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Multiple Fan Systems - Fans in Series and Parallel

In most fan systems a single fan is selected for the required system design rating. Various methods of control are employed such as dampers, variable inlet vanes, variable pitch blades or variable speed to meet other operating points defined by the system duty cycle. This may even include an allowance for future expansions.

Sometimes it is advantageous to use more than one fan in a system. The fans may be located in close proximity to each other such as mounted on a common shaft, or separated by quite a distance such as a supply and exhaust fan. Most often there is some compelling reason to use more than one fan in a single system.

1. One fan may be too large and not fit into the desired space, or it may weigh too much if supported on upper levels.
2. The required operating range of the system may necessitate multiple fans instead of one large fan controlled over a wide operating range. Multiple fans for capacity control may be more economical if cost of operation is critical, especially at very low flow rates for long time intervals.
3. Supply and exhaust fans operating on opposite ends of a system decrease the pressure build-up in a duct or space compared to a single fan. It is usually easier to control a zero point location or maintain low pressures (such as the draft over a fire in a boiler) if supply and exhaust fans are used.
4. Critical systems are often equipped with redundant or back-up fans in case of a fire or accident in a tunnel or some other emergency that requires a sudden increase in flow. Redundant fans are also used to eliminate down-time during fan maintenance.
5. Some systems for process applications may require pressures that are greater than a single fan can produce or when noise may be a special concern. When this occurs, two fans are placed in series with each taking about one-half of the pressure.

Rating Two Fans in Series

Two fans in series are normally rated as a single unit in order for AMCA rating definitions and practices to apply. To simplify selection and control, two fans of the same size are typically used with the required flow rate defined by the inlet conditions of the first fan. The combined total pressure across both fans will be the sum of the individual total pressure of each fan. Total pressures are used instead of static pressure because the fans can actually be different sizes and a change in fan or connecting duct areas has an influence upon static pressure values.

There may also be by-pass ductwork around the second fan if only one fan is run for a period of time. These losses must be added to the normal system resistance requirements.

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What is the quickest and easiest way to select two fans in series?

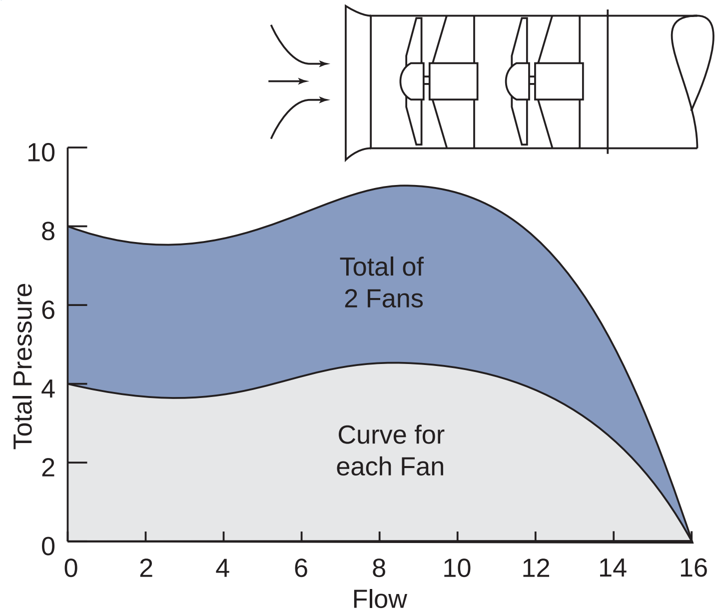
1. Establish the system requirements in terms of total pressure. If they are known only in terms of static pressure, total pressure values may be calculated by adding the velocity pressure corresponding to the velocity passing through the outlet of the second stage fan to the system static pressure requirements.

2. If axial fans or inline fans are being considered, select each fan for the flow rate required and one-half of the system total pressure requirements.
If centrifugal fans are being considered, select each fan for the flow rate required and one-half of the system total pressure requirements plus an allowance for interconnecting ductwork losses, typically one inch of total pressure.

It must be realized that the above selection process is approximate in that the actual individual performance of each fan is not the same. Both fans will handle the same mass flow of air but not volumetric flow rate. This is the result of differences in the inlet densities of each fan caused by differences in the inlet absolute pressures and differences in the temperatures resulting from the possible heat of compression or motor heating etc. by the first stage fan.

The greatest significance is that the rating process can be simplified by making sure the system requirements are in terms of total pressure and that the fans are selected using total pressure.

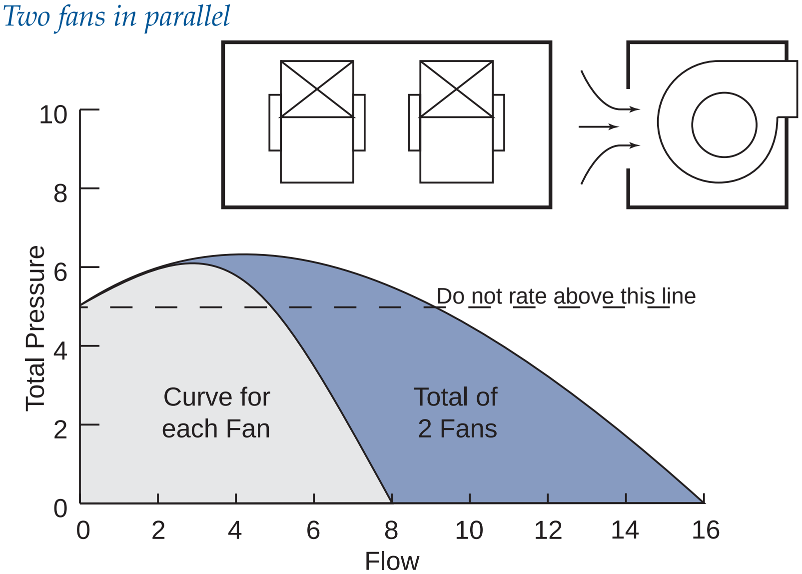
Two fans in series



Rating Fans in Parallel

In most instances fans in parallel will be in some form of plenum application. Unlike fans in series when typically only two fans are involved, parallel fan applications may use multiple fans. In this case the selection process is straight forward in that each fan will be selected for the same static or total pressure with the flow rate being the total flow divided by the number of fans. Use care when selecting

Two fans in parallel



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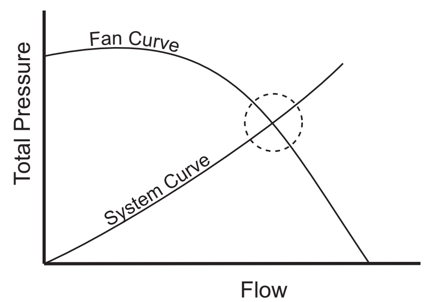
fans in parallel to ensure that the system resistance remains on a stable portion of the fan curve at all times. This is particularly true when the fans have a pronounced surge area or a dip in the fan curve and some form of control is applied. A good rule of thumb is to ensure that the operating point with all fans running is no higher than the lowest pressure in the dip. This minimizes the possibility that the fan will hunt back and forth across the peak of the curve looking for an operating point. This policy also minimizes the likelihood that the fans will experience unequal loading causing differences in motor load or creating unequal velocity profiles within the plenum which may result in a system effect.

Additional Considerations

When a system depends upon more than one fan for proper operation, consideration must be given to those times when only one fan is running. This may be during start-up, during repairs or as part of a flow control scheme based upon the number of fans running.

1. **Start-up conditions** - In general, as long as the system curve and the fan curve always

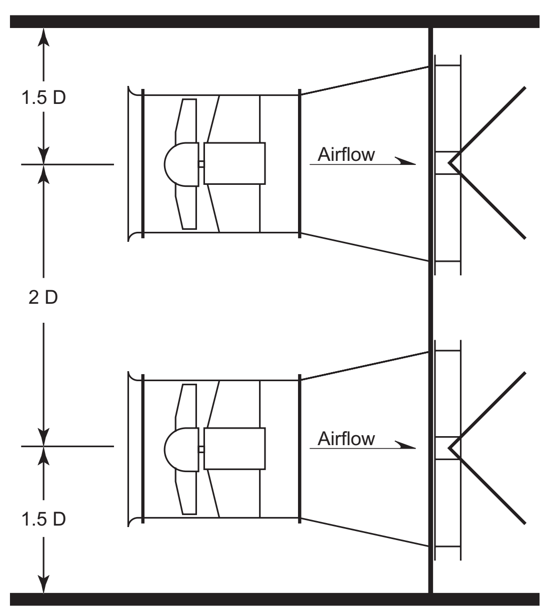


intersect at a stable operating point, no problem should be encountered during start-up or during continuous operation. Centrifugal fans or fixed pitch vane axial fans should be able to be started individually or together. It may be advisable to close inlet vanes or dampers during start-up to minimize horsepower requirements. Once the fans are up to speed, the controls may be opened.

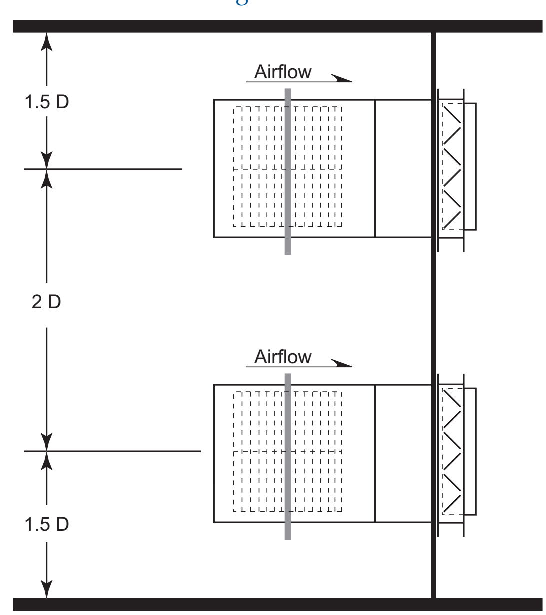
Variable pitch vane axials, which have a dip in the fan curve, should be started with the blade pitch reduced and then opened up once at speed. If the first vane axial has been started, it is best to reduce its blade pitch and start the next fan at the same pitch. When both fans are up to speed and aerodynamically balanced, increase the blade pitch of both fans to the desired operating condition.

Fans in parallel should have some form of isolation damper to prevent the air from an energized fan from going back through a fan that is not energized. The damper also serves to minimize the shock

Vane Axial Fans in Parallel



Double Width Centrifugal Fans in Parallel



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during start-up of bringing a windmilling fan to a stop and then up to speed again. This is not good for the fan, motor or system. A mechanical back stop clutch can also be used to eliminate windmilling of fans installed in parallel.

The type of isolation damper used will vary with the type of fan. Backdraft or opposed-blade control dampers are used at the discharge on double width centrifugal fans. Butterfly dampers are commonly used at the discharge of tubular inline fans (axial, centrifugal, and mixed flow). Isolation dampers for plenum fans should be located farther away from the fan, either up stream or down, in order to minimize the loss through the dampers.

2. **Continuous operation with one fan** - All equipment will need periodic maintenance and repair. This means that at least one fan is shut down while the others are running. For fans in parallel equipped with isolation dampers, this is generally no problem. The motors of the fans left running must be sized properly by taking into account the shape and slope of the horsepower curve further out to the right. With fewer fans running, the system line will intersect the fan curve further out to the right than with all the fans running. As an example, with a vane axial fan the power may drop off but with a forward curved fan the power will increase due to the constantly rising horsepower characteristic.

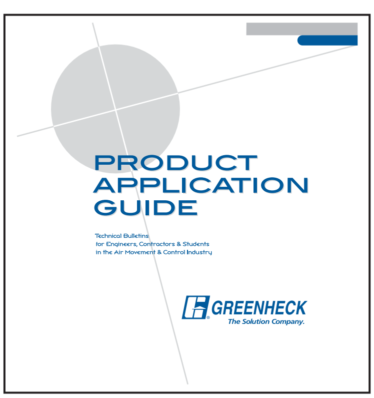
For close-coupled fans in series, it is not advisable to run one fan with the other off. Obviously, an isolation damper will stop the air entirely. Without an isolation damper, the fan shut down will windmill, but the pressure drop across it when added to the reduction in pressure having only one fan running will likely make the system useless.

Summary

When more than one fan is used in a system, whether in series or parallel, the following questions must be answered.

- What is the relative location of the rating point on each fan curve under all operating conditions? Does the operating point remain on a stable portion of the curve?
- Does the location of the rating point allow the use of the fan control desired and the range of flow desired?
- Taking into account the range of operation and numbers of fans, do the motors satisfy the power required?
- Are there properly designed isolation dampers or other provisions in case some fans are shut down? Are leakage requirements satisfied?
- Are the system requirements satisfied under all operating conditions?
- Are sound levels acceptable under all operating conditions?

Satisfactory applications require satisfactory answers to all of the above.



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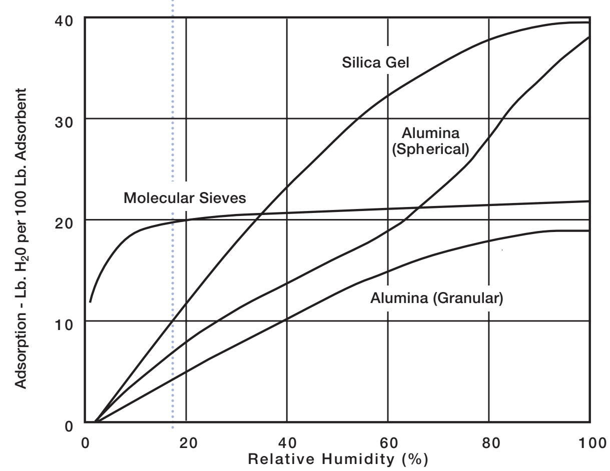
Silica Gel Desiccant

Silica gel is a highly porous solid adsorbent material that structurally resembles a rigid sponge. It has a very large internal surface composed of myriad microscopic cavities and a vast system of capillary channels that provide pathways connecting the internal microscopic cavities to the outside surface of the "sponge".

The characteristic curve for adsorption of water on silica gel is shown in Figure 1 as % weight adsorbed versus relative humidity of the airstream in contact with the silica gel. The amount of water adsorbed rises almost linearly with increasing relative humidity until RH reaches about 60%. It then plateaus out at about 40% adsorbed as relative humidity approaches 100%. The curve for molecular sieves, by contrast, rises rapidly to plateau at about 20% adsorbed at 20% relative humidity. This helps to explain why the molecular sieve is an excellent choice for regenerated applications such as desiccant cooling and dehumidification systems which are designed to reduce processed airstreams to very low relative humidities. On the other hand, silica gel has superior characteristics for the recovery of space conditioning energy from exhaust air.

The use of silica gel on rotary regenerators for energy recovery ventilation applications involves a process cycle where the silica gel is alternately exposed to airstreams having nearly equal relative humidities somewhere in the mid range of this curve (typically between 40 and 60%). When the airstream with the higher relative humidity passes over the silica gel coated wheel, moisture is adsorbed from the airstream into the silica gel. Then when the airstream with the lower relative humidity contacts the silica gel, moisture is desorbed (removed) from the silica gel and put into the airstream.



Source: Davison Chemical Co. *Figure 1*

In this ventilation energy recovery application, the silica gel has all of its surface area covered with at least a monomolecular layer of water because it has a greater affinity for water than any other chemical species.With all of the adsorption sites occupied by water, the silica gel will not be able to transfer other chemical species by adsorption and desorption in its normal form. Species that are soluble in water could become dissolved in the adsorbed water and then released when the water is desorbed but this process is limited by kinetics and does not present a very efficient mechanism for contaminant transfer.

An example of this phenomenon is formaldehyde, a gas which is very highly soluble in water. In the early 1980s when energy recovery ventilators were being used to mitigate excessive formaldehyde levels in mobile homes, concern was expressed by some people that enthalpy type heat exchangers that transferred moisture as well as heat might also transfer excess amounts of formaldehyde gas due to its high solubility in water. Accordingly, tests were conducted by the Lawrence Berkeley Laboratories of the U.S.D.O.E., on two enthalpy type exchangers to determine whether this suspicion was justified. Results were presented in ASHRAE paper No. CH85-03 No. 3 which reported that the rotary type enthalpy heat exchanger (using lithium chloride as desiccant) transferred formaldehyde with only 3-6% efficiency. They concluded that "formaldehyde transfer between airstreams by processes other than air leakage does not seriously compromise the performance of these enthalpy exchangers".

Metric conversion Factors for Fan Application per AMCA Standard

Quantity	"English" Unit	Metric Unit	Conversion Factor
Volume	cubic feet per minute CFM	cubic meter per second m^3/S	0.00047195
		cubic meter per minute m^3/min	0.028317
		cubic meter per hour m^3/hr	1.6990
		liter per second l/s	0.47195
Pressure	inches of water gauge in. wg	pascal Pa or N/m2	248.36
		millimeters mm	25.4
Power	horsepower hp, BHP	watt W or J/s	745.70
		Kilowatt kw	0.7457
Torque	pound-inch lb-in.	newton meter N-m	0.11298
Density	pounds per cubic feet lb/ft^3	kilogram per cubic meter kg/m^3	16.018
Speed	revolutions per minute RPM	revolution per second rps	0.01667
Velocity, Tip Speed	feet per minute fpm	meter per second m/s	0.00508
Dimensions	inch in.	millimeter mm	25.4

Multiply "English" Unit by conversion factor to obtain Metric Unit.
For example: 5000 CFM x 0.00047195 = 2.360 m^3/s
Divide Metric Unit by conversion factor to obtain "English" Unit.
For example: 62.1 Pa / 248.36 = 0.250 in. wg

Metric conversion Factors for Fan Application per AMCA Standard, continued from page 6

Quantity	"English" Unit	Metric Unit	Conversion Factor
Area	square foot	square meter m^2	0.0929
	square inch	square millimeter mm^2	645.16
Length	foot	millimeter mm	304.8
	foot	meter m	0.3048
	yard	meter m	0.9144
Mass (weight)	ounce	gram g	28.350
	pound	kilogram kg	0.4536
Velocity	feet per second	meters per second m/s	0.3048
	feet per minute	meters per second m/s	0.00508
	miles per hour	meters per second m/s	0.44704
Volume (capacity)	cubic inch	cubic centimeter cm^3	16.3871
	cubic yard	cubic meter m^3	0.7646
	gallon (U.S.)	liter l	3.785
	gallon (imperial)	liter l	4.546

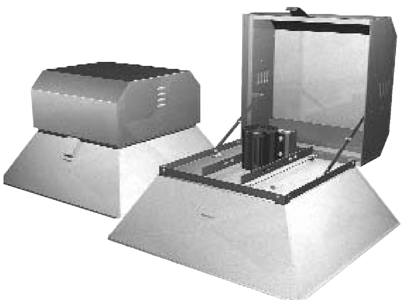
Multiply "English" Unit by conversion factor to obtain Metric Unit.

For example: 5000 CFM x 0.00047195 = 2.360 m^3/s

Divide Metric Unit by conversion factor to obtain "English" Unit.

For example: 62.1 Pa / 248.36 = 0.250 in. wg

What's new at Greenheck



Tamper Resistant Centrifugal Roof Exhaust Fans From Greenheck

Greenheck's Model NYD and NYB tamper-resistant centrifugal roof exhaust fans prevent entry into the fan unit or building. Constructed of galvanized steel, the fan's hinged hood provides easy access to all internal components. Belt drive Model NYB and direct drive Model NYD feature a backward inclined wheel and are available in 10 different sizes, with capacities from 169 to 8,080 cfm and up to 1.25 in. wg of static pressure. Greenheck Models NYD and NYB are licensed to bear the AMCA Seal for Air Performance.

Greenheck Offers Indirect Gas-Fired Make-Up Air Units

Greenheck's new line of Model IG Indirect Gas-Fired Make-Up Air units feature a single-piece design for maximum weather resistance. Model IG features a power vented 80% efficient gas fired furnace and is ETL labeled to ANSI Standards Z83.3 and CGA 2.6. The IG units are supplied with double width, double inlet Greenheck forward curved blowers tested to AMCA standards for high efficiency and low sound levels. Airflow volumes range from 800 to 7,000 cfm with heating capacities up to 400,000 BTU/hr input.

Greenheck's Model IG features large access panels for ease of inspection and maintenance, and is factory assembled and wired to minimize field installation costs. Units are available in horizontal and down discharge arrangements with optional evaporative cooling section and mixing box for recirculation.

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