# FAN ENGINEERING

Information and Recommendations for the Engineer FE-600

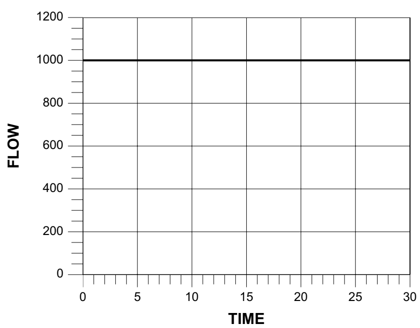
# Surge, Stall, and Instabilities in Fans

## Introduction

Users of fan systems desire a steady, continuous flow of air. In this ideal situation, the pressure generated by the fan is constant. A single instantaneous measurement of the flow rate would be valid for extended periods of time.

Figure 1 shows the flow for an ideal system. Figures 2 through 5 show a variety of conditions for time variation of flow.

*Figure 1. Ideal Flow*




Those involved with measurement of flow rates know that ideal flow conditions are not common. Each point of flow measurement is usually time averaged for ten seconds or more to get an accurate reading. Variations in flow and pressure readings of 10% over short time periods are relatively common. However, fans which are improperly selected or applied can produce variations far greater than this. Conditions can become so severe that the flow through the fan can oscillate between forward and reverse (flow exiting the inlet) many times per minute (see Figure 4).

The variations in flow and pressure not only make it more difficult to measure the flow, they can create a variety of problems:

1. Dramatic increases in noise.
2. Increases in vibration.
3. Structural fatigue damage to the fan due to continuous loading and unloading of components.
4. Damage to the ductwork and other system components.
5. A fan system that does not perform properly due to unsteady flow and/or transmitted vibration.

An understanding of the causes of unsteady flow can be helpful in avoiding these problems. Because some of the causes are very complex, researchers have had some interest. However, there has not been uniform agreement in the conclusions as to what the exact causes are. From their research we can learn the conditions that tend to perform normally and avoid the conditions that don't.

Systems with unsteady flow can perform mysteriously. Complicated terms to describe the phenomena are often used and misused. The result is that there are

*Figure 2. Typical Flow*

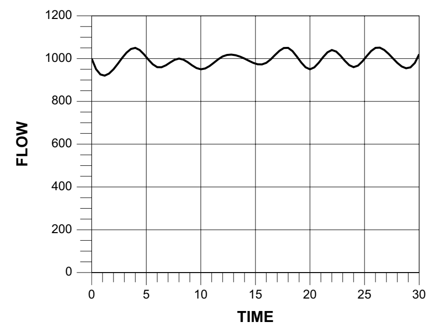



*Figure 3. Stall Condition*

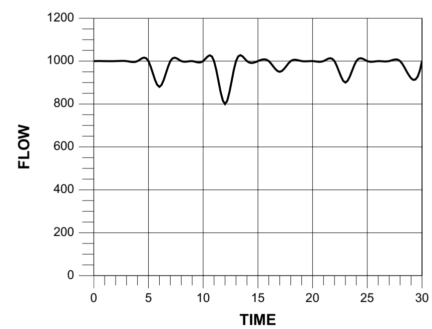



*Figure 4. Surge Condition*

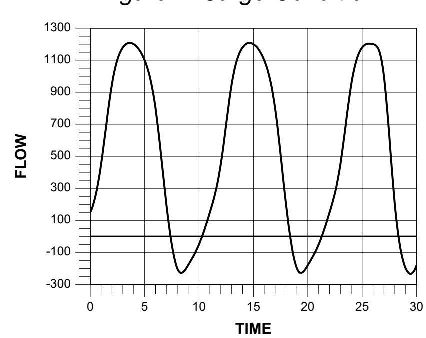
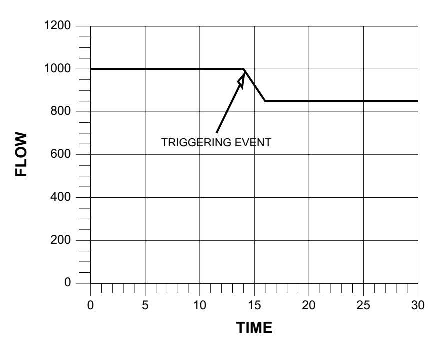



*Figure 5. Bi-Stable Flow Condition*

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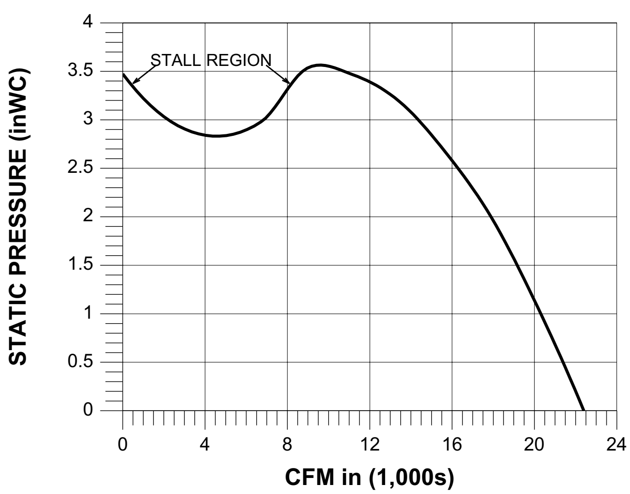
few authoritative reference materials to use as diagnostic guides. We will discuss some of the more common terms and issues.

## Stall

Elongated objects (such as fan blades) when passing through the airstream will deflect the air. If we change the orientation of the object relative to the flow direction, we can increase or decrease the amount that the air is deflected. If we progressively increase the attack angle of the fan blade, it will increase the amount of deflection. It is this change of direction (and relative velocities) that allows the fan to generate pressure. If the attack angle becomes too severe, the air will no longer follow the blade surface in a uniform manner. The amount of deflection and the pressure being generated stops increasing and normally will fall off. This is called the stall point. Figure 6 shows a fan curve of a fan with a significant dip in the stall region.

In a fan, the blades are normally rotating at constant velocity. Therefore, to change the angle of attack, the system to which the fan is attached must be changed. Higher flow rates through the inlet increase the attack angle, and lower flow rates decrease it. Therefore, if a fan is operating in stall, it is because the CFM is too low for it. On a given system, this is caused by selecting a fan which is too large (making the air velocities too low in the fan).

*Figure 6. Curve Showing a Dip in the Stall Region*




In some fans, the angle of attack is not uniform across the full width of the blade. These are normally not the most efficient fans, although the severity of the stall is often less since only part of the blade is stalling at any one flow rate. Some people say that radial bladed centrifugal fans are always in stall since there is a poor match between the directional velocity of the blade and that of the approaching air. This is essentially true. However, these types of fans can have severe varying flows at very low flow rates since the internal losses are dominated by stall and the pressure falls off at this point.

A fan operating at or near the stall point usually will have severe increases in noise. On some fans it will sound almost as if the impeller is being impacted by a solid object (hammering). Pure stall tends to have a random frequency but there are special cases where a pure frequency is generated. This will be discussed later.

There is a time-varying nature for the flow of a fan in stall. However, this is normally not the major cause of concern. The increased noise being generated can be a problem, but this too can often be dealt with. The major concern for a fan operating in stall is the potential for mechanical damage. Those who have had a "bumpy" airplane ride have a feel for how severe aerodynamic shock impacts can be.

A fan continuously operating in stall can sustain structural metal fatigue. This is especially true for axial flow fans having long slender blades, or blades fabricated from sheet metal. Centrifugal fans are less prone to damage. Centrifugal fans designed for relatively high pressures but operating at very low pressures (less than 1" SP) have been known to operate continuously in stall for many years without damage.

There is another downside to having a fan operate in stall. It means the efficiency of the fan is less than optimum. A smaller size fan costs less and has a lower operating cost. It will also likely outlast a larger fan.

## Rotating Stall

This is a special case of stall that normally only occurs in backwardly inclined and airfoil centrifugal fans. Most observers also report that inlet box dampers are involved. Variable inlet vanes do a good job of preventing rotating stall because they provide a more stable flow path for the air through the wheel. These fans are encased in a scroll type housing that helps generate the fan's pressure. The pressure around the periphery of the fan wheel varies relative to how near it is to the fan outlet (where it is highest). These fans have several blades, typically 9 to12.

We will call the passageway between each blade a "cell." The flow through each cell can vary since the pressure around the periphery varies. Near the stall point it becomes possible for most of the cells to have the normal forward flow, while one or two cells have reverse flow. The air that "squirts" backward through these cells has nowhere to go so it moves to an adjacent cell, deflecting the air which was already traveling through it. This change of attack angle now forces this cell to stall, it then also has reverse flow, passes on its bubble of air, and on and on around the fan wheel.

Rotating stall typically occurs in fans which are severely throttled (inlet box damper typically less than 30% open).

Most researchers have reported that the frequency of travel of this rotating stall occurs at about two-thirds of the fan rotational RPM(x). Some have observed two traveling cells at once generating a four-thirds rotational frequency. There are other reports of rotating stall ranging from two-thirds and even higher harmonics (2/3x, 4/3x, 6/3x, 8/3x, ...). If these exciting frequencies coincide with the natural frequencies of the wheel or housing, resonance occurs and damage can result. This frequency will show up in both sound and vibration measurements. Rotating stall is among the most destructive of instabilities in the fan.

## Surge

Several years ago there was a report of a grain drying system that was pressurizing large grain bins. It utilized a burner section near the inlet of a large centrifugal fan. Periodically this system would "belch" fire back out the inlet of the burner. This was likely a severe case of system surge.

The sound a surging fan makes is commonly described by observers as "whoosh" or "whoomp." There are several criteria which must be met to have surge:

1. There must be a relatively large volume of air which is pressurized (such as the grain bins or a large plenum).
2. There must be a section of ductwork with relatively high velocities.
3. The operating point of the system is to the left of the peak pressure (at lower flow rates). In this region the fan curve has a positive slope such that increasing the flow also increases the fan static pressure.

In concept, a system in surge is like an oscillator. The energy imparted to the air alternates between creating kinetic energy (high velocity in the duct) and potential energy (compressing the air in the plenum). The positive slope on the fan curve allows large amplification of this oscillation to occur. In extreme conditions, the air can temporarily blow back through the inlet.

In a fixed system, the frequency of the surge is constant. Usually the frequency is low enough that you can count the number of cycles per minute since it is quite audible. Most severe reports occur at a frequency below 300 cpm. One researcher reported that this effect seems to disappear at frequencies above 450 cpm.

The frequency of surge can be be calculated for simple systems:

Frequency (Hz) = 175 * Square Root [Duct-area / (Plenum-volume * Duct-length)]

Note: Keep all dimensions in feet.

For those wishing further information, research the term “helmholtz resonator.”

## Hunting

Some people use the term “hunting” to apply to any time-varying flow. However, the appropriate use of this term should apply to an under damped control circuit. In variable volume systems, sensors are used to provide information that controls dampers, vanes, speed controls, or other means of setting the flow rate. If the control system responds too quickly, it will overcorrect and have to readjust the other direction. In the extreme condition, a system may continuously “hunt” back and forth.

## Stability and Bi-Stable Flow

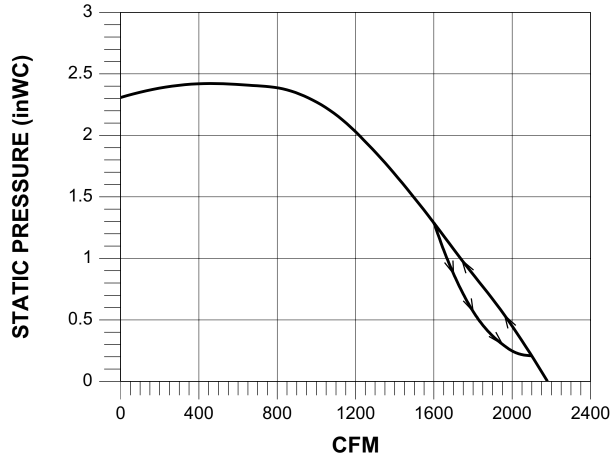
Stability refers to the ability of a system if temporarily displaced to return to its un-displaced position. A coin balanced on edge is unstable. A coin at rest on a flat surface is stable. Some fans are not stable for all flow ranges. For example, walking by the inlet (*don’t try this!*) of a large centrifugal during an air test caused the flow to reduce by over 15%. This fan continued to operate at the lower flow rate until the test was restarted.

We can determine the stability of a fan by performing two air tests. On one test, we start at full flow (free delivery) and measure the flow and pressure as we progressively add more resistance. In the second test, we start at shut-off and progressively reduce the resistance. We now have two flow vs. pressure fan curves. If they do not overlay, we have a region of instability. Figure 7 shows a sample fan curve with this property. Since there are only two possible conditions on any system, this is called bi-stable flow.

Although the noise changes between the two flow conditions, neither is particularly objectionable. If the fan is rated in the high flow condition and it “trips” to the lower condition, the loss of flow can be a problem.

Bi-stable flow has been observed in backwardly inclined centrifugal fans, usually at performances close to free delivery and almost always at flow rates higher than that where the best efficiency occurs.

*Figure 7. Bi-Stable Flow*



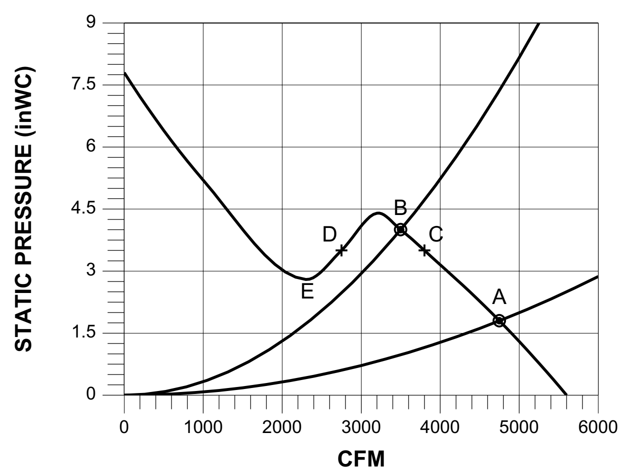

## Parallel Flow Operation

It is relatively common for two or more fans to be operating in parallel. In the two fan system, each fan is selected for half of the design flow rate. Fans that have a large “dip” in the stall region can have another type of problem. Vaneaxial and forward curve centrifugal fans are two types of fans which can have large “dips.”

The problem with parallel flow systems can occur in the starting sequence. If the fans are properly sized, started simultaneously and brought up to speed at the same rate, there is no problem. However, if one fan is started first, the second fan is already exposed to back pressure while it is coming up to speed. At full speed, a condition can arise where one fan is operating at a flow rate to the right of the peak static pressure point, while the other fan is trapped on the left side of the peak.

Skeptics will not think this is possible, so here is an example. A fan was to be selected for 7000 CFM at 4" SP. Due to severe space limitations, two 18" diameter vaneaxials were selected to operate in parallel. Referring to Figure 8 (B), we can see that each fan was to deliver 3500 CFM.

*Figure 8. Two Identical Fans Not Sharing the Load Equally*



Those familiar with the system will recognize the system curve as a parabola with the equation:

$$SP = CFM^2 \div \text{Constant}$$

For this system we get:

$$4 = 7000^2 \div \text{Constant}$$

or

$$\text{Constant} = 49{,}000{,}000 \div 4 = 12{,}250{,}000$$

We can determine what this system will do with one fan running by plotting this parabola on our fan curve. With fan #1 running we can see that we will get about 4750 CFM at 1.8" SP (A).

At this point, flow would be going backwards through fan #2 unless we stop the leak. Commonly, heavy duty backdraft dampers are used for this. When starting fan #2, it will contribute no flow until it reaches a speed such that the pressure (at shut-off) exceeds the back pressure it sees from fan #1. As the speed continues to increase, the flow will eventually reach point "D". Meanwhile, the pressure fan #1 sees continues to increase until it reaches point "C".

We then have the final condition:

Fan #1 is delivering 3800 CFM @ 3.50" SP
Fan #2 is delivering 2750 CFM @ 3.50" SP

From the system curve equation:

$$(2750 + 3800)^2 \div 12{,}250{,}000 = 3.50"$$

- The system is happy because the flow (total of both fans) and pressure readings are on the system curve.
- Fan #1 is happy because point "C" is on the fan curve.
- Fan #2 has mixed emotions. It is happy that point "C" is on the fan curve, but unhappy about being trapped in stall.

If these were your fans you would probably be unhappy because:

1. The total CFM is only 6550, not 7000 as expected.
2. Fan #2 is probably noisy.
3. Fan #2 is prone to damage due to operation in stall.

This example shows that it is possible to have two identical fans not sharing the load equally. A more severe condition can exist if non-identical fans are operating in parallel. Some years ago a complaint regarding a system with two fans in parallel was received from a customer. After installing a second larger fan in parallel with a smaller fan that had been in operation, the combined flow wasn't what was expected. Measurements revealed that the second fan by itself was generating more pressure than the first fan was capable of at any point on its fan curve. The original fan was completely overpowered, and flow was blowing back out of its inlet. The customer was advised to shut off the original fan (saving the power) and block solid the duct branch to

*Figure 9. Normal Width Fan* *RPM=1282, BHP=13.14*

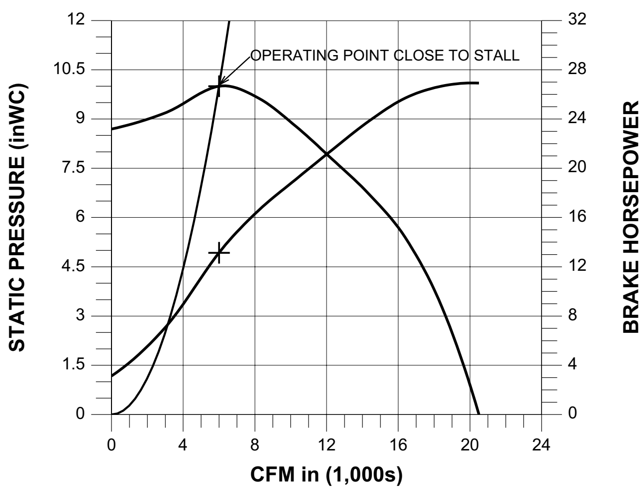



*Figure 10. Reduced Width Fan* *RPM=1342, BHP=12.56*

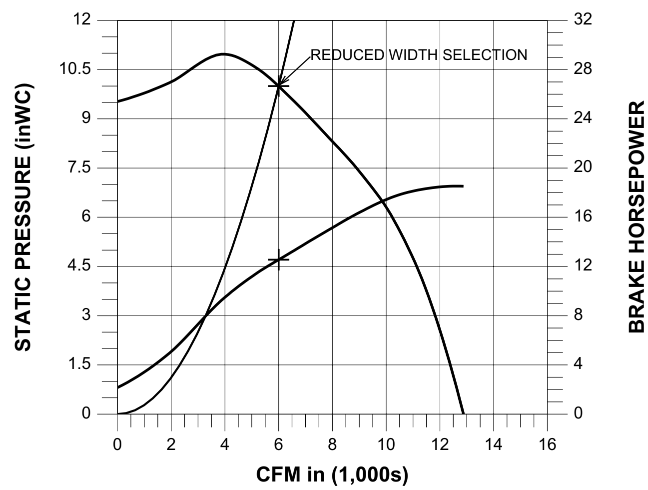



*Figure 11. Blade Angle=45°* *RPM=1388, BHP=15.64*

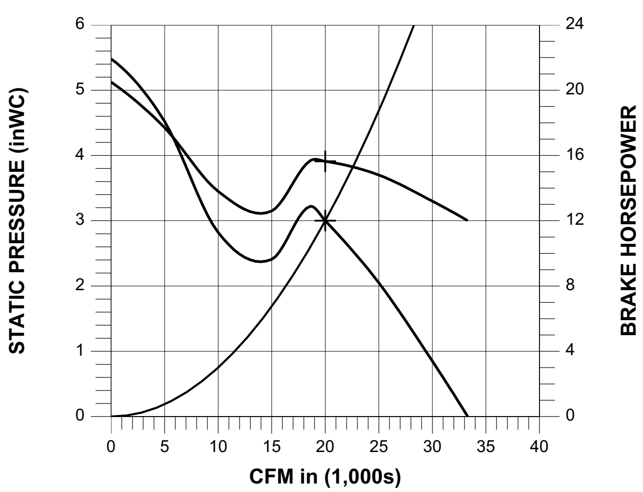
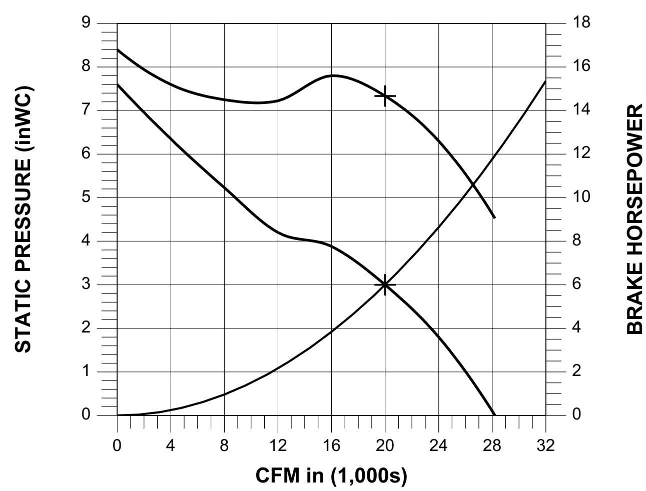



*Figure 12. Blade Angle=35°* *RPM=1663, BHP=14.67*



the original fan (plugging the leak). There were two lessons learned here:

1. Don't mix two different fans (or operate two identical fans at different speeds) for parallel operation.
2. If more flow is required on a constant system, boost the pressure capability of the fan or add a second fan in series.

## Tips On How to Avoid Problems

1. Don't select too large a fan. Some people think that selecting a larger fan will give them an extra safety margin should the system be miscalculated. This can put you in the undesirable part of the fan curve where stall and surge can occur. A fan too large for an application is also uneconomical to operate (waste of horsepower). You are better off selecting a higher class of fan that can be sped up should the system calculations be incorrect.

2. Centrifugal fans can be designed with "narrow width" construction to help avoid stall. Comparing Figure 9 and Figure 10 shows how reducing the fan width not only helps avoid stall, but also reduces power consumption (BHP).

3. On adjustable pitch axial flow fans, reducing the blade angle and increasing the speed can help avoid stall. Comparing Figure 11 and Figure 12, the operating point moves away from stall and the power requirement decreases by changing the blade angle from 45° to 35°.

4. On variable volume systems, the use of inlet vanes can allow lower turndowns and still avoid stall.

5. Severely undersized fans are not only inefficient; they can make the fan operate in a condition where stability can be a problem.

6. Avoid operating fans for long periods in a severely throttled condition.

7. On fans that need to operate at low damper settings, utilize one of the following:
   a. Variable speed drive
   b. Bypass duct from discharge to inlet
   c. Variable inlet vanes (VIV)
   Note: Rotating stall has not been observed with VIV*
   * See attached references.

8. For fans in parallel, there are four suggestions:
   a. Make sure fans start simultaneously.
   b. Select fans so that the operating static pressure is lower than the lowest point of the dip in the curve. (Below the pressure at point E on Figure 8.)
   c. Select identical fans.
   d. Operate fans at identical speeds.

## Quick Fixes for Eliminating Stall

1. Allow greater flow through the fan by creating a "leak". On closed systems, a small duct can often be run from the outlet back to the inlet (with recirculation).
2. On fans that have internal cones, reposition the fan wheel so that air can leak from the periphery of the fan wheel back into the inlet.

**Note:** Both of these fixes will reduce the system efficiency.

## Conclusion

The words "stall" and "surge" often "put the fear" into inexperienced people when applying fans. We all would like to simply plug a fan into a system and have a continuous steady flow. It would be nice if system calculations were ultra precise making it easy to avoid bad operating points. However, in the real world fans are often applied in less than optimum conditions, and many times in conditions where stall is likely. Even then, severe problems are rare. When problems do occur, there are methods to identify the problem type, and once identified, solutions can be implemented.

## Reference

A summary of experiences with Fan Induced Duct Vibrations on Fossil Fueled Boilers; Presented at American Power Conference; Chicago, IL; April 21-23, 1975.

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