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Understanding fan test standards and ventilation system performance.

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ABSTRACT: For research and test data to serve a useful purpose, it is essential that it be derived from some known standard test procedure that allows for comparative analysis as well as independent replication and verification. Fan performance data provided by manufacturers is not absolute, and is influenced by the geometry of the test installation. For these reasons, fan test standards have been prepared in order to establish a clearly defined test procedure that allows for comparison of different fan designs, and also allows for the independent verification of test data. Internationally recognized Standards such as BS848, ISO5801, and AMCA 210 etc. are highly prescriptive, and precisely specify test duct configurations, measurement procedures, measurement planes and instrumentation to be used. Provided the laboratory tests are conducted strictly in accordance with the relevant Standard, the performance data will accurately reflect fan performance within the appropriate tolerances for efficiency, pressure and flow when tested in the defined configuration. Major differences in performance exist between a fan tested in accordance with a Fan Test Standard, and an identical fan installed in a mine ventilation system. The aerodynamic performance of any fan will be influenced by the quality of the fan manufacture, the flow system within which it is operating, the location of measurement planes, and the system stability. The flow distributions present in a standardized Laboratory test are unlikely to be repeated in mine ventilation systems, and this will also disadvantage fan performance. Unless the fan supplier takes all of these influences into account, there will an inevitable reduction in fan aerodynamic performance, and the end user will experience a shortfall in expected flow through the mine. The effect on fan performance can range from minor inconvenience to catastrophic fan failure resulting in major rectification costs and significant loss of mine production.

1 Introduction

Provided a fan has been constructed in accordance with the precise geometry specified by the designers, it will perform within the tolerances of the testing standard used to quantify the performance. It cannot be assumed that when the fan is installed in any other flow system the fan performance will necessarily be maintained.

A typical laboratory fan test installation is illustrated in Figure 1. The inlet duct system includes an inlet cone, suitable flow straightening devices and duct transitions specifically designed to produce uniform flow conditions at the fan inlet. It can be seen that the test installation for a relatively small centrifugal fan (1m diameter) is substantial.

The discharge duct geometry illustrated in Figure 1 is designed to ensure uniform flow diffusion with minimal flow separation, with pressure measurement points located towards the downstream end of the discharge duct. This separation is essential for delivering uniform and stable flow conditions required by any of the recognized test Codes and allows for additional static pressure recovery to take place through the mechanism of velocity re-distribution.

It should be noted that flow disturbances downstream from a device will persist for up to 100 duct diameters. A distance of 30 duct diameters is sufficient to recover 97-98% of recoverable energy, but most test rigs operate with less than 10 duct diameters.

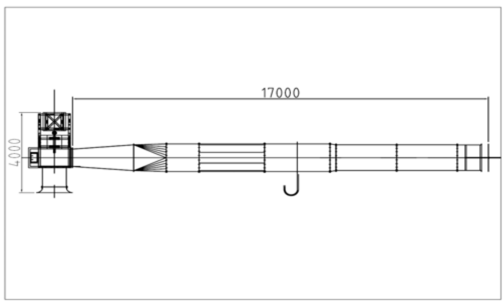

Figure 1. Typical test installation for 1m diameter centrifugal fan.

The measurements recorded at the test stations are subsequently corrected for the duct friction losses between the fan under test and the measurement planes, and the performance of the fan is then calculated. This process will inevitably include all of the pressure recovery that takes place in the length of straight duct between the fan and the measurement station.

When such a fan design is subsequently applied to a mine ventilation system where duct geometries and velocity distributions are not reproduced, the fan performance will be reduced. This fact is the essence of the problem addressed in this paper.

Mine ventilation engineers provide fan suppliers with the estimated mine ventilation duty requirements. The mine ventilation will not include:

1. Allowances for fan manufacturing tolerances
2. Fan installation effects
3. System losses associated with any ductwork or airways that may be required to integrate the fan into the ventilation system.
4. Assessment of fan stability

It is the fan supplier's responsibility to make additional allowance for all of the fan related influences and losses that affect the function of the mine ventilation system, and adjust the specified mine ventilation duty accordingly. Failure to do this will result in an under-performing installation.

2 Fan Related Influences and Losses

2.1 Allowances for manufacturing tolerances

The performance of all fans is determined by the geometric relationships that define each particular design. Any departure from this basic geometry will change the fan performance. Geometry includes dimensional accuracy, surface finish, blade tip clearances, stationery inlet cone clearances etc.

Competitive pressures are the principal reason for failure to always satisfy the geometric proportions and manufacturing tolerances, and the consequences will be reduced aerodynamic performance, high levels of maintenance, higher noise levels and wasted power.

Common departures from basic axial fan geometry are briefly described below:

- The omission of an inner fairing downstream from the impeller will seriously degrade fan performance. Figure 2 illustrates an axial fan with inner fairing fitted. It can be seen that if the inner fairing is omitted, the change in cross sectional area between the fan blade annulus and the diffuser inlet (Plane A-A) will be extreme, eliminating any possibility of a reasonable diffusion process taking place downstream from the impeller. In such circumstances, the velocity pressure loss will be the annulus velocity (generally in the order of 50 m/s).
- It would be incorrect to base the velocity pressure loss on the diffuser nominal area, as the diffuser would be stalled.
- The inclusion of a self closing door at the diffuser inlet (Plane A-A), or some short distance from the inlet to the diffuser will disrupt the inlet velocity profiles, degrading fan performance.
- Failure to maintain accurate blade profiles and blade tip clearances. This will reduce fan efficiency, flow and pressure, and will also increase noise on larger fans by 3dB to 6dB.
- Lack of concentricity between impeller and casing. This will reduce performance and increase noise.

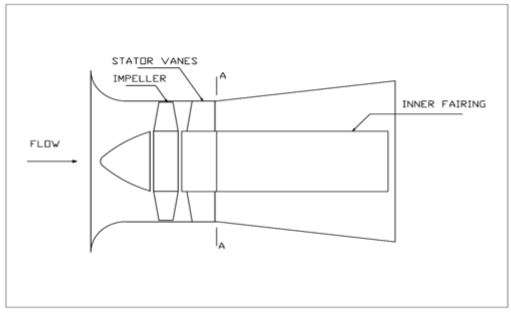

Figure 2. Typical Axial flow fan

Common departures from basic centrifugal fan geometries are:

- Incorrect inlet cone clearances and axial location as shown in Figure 3(a) and 3(b). The purpose of the cone clearance is to provide a uniform flow through the gap between the impeller inlet ring and the lip of the inlet cone. This relatively high velocity high energy flow creates a Coanda effect that causes the flow entering the impeller inlet to "stick" to the inner surface of the impeller inlet ring and the impeller side sheet. If this clearance is too large or irregular, or the inlet cone is incorrectly positioned inside the impeller inlet, the flow will separate from the inner surface of the side sheet. The separated region inside the impeller passages acts as a physical obstruction to the flow through the impeller, and deflects the flow away from the impeller side sheet. The consequent reduction in flow area reduces the total flow rate and the pressure generated by the impeller.
- Incorrect positioning of the impeller inside casing the fan casing affects the aerodynamic performance, and influences impeller axial thrust loads. There is an optimal position that minimizes the axial thrust generated by the impeller.
- Incorrect inlet cone geometry can lead to flow separation at the throat, causing the impeller to stall.
- Incorrect impeller geometry. This results in an impeller of unknown aerodynamic characteristics.
- Incorrect blade setting angles. This also results in an impeller of unknown characteristics.

2.2 Installation effects

The loss of fan performance resulting from poor inlet flow and discharge flow conditions is not always obvious.

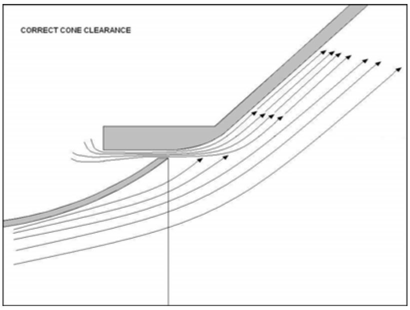

Figure 3(a). Correct inlet cone setting

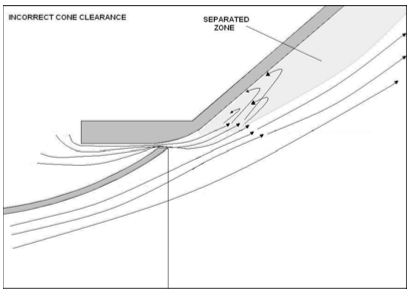

Figure 3(b). Incorrect inlet cone setting

Poor inlet flow conditions will occur when the fan inlet is located too close to adjacent fans, sidewall, footwall or hanging wall. The position of fan accessories such as self closing isolation doors too close to the fan inlet, or the parking of vehicles or storage of equipment close to the fan inlet will also affect inlet flow conditions and reduce fan performance. All of these influences will generate velocity mal-distributions and produce counter or co-rotational velocity components that will affect the angles of incidence between the incoming flow and the internal elements of the impeller. The consequence of these this will adversely affect the flow pressure generated by the impeller, and the overall fan efficiency.

Axial flow fans are particularly sensitive to inlet flow conditions as the fan performance is primarily determined by the blade lift coefficients. The fan blade lift coefficient is primarily dependant on the angle of incidence of the flow entering the blade passages. This fact is often utilized to intentionally modify fan performance by introducing contra or co-rotation to the incoming flow.

Typical blade lift characteristics are shown in Figure 4. It can be seen that relatively small variations in the angle of incidence have a significant affect on the coefficient of lift. When the inlet flow velocity distribution is distorted in an uncontrolled manner, the angles of incidence can easily become extreme, and drive the aerofoil into an unstable region of operation or stall.

Centrifugal fans are also affected by poor inlet flow conditions, and the loss of fan performance can be equally dramatic.

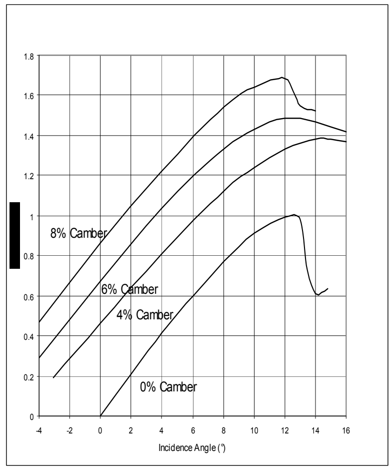

Figure 4. Blade lift coefficients

Figure 6 illustrates the typical effect of poor inlet flow conditions on the performance of a centrifugal fan. The Pressure/Volume characteristic is greatly reduced, and the fan is exposed to the risk of stall operation.

It is not always practical to eliminate unsatisfactory inlet flow conditions, and provision should be included to minimize the effect on fan performance. Unfortunately, there is no reasonably practical analytical method of quantifying the loss of performance, and professional judgment is generally relied upon to estimate the probable effects of inlet flow effects.

2.3 System losses

Whenever a flow system is subjected to a change in velocity, the velocity distribution will be disrupted and separation may occur. This will result in a loss of total pressure. As velocity re-distribution takes place, there will be some recovery of total pressure.

Most importantly, it must be recognized that, without exception, velocity re-distribution does not take place instantaneously. Effective velocity re-distribution in any internal flow system requires time and consequently, distance.

Published data for flow system elements such as bends and diffusers are based on tests that are generally carried out in test installations that have some similarities with fan test standards.

This is for precisely the same reasons ie. for comparative and independent verification of data.

A typical set of published data for circular cross-section bends with mean radius to diameter ratio of 1.00 and Re = 10^6 is shown in Table 1. This information has been extracted from Miller p.207.

Most published data does not define the test duct geometry, and loss coefficients should therefore be treated with some caution and a measure of professional judgment.

Table 1. Bend loss coefficients

Bend Turning Angle(Degrees)	Loss Coefficient
20	0.035
45	0.100
90	0.250

As with fan test data, the loss coefficients associated with any given element of an internal flow system will depend on where the measurements were recorded. The uncertainty is related to the degree of static pressure recovered through velocity re-distribution.

Mine ventilation bend and duct arrangements will not incorporate the lengths of straight duct necessary to allow for the flow to fully redistribute and the pressure to recover. This fact, together with the interaction with other elements in the internal flow system (shaft top bend, bifurcation and section transitions) will in most instances result in system loss coefficients that are much higher than anticipated.

Test rigs usually operate with Reynolds Numbers of 0.5 x 10^6 to 1.0 x 10^6, with the upper limit being controlled only by the physical size and cost of constructing and operating a large installation. The lower level of 0.5 x 10^6 is acceptable as variations in loss coefficients above this Reynolds Number are not significant. It should be noted that most major industrial and mining applications involve Reynolds Numbers from 1.0 x 10^6 to 1.0 x 10^8, and there is no significant body of published test data within this range.

It would not be appropriate to apply the loss coefficients shown in Table 1 to any close coupled application that does not allow sufficient distance for velocity re-distribution to take place.

The use of published data for elements such as bends, contractions and expansions must therefore not be used without correction as actual losses may be 2-3 times higher than published.

For completeness, it should be noted that there are geometric configurations of bends and other elements that can under very specific circumstances result in system losses that are less than the sum of the individual element losses, but these circumstances are rare. D.S. Miller et al have published information that addresses the issue of system influences on loss coefficients.

For these reasons it must be understood that there are no absolute values for loss coefficients for any specific component. The Fluid Dynamics engineer needs to have extensive practical experience in assessing published data in order to estimate the most probable loss coefficient.

3 The Effects of Fan Related Influences and Losses on Fan Performance

3.1 Inaccurate system loss estimates

A typical centrifugal fan performance curve is illustrated in Figure 5. The Design point is shown at point A.

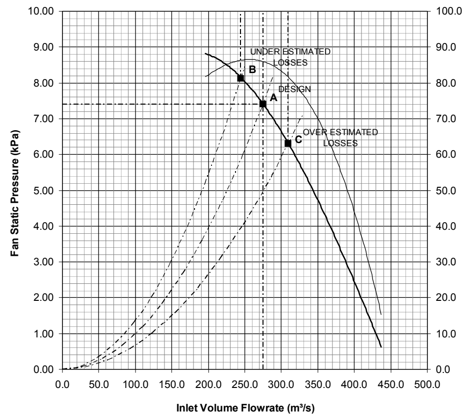

Figure 5. Effect of System Resistance estimates

When system losses are overestimated, the fan will operate further down the curve at some point C. This results in an increase of airflow, and is generally not a serious issue for fans with a non-overloading power characteristic as the absorbed power may not be significantly affected. For fans with an over-loading power characteristic such as axial flow fans operating at high blade pitch settings, the increase in power can be excessive. In such cases the effects of underestimated system losses can be offset by reducing the blade pitch setting.

When system losses are underestimated, the fan will operate higher up the curve at some point B, and this will result in a decrease in airflow. More importantly, the fan will be operating with a reduced margin to the onset of instability or stall, and this could result in the catastrophic failure of static and rotating components.

3.2 Inaccurate Installation Effect estimates

Figure 6 illustrates a typical centrifugal fan performance curve derived from a standard testing procedure. Installation effects as discussed above will degrade the fan performance, and result in some new (but unknown) characteristic X-Y. Flow rates are reduced, and the fan will be operating much closer to regions of instability and stall.

3.3 Assessment of fan stability

The issue of fan stability is very important for any mine ventilation system. The effects of manufacturing tolerances, inaccurate loss assessments and installation effects can transform a nominally stable system into a highly problematical installation. Regardless of whether the mine ventilation system is based on a single fan or multiple fans operating in parallel, the issue of fan stability is pivotal in establishing proper system performance.

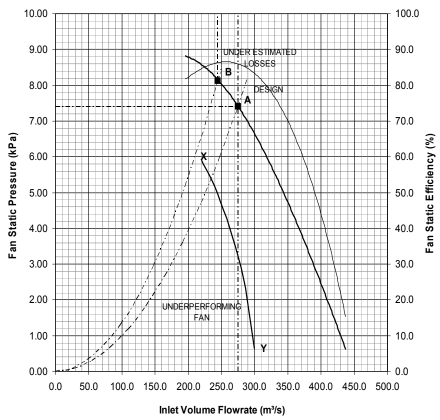

Figure 6. Typical loss of fan performance

The stability analysis is dependant on the accurate assessment of the effects of manufacturing tolerances, installation effects and system losses, and any failure to satisfy all of these requirements will result in a completely meaningless stability analysis.

Centrifugal and axial flow fans are both vulnerable to instability and stall, with axial flow fans being the most sensitive due to the dramatic fall in pressure once the fan is stalled. For this reason, unstable and stall operation of axial flow fans is examined in some detail.

Axial flow fans exhibit a progressive stall characteristic, with multiple stall points occurring as the system resistance is increased. Centrifugal fans exhibit a single stall point at the peak pressure, and further increase in resistance does not result in additional stall points becoming established.

Figure 7 illustrates the progressive stall characteristics of an axial flow fan, and the combined performance of two and three axial flow fans operating in parallel. Curve A-B represents the PV characteristic for a single fan, with C-D and E-F representing two and three fan operational characteristics. The system resistance for the mine is represented by the parabola G-H-J.

Considering the single fan characteristic A-B, the fan will stall if the system resistance is increased to the point where it exceeds point A. At this point A, a single rotating stall cell involving two or three blades will become established, and this will cause the pressure to fall to point K. Any further increase in system resistance will cause the fan to operate along the line K-L. If the resistance is increased beyond point L, a second stall cell will become established and the pressure will drop to some point M. Once again, a further increase in resistance will cause the operating point to move along the line M-N.

This process will be repeated as more stall cells become established, and eventually a point at O is reached when the impeller is fully stalled. At this point, there is no significant flow through the impeller, and the impeller is functioning in very much the same way as a radial bladed centrifugal fan. The characteristic O-P is stable but cannot be considered "useable".

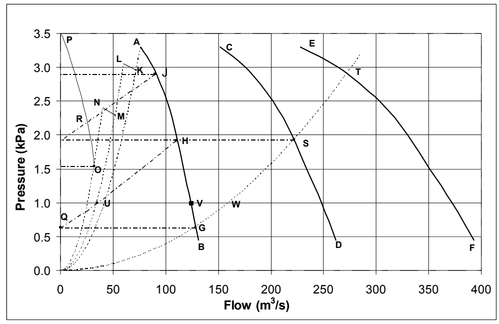

Figure 7. System stability analysis

Fans must never operate close to the first stall point A as this will expose the fan rotating assembly to damage from stall operation.

The system stability is controlled by the fully stalled condition at point O. With only one fan in operation, the fan will operate with complete stability within the range A-B. For the system line shown, a single fan would operate at point G, at pressure Q.

For a second fan to be brought into service, this fan must at start-up generate a pressure that is higher than the line Q-G. It can be seen that this requirement is satisfied, and as the speed of the second fan increases, the pressure that both fans generate will follow the parabola G-S-T. At no time can the Hagen Line Q-H pass above point O.

The Hagen Line is established by constructing several horizontal pressure lines between the pressures associated with points Q and H, and extending these to intersect the fan curve A-B and the system line G-S-T. The difference in flow (at the selected pressure) between the curve A-B and the system line G-S-T is measured. Each flow difference is plotted on the associated pressure line, each time starting from the zero flow point.

The process is best illustrated by an example. Considering the 1.0kPa pressure line that intersects the fan characteristic A-B at point V (at flow of approximately $125m^3/s$) and the system line G-S-T at point W (approximately $160m^3/s$). The difference of $35m^3/s$ represents the flow contribution provided by the second fan when both fans are operating at 1.0kPa. The $35m^3/s$ is plotted on the 1.0 kPa pressure line (point U). This process is repeated for several different pressures, and the Hagen Line Q-U-H is constructed. The Hagen Line passes below the fully stalled condition (point O), and the second fan will operate without instability or stall. When both fans are operating at the same speed, the pressure and flow will be established by the point H.

In this particular example, there will be a problem when the third fan is started. Applying the same analysis again, it can be seen that the pressure H that the third fan has to overcome is higher than the fully stalled point O. The third fan would therefore be restricted to a portion of the stall characteristic O-P. The overall system pressure would therefore be limited by the pressure at R, and full system flow will not be achieved. The system is therefore unstable and unacceptable.

4 Consequences for Performance Degradation

The possible effects of performance degradation on a mine ventilation system are extensive. These include:

- Reduction in air flow
- Excessive noise
- Reduced fan efficiency
- Reduced equipment life
- Higher maintenance costs
- Fan operation in an unstable region or stall.
- High risk of catastrophic failure

Mine production is inextricably linked to ventilation capacity and ventilation system availability, and all of the above consequences will to a lesser or greater degree impact on the mine viability. The cost of correcting any shortfall in fan performance will depend entirely on the specific cause of the problem, and the magnitude of the shortfall.

In some instances, a reduction in airflow, increase in noise, and reduced fan efficiency may be acceptable. However, if these effects are of such magnitude as to become unacceptable, considerable expenditure will be required to modify or replace the complete fan installation.

The consequences of reduced equipment life and higher maintenance costs will also depend on a cost/benefit analysis.

Fan and system instability and stall represent the most serious consequence of performance degradation, and will be the most costly to remedy. Instability and stall will result in accumulative fatigue damage to static and dynamic components and will inevitably result in the catastrophic failure of the rotating assembly.

A catastrophic rotor failure would also represent a major occupational health and safety issue as fan rotors store considerable levels of kinetic energy. A 3m diameter centrifugal fan with a rotor polar moment of inertia of $4000kgm^2$ operating at 1000 RPM stores almost 22MNm of kinetic energy. This is equivalent to a 20 ton vehicle traveling at a velocity of 46m/s (160km/hr or 100mph).

Large centrifugal and axial flow fans therefore have the capacity to cause fatalities as well as considerable consequential loss and damage to adjacent equipment and facilities.

5 Conclusion

Fans may appear to be “agricultural”, but they are in fact highly stressed precision machines. The delivery of a fully functional mine ventilation installation that meets all performance expectations and is reliable and safe to operate requires a high level of engineering design and internal flow systems design expertise.

Manufacturing tolerances, system loss assessment, installation effects and system stability will all affect fan aerodynamic performance and may also have serious fatigue failure implications for all cyclically loaded mechanical components in the fan. Failure to take full account of these considerations greatly increases the risk of catastrophic fan failure and maintenance costs.

Mine production is dependant on ventilation. Any reduction in air flow or loss of system availability inevitably reduces production. The cost of the lost production can run into hundreds of millions of dollars over the life of a mine.

For these reasons, mine owners should not treat the investment in a mine ventilation system as a routine purchase. They should ensure that potential suppliers demonstrate a high level of technical expertise in fluid systems and fan mechanical design. Referees such as previous customers should be consulted before any decisions are made. Price alone must never be the criterion.

References

Miller, D.S, *Internal Flow Systems 2nd Ed*, BHR Group Limited (1990), pp17