdoor dry bulb temperature.

Example: 200,000 BTU/hr to be removed, 70 degree desired indoor temperature and 90 outdoor dry bulb temperature.

$$\text{CFM} = 200,000 \text{ (BTU/hr)} / (1.08 \times 90 - 70) \quad \text{CFM} = 9,260$$

(**Note:** rule of thumb outdoor / indoor temperature differential is 20 degrees)

Applications

Listed below are ventilation rates for some common applications. Where more than one method is shown use the method that results in the higher airflow rate.

Application	Area Method	Air Change Method	Occupancy Method		Other
			cfm/ person	sq ft/ person	
General Ventilation	1 cfm				
Attic Ventilation	.7 cfm	10			
Auditoriums		4 to 15			
Banks		4 to 10	20	50 to 150	
Battery Charge Room	1.5 cfm	4			
Boiler Room		15 to 30			
Chemical Storage Area	1.5 cfm				
Class Room			15	0.05	
Computer Room		15 to 20	20	80 to 150	
Conference Room			20	0.05	
Corridors	.05 cfm				
Churches			20	5 to 20	
Elevators	1 cfm				
Electrical Room	2 cfm	10			5 cfm / KVA of transf.
Fire Station		4 to 10	20	100 to 500	
Garage (Parking)	1.5 cfm	5 to 15			
Garage (Repair)		6 to 30			5000 cfm / car
Gymnasium		6 to 30			
Hospital (Patient Rooms)			25	0.01	
Kitchen Hood (Charbroiler)					300 cfm x linear. ft. .
Kitchen Hood (Island Type)					150 cfm / sq. ft.
Kitchen Hood (Wall Mounted)					100 cfm / sq. ft.
Kitchen (General Ventilation)	1.5	12 to 15	15	50 to 150	
Libraries / Museums			20	30 to 100	
Locker Room		12 to 30			
Manufacturing (Light)		5 to 10			
Manufacturing (Heavy)		10 to 20			
Mechanical Room	2 cfm				
Mech. Room (Combustion Air)					8 cfm / BHP
Mech. Room (Comb. Exhaust)					2 cfm / BHP
Medical / Dental Office		8 to 12	20	50 to 150	
Municipal Buildings		4 to 10	20	50 to 150	
Police Station		4 to 10	20	100 to 500	
Retail Store	.3 cfm		15	15 to 75	
Supermarket			15	50 to 100	
Toilet Room (Public)					75 cfm p/ WC & urinal
Toilet Room (Residential)	1 cfm	8			
Warehouse Ventilation	1 cfm	6 to 15			
Welding Operation					2500 cfm p/ welder

Systems

General Ventilation

- Plan and Spec Jobs - required airflow and system static pressure is calculated by the engineer and is listed on the fan schedule.
- Retro Fit and Design Build Jobs - the system static pressure can be calculated using the required airflow, the duct size, duct layout and total duct length and duct friction loss chart (Friction Loss Chart needed).
- When the duct layout is not known the following rules of thumb can be used to estimate system static pressure
 1. Non Ducted Applications - .10" to .25"
 2. Ducted Applications - .20 to .40 per 100 feet of duct (based on duct velocity of 1000 to 1800 feet per minute). Fittings .08" per fitting (elbows, register, grille, damper)

Example: Retrofit a fan for a mechanical room. Area of room is 20' x 60' or 1,200 sq. ft. Fan is ducted to roof with 16" x 16" duct. Total duct run is 25'. There are 2 elbows and a grille in the system.

Required Air Flow - 2,400 cfm. (2 cfm p/ sq. ft. x 1200 sq. ft.)

Duct Velocity – 1,400 fpm (air flow / duct area)

Duct Losses - .08" (.30" / 100 ft of duct x 25')

Fitting Losses - .24" (.08" x 3 fittings)

Estimated SP. = .08" + .24", round to nearest 1/8". Size Fan for 2,400 cfm @ .375" sp

- Another rule of thumb that can be used if number of fittings is not known is;

Estimated SP. = $1.5 \times \frac{\text{System Length (ft.)}}{100} \times \frac{\text{Friction Rate (Inches W.G.)}}{100}$ *

*Friction Loss Chart needed

100

100

Kitchen Hoods

- Use .625 to 1.50" sp.
- Minimum duct velocity 1,500 fpm
- Make up air 80% of exhaust air

Sound Level Guidelines

Application	Sones
Bathroom	13 to 18
Conference Room	1.7 to 5
Corridors	9 to 13
Church	1.5 to 5
Hospital Private Room	1.7 to 5
Hospital Ward	2.5 to 9
Kitchens	10 to 18

Application	Sones
Manufacturing (Light)	12 to 36
Manufacturing (Heavy)	20 to 50
Office (Open)	4 to 12
Office (Professional)	3 to 9
Post Office	4 to 12
Schools & Classrooms	2.5 to 8

Basic Fan Laws

Fan Laws are equations used to predict a fan's performance at other conditions. Once a system is installed if the CFM, SP and BHP are known (by measurement) the CFM, SP and BHP at every other point along the system curve can be calculated.

VARIATION	FAN SPEED CHANGE	DENSITY CHANGE
VOLUME	Varies DIRECTLY With Speed Ratio $cfm2 = cfm1 (rpm2 / rpm1)$	No Change
PRESSURE	Varies with SQUARE of Speed Ratio $P2 = P1 (rpm2 / rpm1)^2$	Varies DIRECTLY with density Ratio $P2 = P1 (D2 / D1)$
HORSE POWER	Varies with CUBE of Speed Ratio $hp2 = hp1 (rpm2 / rpm1)^3$	Varies DIRECTLY with density Ratio $hp2 = hp1 (D2 / D1)$

Example: Fan is selected at 4,000 CFM @ .50" SP, 825 RPM and .72 BHP. Balancer reports 3,000 CFM @ .375" SP, 825 rpm and motor operating at nameplate full load amps.

- Calculate the required RPM using fan laws
 $CFM2 / CFM1 = RPM2 / RPM1$
 $4,000 / 3,000 = RPM2 / 825$, **New RPM = 1100**
- Calculate the new SP using fan laws
 $SP2 / SP1 = (RPM2 / RPM1)^2$
 $SP2 / .375" = (1100 / 825)^2$, **New SP = .67**
- Calculate the new BHP using fan laws
 $BHP2 / BHP1 = (RPM2 / RPM1)^3$
 $BHP2 = .75$ because fan motor was operating nameplate amps
 $BHP2 / .75 = (1100 / 825)^3$, **New BHP = 1.78**

The system SP is actually higher than calculated. To meet the operation point at the new SP the fan will need to operate at 1,100 rpm and will require a 2 hp motor.

Air Density Correction

Fan performance tables are based on standard air conditions. An installation's altitude and the air temperature will affect the air density. Use the Air Density Correction Factors (df) listed below to determine the corrected fan performance. Correction for temperatures between 40 deg. F and 100 deg. F and/or elevations between – 1,000 ft. and +1,000 ft. are generally not required.

Air Density Correction Factors (df)

Altitude (Ft)	TEMPERATURE (Degrees F)								
	70	100	200	300	400	500	600	700	800
0	1.00	0.95	0.80	0.70	0.62	0.55	0.50	0.46	0.42
1000	0.96	0.91	0.77	0.67	0.59	0.53	0.48	0.44	0.41
2000	0.93	0.88	0.75	0.65	0.57	0.51	0.46	0.42	0.39
3000	0.90	0.85	0.72	0.62	0.55	0.49	0.45	0.41	0.38
4000	0.86	0.82	0.69	0.60	0.53	0.48	0.43	0.39	0.36
5000	0.83	0.79	0.67	0.58	0.51	0.46	0.42	0.38	0.35
6000	0.80	0.76	0.64	0.56	0.49	0.44	0.40	0.37	0.34
7000	0.77	0.73	0.62	0.54	0.48	0.43	0.39	0.35	0.32
8000	0.74	0.70	0.60	0.52	0.46	0.41	0.37	0.34	0.31
9000	0.71	0.68	0.57	0.50	0.44	0.39	0.36	0.33	0.30
10000	0.69	0.65	0.55	0.48	0.42	0.38	0.34	0.31	0.29

Example: A fan is selected to deliver 4,000 CFM @ .25" SP. Fan will be installed 5,000 ft above sea level; air temperature is 70 deg. F. From the **Air Density Correction Factor** table above the density correction factor (df) is determined to be .86 by using the fans operating altitude.

Divide design SP by the Air Density Correction Factors (df) $.25''\text{sp} / .86 = .29''\text{ sp}$

It is determined by from the fan performance table that the fan must operate at 774 rpm to develop the design CFM at the corrected SP at 5,000 feet above sea level at the operating temperature of 70 deg. F.

From the fan's performance table it is determined that at standard conditions .51 BHP is needed meet the design point. This is corrected to conditions at altitude by multiplying the BHP by the **Air Density Correction Factor**

.51 BHP x .86 = .44 BHP

The final operating conditions at altitude are determined to be 4,000 CFM @ .25" SP, 774 rpm and .44 BHP