

## FAN SELECTION AND SIZING TO REDUCE INEFFICIENCY AND LOW FREQUENCY NOISE GENERATION

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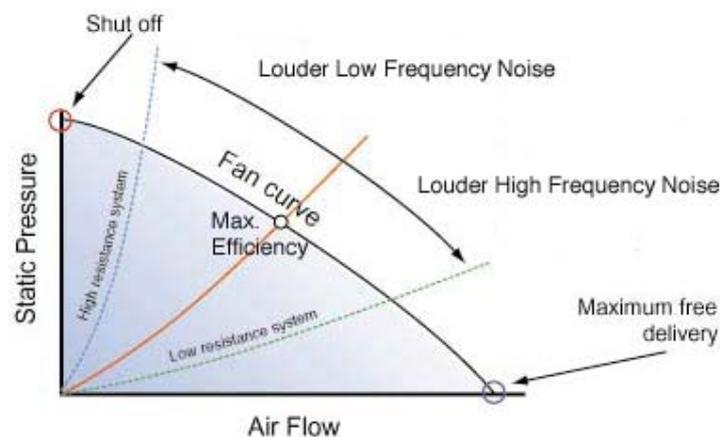
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### SUMMARY

Fans are oversized for many reasons, including overly conservative demand estimates and designing for future demand. Fans that operate at inefficient lower volumes are subject to rotating stall or surge, which wastes power and generates excessive low frequency noise. A method of fan selection is recommended to size fans for maximum efficiency at a point on the system curve below peak demand.

### INTRODUCTION

Energy efficiency and noise generation are functions of fan operation. Fan selection and sizing that recognizes fan efficiency parameters and system effects can reduce operating costs and prevent generation of excessive low-frequency noise. Operating to right or left of the fan curve maximum efficiency point causes increased noise and energy consumption (see Fig. 1), which may require otherwise unnecessary attenuation measures. In addition to the increased energy costs, additional attenuation materials and construction efforts add to installation costs.



*Figure 1: Generic Fan Curve<sup>1</sup>*

Fan selection and sizing should consider all aspects of fan efficiency, including external system effects such as inlet and discharge conditions. Fan size should be “optimized” to the anticipated “normal” demand, or most often required air quantity demand. In lieu of selecting a fan for maximum efficiency at peak demand, the designer should permit the fan to operate to the right of maximum for short periods of peak demand. In a variable volume system, the operating range on the system curve should be above the maximum efficiency point for short durations of peak demand, below the maximum efficiency point for low demand, and at, or very near the point of maximum efficiency for the most common or average demand. This selection procedure will result in lower average energy costs, and generally quieter operation.

## FAN TYPES

There are many generic fan types with varying air volume and pressure capacities. Selection should be made based on the application requirements. Mechanical energy drives the impeller via shaft connection. The impeller changes the mechanical energy to air velocity and pressure. Various types of fans propelling air centrifugally, axially, or in a combination of modes produce different operating and noise characteristics.<sup>2</sup> The most common types are:

### Centrifugal Fans

Forward curved (FC), airfoil (AF), backward-inclined (BI) and radial fans are centrifugal. Air entering a circular aperture is moved radially, or out tangentially from the center (see Fig. 2). Forward curved blade impellers operate at lower speed than other centrifugals and produce higher velocity than the blade tip speed with somewhat turbulent airflow. The combination of lower rotational speed and greater turbulence results in a typical low frequency dominant spectrum. Airfoil and backward-inclined blade impellers produce air velocity slower than blade tip speed, but require greater rotational speed than FC types. At the cost of more maintenance and some increase in mechanical noise, the AF and BI fan types generate lower sound levels overall. The “plenum (plug) fan” is a single inlet AF or BI fan impeller in a plenum enclosure instead of a scroll. Radial fans, used for high-pressure applications, produce high sound levels with pronounced blade passage tones.

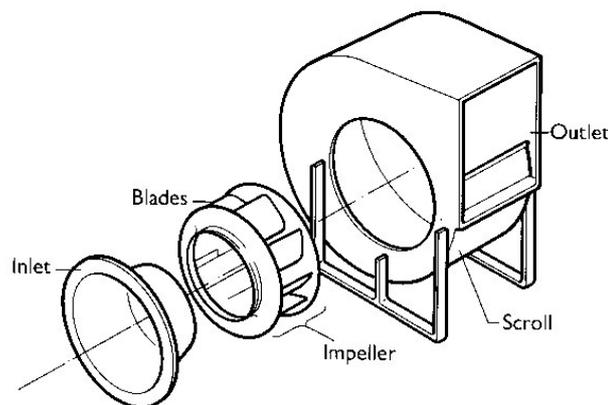


Figure 2: Centrifugal Fan with Scroll Enclosure<sup>3</sup>

## Axial Fans

Air flows along the impeller axis of tube-axial and vane-axial fans (see Fig. 3). The mechanical energy of the driver is converted into spiral airflow at the fan discharge. The tube-axial has a fan impeller in a cylindrical housing. The vane-axial is a variation with guide vanes incorporated to straighten airflow and increase efficiency. High tip speeds are required to generate airflows comparable to centrifugals. Typically the motor, with a direct-drive shaft arrangement, is in the air stream, and can contribute to turbulence if other conditions, such as duct geometry, are not ideal. Although the sound level generated is similar to centrifugals, it is a more balanced spectrum, or proportionately more high frequency and less low frequency noise. However, stronger tones at blade-passage frequency can occur. This can be particularly true when ducted inlet or discharge configurations add turbulence to the system.

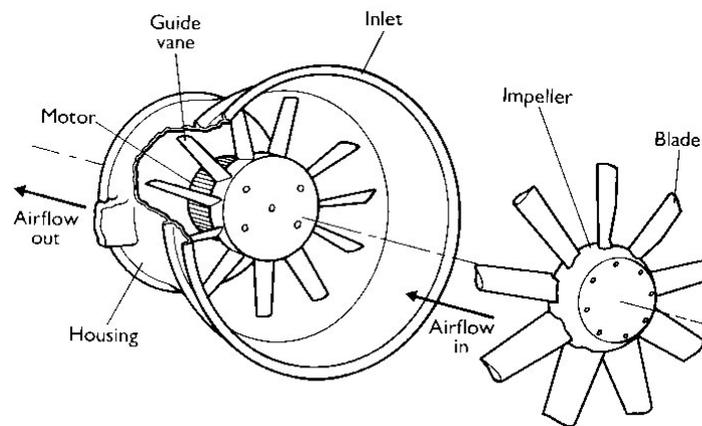


Figure 3: Vane-Axial Fan<sup>4</sup> (note motor in air stream)

## Mixed Flow Fans

A mixed flow fan is an axial flow impeller modified to generate a radial component of flow, which is added to the spiral flow, achieving a capacity for high volume and high pressure. Along with operational characteristics combining the benefits of centrifugal and axial fans, the sound levels are slightly more moderate than other axial fans with a spectrum between the typical strong low frequency centrifugal and the more balanced spectrum axial.

## Fan Curves

There are operating characteristics typical to the various generic fan types, which make them more or less suitable to variable volume operation (having a larger or smaller range on the fan curve with relative operational efficiency).<sup>5</sup> The shape of each curve gives some indication of the sensitivity to system effects, such as inlet and discharge conditions, and potential noise (see Fig. 4). Each of the fan curve illustrations in Fig. X below, shows (a) system resistance intersecting the fan curve at “maximum efficiency” point (quiet operation), (b) system resistance intersecting the fan curve in region of “rotating stall” (low frequency rumble), and (c) system resistance in the “surge” region (low-frequency fluctuating noise). On each of these curves, moving farther out on the fan curve (to the right of maximum efficiency) also increases noise level, but predominantly in mid- and high-frequencies and at blade passage

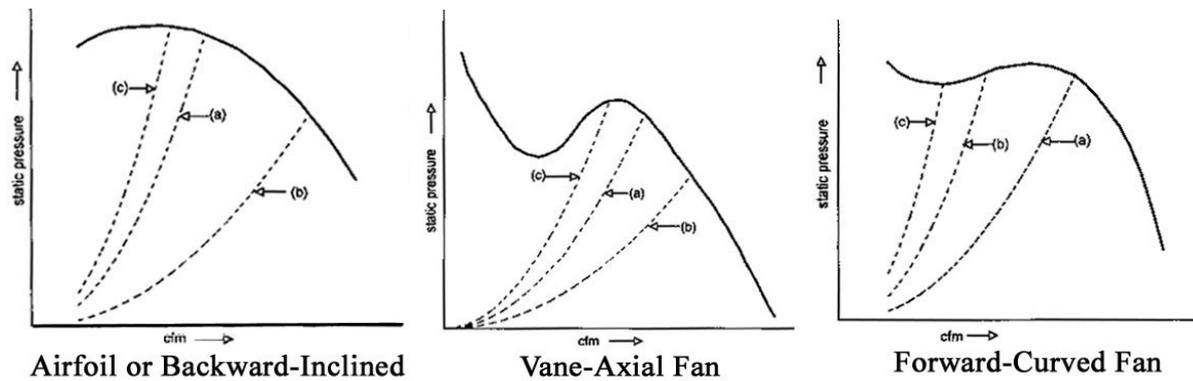


Figure 4: Fan curves showing various system resistance conditions

frequency (equation 1), which varies with rotational rate according to the following:

$$\text{b.p.f.} = \left( \frac{n}{60} \right) \times N \quad (1)$$

where:

b.p.f. = blade passage frequency (Hz);

$n$  = fan wheel revolutions per minute (rpm);

$N$  = number of impeller blades

Note that the blade passage frequency can be perceived as a tone. If it coincidentally occurs within a frequency span of the fan noise spectrum that is prominent, it can emphasize the tonality of an otherwise relatively smooth fan noise spectrum. Harmonics and sub-harmonics are possible at multiples of the blade passage frequency, but at lower amplitude (sound level).

## CONSTANT VOLUME AND VARIABLE VOLUME SYSTEMS

Constant volume systems allow the designer to select the fan for a specific point of operation. If the estimates of actual required air volume and system static pressure are accurate, the fan can be selected to operate at or very near the point of maximum efficiency until and unless some element of the system is changed. Variable volume systems must accommodate a range of demands in addition to varying system conditions. The design volume can greatly affect whether the fan operates efficiently and quietly over its entire range. The method of volume control also affects the efficiency and noise generation. Inlet vanes and modulator discs degrade inlet conditions. Scroll (discharge) dampers increase static pressure. Variable frequency drives, the preferred volume modulation method,<sup>6</sup> reduce the fan rotational speed, allowing air volume and static pressure to reduce. Ideally the fan moves up and down the system curve in response to varying demands. Of course, the harsh reality of varying system effects may still push the fan operating point somewhat to the right or left.

The greatest energy and noise problem with system effects and variable volume systems is the possibility of rotating stall or surge, as mentioned in the fans discussion above, which occurs with disturbed conditions at inlet or excessive air flow reduction. The key to achieving

efficient and quiet fan operation is to select a fan whose operating range does not approach the stall or surge region, even at the minimum volume demand. The chart below (Fig. 5), illustrates two different fan curves for the same demand (system curve).<sup>7</sup> The larger fan is selected for maximum efficiency at maximum demand, and the smaller fan is somewhat “undersized.” At peak demand, the smaller fan moves to the right of maximum efficiency on the fan curve. At the lower end of the demand range, the larger fan approaches surge, wastes energy and generates excess low frequency noise. Attenuation of low frequency noise is difficult and costly. The smaller fan may generate some excess mid- to high-frequency noise at peak demand (which is relatively easy to attenuate), while at the lower end of the demand range, it is still running fairly efficiently, some distance from potential rotating stall or surge and the attendant low frequency noise.

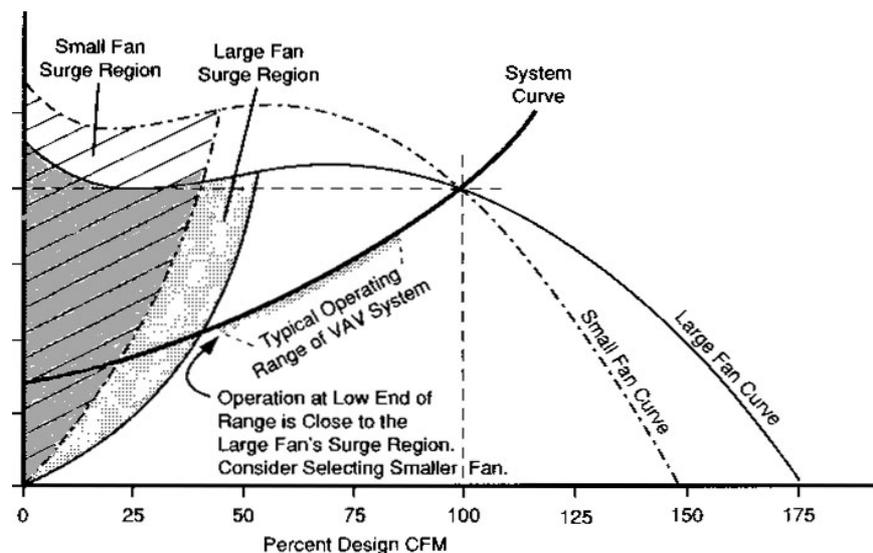


Figure 5: Comparison of Two Fan Curves versus Single Demand Range<sup>8</sup>

## CONSERVATIVE DESIGN IS NOT ALWAYS THE BEST DESIGN

Why are fans oversized? Conservative estimating often results in determination of the worst case—the greatest possible demand or perhaps unrealistically high static pressure. While the fan may be sized to operate at maximum efficiency under the heaviest demand, it actually operates at lower efficiency at the more moderate realized demand. This condition is aggravated in a variable system, as the demand reduces to a small percentage of maximum. Furthermore, engineers designing a variable system can easily underestimate diversity—the relatively smaller demand range when some parts of the system reduce demand as other parts increase. For example: demand changes with variability at terminal units, or in laboratories, exhaust systems vary with fume hood sashes either fully open, partly open, or closed.

Making design allowances for future demand often results in fans oversized for current or actual demand. In the effort to economize, an engineer may install a fan that will accommodate demand growth that may or may not materialize. Over time, the cost of increased power usage to operate an oversized, inefficient fan may become significant. This expense could be more than it would cost to replace a smaller, more efficient fan with a larger one if the demand requirement increases.

Engineers designing a system can also easily underestimate system effects. While this should not cause them to select oversized fans, it could exacerbate some of the noise problems

associated with oversized fans. System effects, especially near fan inlet and discharge, cause the fan to operate under different conditions than the ideal conditions under which the fan is rated. Failure to properly account for turbulence, added static pressure and other inefficiencies can produce a different system resistance curve than the engineer is expecting. If the fan is oversized, system effects may encourage the fan to approach stall or surge, waste energy, and generate excess low frequency noise. With a smaller fan, system effects might cause the fan to waste energy and generate some excess noise, but again this would be mid- to high-frequency noise, which would be relatively easier to attenuate.

## PROPOSED FAN SIZING PROCEDURE

Fans should be selected and sized to operate at maximum efficiency at the design volume. This is hardly a novel idea. It is included in ASHRAE publications and numerous books, including the ones that are referenced in this paper. However, applications engineers often oversize fans for a variety of reasons, some of which are noted above. As a result, building owners pay for increased construction costs for new buildings and facility modifications, and they continue to pay for excessive energy usage. In addition to the economic costs, building occupants are confronted with the annoying and unpleasant low-frequency rumble of fans operating in stall or surge conditions. Therefore, we propose:

- Estimate fan demand based on accurate predictions of “normal” conditions, rather than “worst case” conditions for air volume requirements, static pressure, building use, climate changes, etc. Select the fan to operate at point of maximum efficiency under “normal” conditions. This will cause the fan to move to the right on the fan curve when demand increases or peaks, but only for relatively short durations. The high demand excess noise will be in easier to attenuate mid-to -high frequencies.
- Select and size fans for current or anticipated near-term demand. The high efficiency of operation will save energy costs. Those savings can be allocated toward purchase of larger fan wheels, to be installed when demand growth actually occurs. Note that air-handling units are typically manufactured with a few modular cabinet sizes that each accommodate a range of fan sizes. If future demand is anticipated, specify the larger air handler cabinet needed for the future fan, but with the smaller fan wheel installed for now.
- Design inlet and discharge ducts that permit laminar flow to and from the fan to minimize system effects.
- Select variable volume system components that have greatest efficiency with least undesirable system effects. Match the modulating elements to the fan type. Forward curved fans may be more appropriate for smaller loads, but more susceptible to inlet disturbance from vanes. Airfoil or backward inclined fans may have a flat, broad fan curve that is more forgiving of non-ideal inlet conditions, but are more suited to larger volume demands. Variable speed controllers are generally superior to inlet vanes, modulator discs, or discharge dampers.
- Optimize fans in variable volume systems to have fan curve maximum efficiency point at 75% - 85% of peak demand, or a similar proportional amount, so that minimum demand occurs in a relatively stable and efficient region of the fan curve.

With careful consideration of potential operating conditions during design, including fan efficiencies and system effects, more economical operational costs and more moderate low-frequency fan noise levels can be achieved.

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<sup>3</sup> *Ibid.*

<sup>4</sup> *Ibid.*

<sup>5</sup> C. Ebbing, W. Blazier, editors – *Application of Manufacturers’ Sound Data*. American Society of Heating Refrigerating and Air Conditioning Engineers: Atlanta, **1998**, pp. 41--53.

<sup>6</sup> *Ibid.*

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<sup>8</sup> *Ibid.*