

Considering centrifugal fans

Fans have been the workhorse of industry for over 100 years. The importance of fans was traditionally ignored because they were considered a secondary part of the process. They were seen as less important than, for example, the turbine in a coal-fired power station or the kiln in a cement works. However, two important shifts in focus have led to a drive towards more efficient fans and improved cement production.

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The first shift in focus with fan performance was a growing awareness that if the fans were not running then the process had to stop. This led to the situation where it is now common for fans to have at least weekly vibration checks, if not continuous monitoring. There is now a greater move towards using vibration monitoring to improve reliability and manage outages.

The second shift in focus was an appreciation of the amount of energy that fans consume. This can be quite substantial, with cement plants reporting that about one-third of plant energy consumption is due to fans. This has led to a drive towards more efficient fans. In the case of cement fans, this has also led to improved cement production through the supply of more air for the same energy consumption.

The net effect of these changes is that often what is required is a larger, more efficient fan that should fit on an existing foundation and operate with low vibration levels.

The correct blade geometry

Centrifugal fans work by creating a vacuum behind the blade that pulls the air through. This means that the bottom surface of the blades is not too important for efficiency. Fan manufacturers make use of this fact to strengthen blades with nose and tail strengthening strips on the bottom surface.



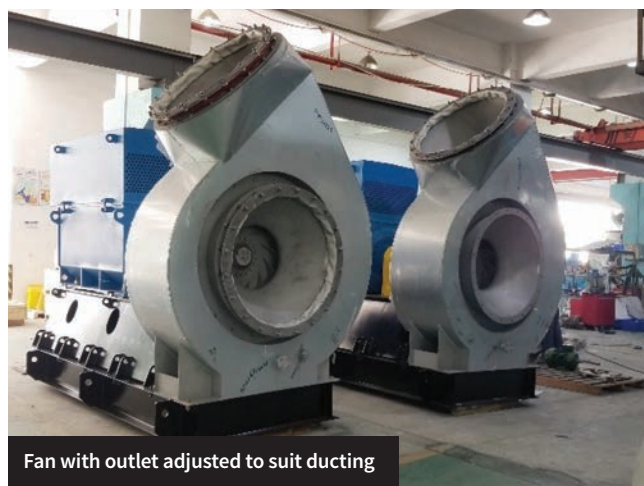
Bolts attaching a wearplate to an impeller

Liners are bolted to blades or welded to blades with cut-outs for wear protection. It also means that an efficient curved blade fan is maybe only a few per cent less efficient than a high efficiency aerofoil bladed fan. The main use of aerofoil blades for large, wide fans is that they provide a box section which is stronger than a plate blade.

If there is a lot of dust, it often collects under a curved blade. In this case, aerofoil blades also have the advantage of having a flat bottom. Another option is a flat inclined blade. However, for inclined blade fans an efficiency of up to 80 per cent can be obtained – ie, with the Halifax Fan BFBI.

The choice of the blade geometry depends on whether the particles will stick to the impeller. If the particles easily stick to the impeller then a flat bottomed blade is preferred, ie, a flat inclined blade or an aerofoil blade with a flat bottom. The underside of curved blades can form a dust trap, especially for wider impellers which have less radial blades. This situation would be made worse with blade tail strips, which create a ledge to catch the dust. Blade nose strips are less of an issue.

In the cement industry the ability of dust to stick to an impeller is dependent on the moisture content of the dust. This means that if the fan is operating above the dew point temperature there are no dust adhesion issues. Therefore, it is common to cement applications that have a high dust content where backward curved impellers with blade nose and tail strips are used successfully.



Fan with outlet adjusted to suit ducting

The other issue with dust is wear. Aerofoil bladed fans have the biggest wear issues. These can be addressed by using a solid nose on the aerofoil with wear protection on the nose and top skin. Flat inclined blade impellers have the simplest blades to apply wear protection to, as no rolling of the wear plate is required.

Defining the fan duty

The static pressure is the pressure that a barometer would read if put in the duct. The dynamic pressure is the pressure that would be measured from stopping the air flowing. The total pressure is a combination of both of these. So static pressure can be measured at 90° to the flow (eg, duct wall tappings) and total pressure is measured with a pipe facing into the flow. The total pressure is duct independent – a smaller duct will give higher dynamic pressure (higher flow), so lower static pressure. This means that total pressure is installation independent.

Fan pressure is defined in three ways, total pressure rise, static pressure rise and static pressure difference:

- total pressure rise = total pressure outlet – total pressure inlet
- static pressure rise = static pressure

outlet – total pressure inlet

- static pressure difference = static pressure outlet – static pressure inlet.

The total pressure rise gives the total energy increase through the fan, so is the value used in codes and standards to define efficiency, eg, AMCA FEG and ISO 12759.

Typically, a process engineer will establish the required static pressure and volume flow at the process. The engineer will then assess the system pressure losses. The pressure losses will be combined with the required static pressure to give the engineer the static pressure at the fan. Using a similar methodology on both sides of the fan gives the required static pressure difference across the fan. Although these calculations are dependent on duct size, static pressure rise and static pressure difference are used by many companies to define fan duties. Using total pressure rise is the more accurate option as it is independent of duct size.

When a fan is a replacement, which is common in the cement industry, the fan inlet/discharge velocity can change because of a change in the fan inlet/discharge areas. For this kind of situation, it is better to select the fan using total pressure rise. This will ensure that the static pressure in the ducting downstream of the new fan will be the same as for the old fan.

Often the original fan has been over-specified, the original fan curve is lost, the original fan curve is not trusted or, for some other reason, the replacement fan must be based on test results. A Halifax Fan test engineer would typically measure temperature, static pressure and total pressure at the inlet, discharge and

possibly other locations.

The static pressure, gas properties and inlet temperature would be used to establish the process gas density at the measurement points. This, with the dynamic pressure, would give the calculated flow. Averaging of the flow at the different measurement points can be used to improve the accuracy of the flow measurements.

Improving the selection

In addition to the pressure and flow, there are other features of the process that can affect the fan selection. These can affect the position on the performance curve, the geometry of the fan and the method of controlling the fan.

Most fans operate with a stable load. This means that they can be run safely at five per cent of the peak pressure. Not all fans operate with a stable load. An example of a fluctuating load is clinker cooling. As the clinker is fed on to the grate cooler, it will not form a uniform bed. There will be significant changes in bed thickness and density. For these less stable processes the fan has to be run further from the peak of the curve, usually at least between 10-15 per cent below peak pressure. Even if the process is stable, the process designer may want a pressure margin safety factor to allow for future expansion or process calculation approximation effects.

The duty a fan runs at depends on the system resistance line and the fan curve.

Manual box damper retrofitted on centrifugal fan – used to match fan VSD performance to system curve



This combination affects the flow control method used.

The most efficient form of flow control is typically variable speed control. However, with variable speed control the pressure and volume are dependent on the speed:

- pressure = const x speed²
- volume flow = const x speed

This gives the same relationship as the square law relationship:

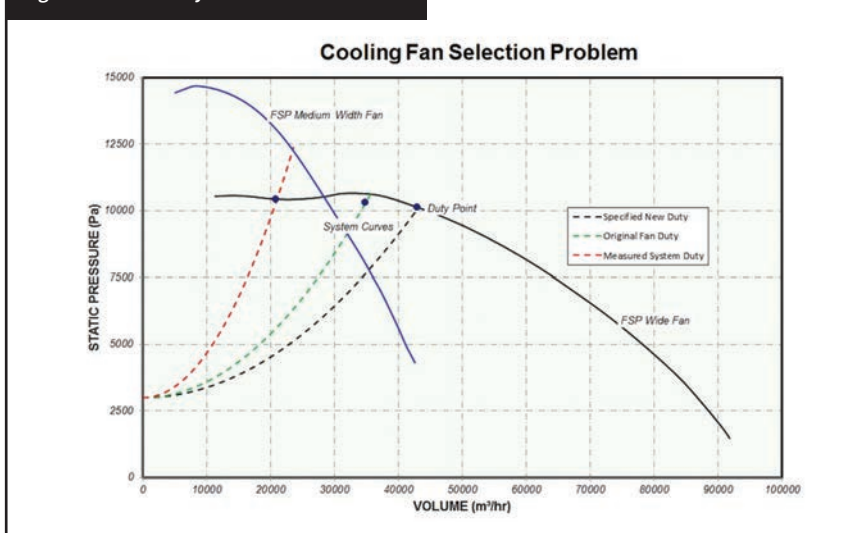
- pressure = const x volume flow²

For these systems, if the fan has been selected close to peak efficiency at the design flow, then this efficiency is maintained at any flow.

Not all fans have a square law system resistance. Fans often have a system resistance curve that does not start at zero but that has a substantial offset. A typical example is clinker cooler fans which often have about half the full low system resistance at no flow. Another example is a fluidised bed, which often has very little pressure change between no flow and full flow. For these applications reducing the speed will move the fan away from peak efficiency or even push the fan into stall. The best choice for flow control of these applications would be dampers.

Although discharge dampers reduce flow they do so by increasing the system resistance, so increase the system losses and can drive a fan into stall. If a fan has been selected with a low efficiency, using a discharge damper can improve the efficiency by making the fan operate closer to its peak efficiency point. However, the efficiency of the fan and damper would always be less than the efficiency of a better selected fan. Another issue with discharge dampers is that they have poor

Figure 1: different system resistance curves



system control, which only really starts to work when the dampers are at least 50 per cent closed.

Inlet dampers close to the fan reduce the flow by changing the direction of the flow approaching the fan blades. This is a more efficient method of controlling fan flow, but also introduces losses into the system. Inlet radial vane controls are the most effective and efficient inlet vanes. However, they are the most easily worn and damaged, so they are not used for applications with a high dust load. With a high dust load the preferred inlet damper is a box damper on top of an inlet box.

With inlet dampers it is possible to have a stable reduction of the flow down to about 10 per cent of full flow. This will mean a drop in peak pressure from open vanes to vanes closed of about 10 per cent. As with any damper the efficiency of the fan with partially closed dampers would always be less than the efficiency of a better selected fan. The main purpose of inlet dampers should be to obtain low flows while maintaining a high pressure rise.

Design for installation

If the fan is a new installation then the fan foundation can be designed to suit the new fan. This leads to the decision of directly bolting the fan to its support or mounting the fan on anti-vibration mounts. For very small fans the flexible connections can be an issue so it is better to bolt the fan down. As fans get bigger anti-vibration mounts are an effective way of preventing the fan's out of balance forces being transferred into the support structure. They are also a good method of ensuring good fan vibration independent of the support structure. Therefore, anti-vibration mounts are typically used for fans mounted on elevated floors or fans where the foundation is not very deep. Larger fans, 2m and above double inlet fans, tend to be on foundation blocks with piers to ensure adequate stiffness. The pedestals are small box pedestals.

Many fans are retrofits. If there is space, the new fan can be installed beside the old fan and then the ducting moved. However, what is usually required is that the new fan fits on the original location. To minimise civil work the existing fan supports should be provided to the fan designer, who should make efforts to adjust the new fan to the original supports. If it is a larger fan on an existing foundation block, this can be especially

Large cement fan on the floor using anti-vibration mounts



A large steelworks fan on a concrete foundation



challenging. The fan casing and inlet boxes will need to be designed to make the fan fit with minimal adjustments to the foundation. Additional bolt holes and adjustments to the foundation geometry may be required. There is also the issue of the condition of the original fan support. This may have been damaged with age or may lack enough stiffness, especially if the new fan is larger or runs at a faster speed. The foundation can be surveyed to determine its condition and tests can be carried out to assess the foundation stiffness. These will reduce the risk of problems being encountered when the new fan is installed.

Another feature to consider is centre-line supporting. This is required for hot fans where the casing growth from temperature would cause the shaft seals to rub excessively on the shaft, giving high shaft seal leakage. If the fan is on a foundation block, this can be

accommodated by having casing feet sitting on the top of the foundation. For medium-sized fans, steel stools would be used. For smaller fans a pedestal mounting arrangement can be designed, even up to 500 °C.

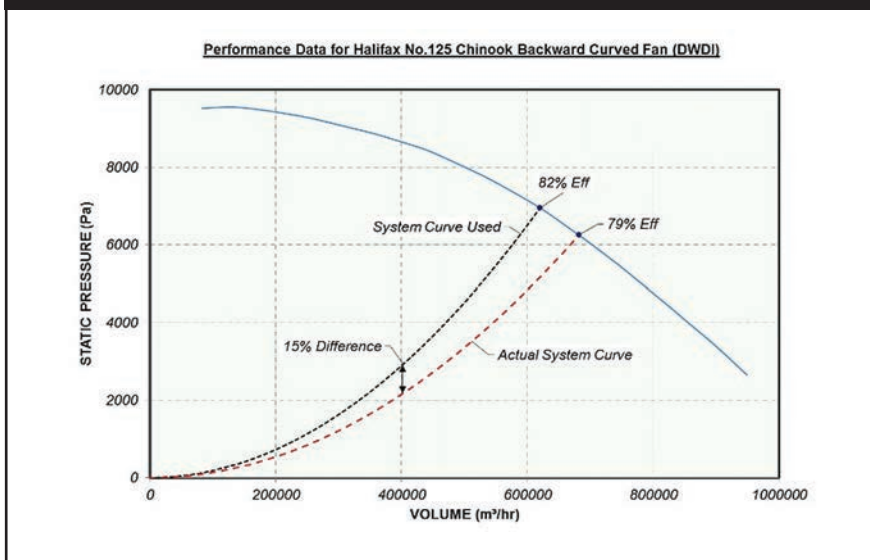
Realising the benefits of improved fans

A new or replacement fan involves a balance between installation costs and running costs. Installation costs include the initial cost of the fan, the motor and the size of the fan. The installation costs are balanced with the running costs and other factors affecting the fan design. However, fans are usually selected using the best information available, which may not be accurate. So, when fans are being specified, engineers usually add in safety factors.

Safety factors can be significant. It is not uncommon to see design pressure

Table 1: average values for two cement plants in 2012

Process	Average fan efficiency	Power consumption (% of total)
Raw grinding	66	25.5
Burning	63	51.9
Finish grinding	66	18.6
Coal grinding	45	5.8

Figure 2: effect of a reasonably small change in operating duty

margins of 20 per cent that result in a fan normally operating with 50 per cent of the design pressure. This can lead to a fan that is supposed to be efficient but is running at a very inefficient point on the curve. This very rarely results in the motor being overloaded.

The most common issues are unexpected performance problems, such as more power being consumed or less flow being obtained. Although more performance is often the main reason for changing fans, this is often also a



Fan with additional casing bracing to prevent cracking from running in stall

consideration when changing fans.

When a fan is being replaced there is the opportunity to measure the duty of the current fans. However, this is usually only part of the information required for the new fan. Typically, there is the requirement to project the current duty to a new proposed duty by taking into consideration system



Installing a new rotor in an existing fan casing with enhanced wear protection

effects. System effects will have a strong influence on both the operating range the fan has to cover and the new operating duty. For sinter cooler fans, if consideration is not taken of the system resistance curve, fans can be selected for a new duty that puts the fans in stall. This has happened when a Halifax Fan customer proposed that more volume could be obtained with no pressure increase. The result was a new fan with the casing cracking.

Ensuring that a fan works close to its peak efficiency can result in significant operating cost savings. Assuming the fan runs for 95 per cent of the time, for each extra kW of power used, the electrical costs increases by approximately GBP500 per year (€563). For a fan with a 100kW power consumption, this gives GBP500 for each one per cent change in efficiency or GBP10,000 over the life of the fan for a 20-year fan life.

A study of cement fans in 2012 found that about 32 per cent of the total plant energy cost of cement plants was from running the fans. Table 1 gives the results by process.

However, electricity is not the only cost in running a fan. Being able to run a fan continuously between planned outages has an impact through reducing unplanned outages to repair a fan or clean a fan. Addressing these costs can have an equal or larger impact than an efficiency improvement of five per cent. Therefore, it is also necessary to consider the impact of fitting a new fan on an existing foundation, especially for larger fans. ■