As a matter of fact, rendering the task of those in charge of operations and maintenance easier should be the main outcome of recommendations made to the consulting engineers working on the tunnel design, while at the same time ensuring greater safety for motorists.

Variable Pitch Axial Flow Fans for Tunnel Ventilation: A Comparison with Centrifugal Fans

HANS DIETER BAESEL

Variable pitch axial flow fans for application in vehicular tunnel ventilating systems are described. The fan design is of German origin and is considered to represent the state of the art, matured over more than two decades of operating experience. Its use for tunnel ventilating systems in the United States appears to be justifiable. The design and performance are described. Three major areas—energy conservation, space requirements, and number of pieces of equipment—provide the largest savings potential and make this fan design most attractive.

The ever-increasing cost of energy is influencing our personal lives as well as our ability to be competitive in a world market. In that regard, the demand for energy conservation has become not only a political factor, but it is also beginning to play a significant role in today's capital-investment decisions, in that operating costs frequently equal or exceed the cost of the initial capital investment. As engineers, therefore, we are obliged to use all available technology or develop new means to reduce or optimize the use of energy to balance and control its impact on the economy.

The purpose of this paper is threefold:

1. To identify major factors that make axial flow fans the best economic choice for capital investment decisions (i.e., reduction in number of operating equipment and reduction in space requirements);
2. To demonstrate that the variable pitch axial flow fans (VPAFFs) used for continuous flow control will significantly reduce total power consumption, which will result in operating cost savings; and
3. To briefly discuss the VPAFF's reliability, noise emission, maintenance, and general vulnerability to tunnel fires.

This paper will not discuss the theory behind fan laws. It will state theory where necessary and use resulting design criteria as required to demonstrate points and make comparisons.

AXIAL FLOW FANS--THE BEST ECONOMIC CHOICE

Reduction in Number of Operating Equipment to Satisfy Ventilation Requirements

A brief explanation of centrifugal and axial fans is given below:

1. Centrifugal fans: A centrifugal fan is a fan whose inlet air enters the fan parallel to the axis of impeller rotation, is turned, and then leaves the fan perpendicular to the axis of rotation. Centrifugal fans are best suited for low-volume flow, high-pressure application. Their use for higher volume flows is accomplished by using the double-width double-inlet (DWDI) design, where two centrifugal fans essentially operate in parallel.
2. Axial fans: An axial fan is a fan where the air enters the fan parallel to the axis of impeller or rotor rotation and leaves the fan with the air still parallel to the axis of impeller rotation. Axial flow fans are best suited for high-volume flow, low-pressure application. Two or more impellers in series are used to extend the axial flow fans use to higher pressure ranges.

Figure 1 shows a simplified method to determine what type of fan or how many fans must be used to meet a given set of conditions. Three variables (volume flow, adiabatic head, and fan speed) characterize the fan's specific speed and indicate the preferred fan type (axial or centrifugal).

To help explain this concept, the following example is given. The total volume flow for a tunnel ventilation section may be 900 000 actual cubic ft/min (ACFM) at 2-in watergauge (w.g.). In selecting DWDI centrifugal fans, it can be recognized that at least three fans, operating at 300 revolutions/min (rpm) or below, should be used to meet volume flow.

When considering axial flow fans, it can be seen that one or two fans operating at 350-600 rpm or above can be used. Considering the example, the following observations can be made:

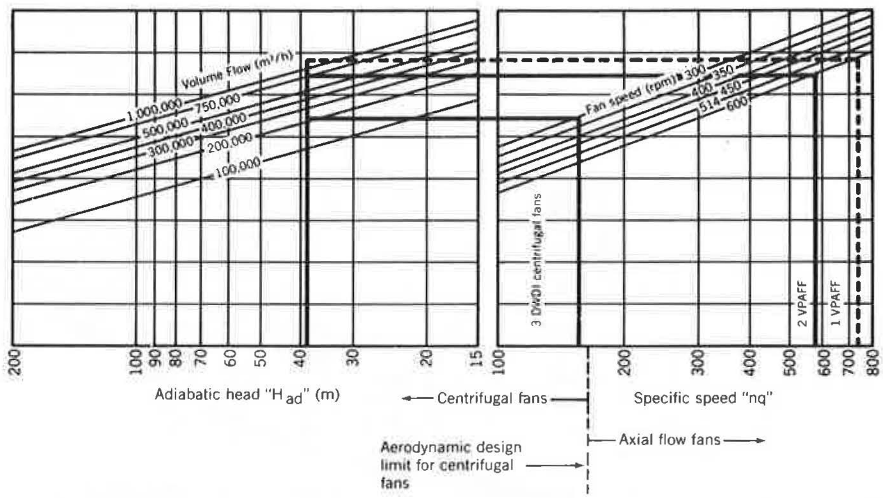
1. Fewer fans are required to move the ventilation air volume flow when using axial flow fans;
2. The higher operating speed of axial flow fans usually permits the use of directly coupled drive motors;
3. Equivalent to the number of fans, fewer drive motors are required;
4. Less motor starting and auxiliary electrical equipment are required;
5. Fewer dampers and damper drives are needed;
6. Fan or motor controls will be simplified;
7. Fewer foundations are required; and
8. The ductwork requirements will be reduced.

Although not quantified in this paper, the items listed above will reduce the overall equipment costs, particularly installation costs.

Reduction in Space Requirements (Reduction of Fan Building Costs)

The costs of the ventilation fan building represent a substantial portion of the total tunnel costs. To keep the fan building costs at their lowest, it is desirable to minimize its physical size. The size of the building, however, is in most cases primarily affected by the number of fans to be installed and their dimensions. With this in mind, the objectives

Figure 1. Fan type selection diagram.



$$nq = n \times \frac{V^{1/2}}{H_{ad}^{3/4}} \qquad (1)$$

nq = Specific Speed

n = Selected fan speed (rpm)

V = Fan volume flow [m^3/s]

H_{ad} = Adiabatic head $\frac{\Delta P}{\delta}$ [m]

ΔP = Total pressure increase [kp/m^2]

δ = Gas density [kp/m^3]

Example:

V = 900,000 [CFM] = 1,529,000 [m^3/h]

ΔP = [2 inches wg] = 50.8 [kp/m^2]

δ = 0.081 lbs/ft³ = 1.293 [kp/m^3]

H_{ad} = = 39.3 m]

for the engineer can be defined as (a) reducing the number of fans necessary for the ventilation system and (b) minimizing the physical dimensions of the fans.

In comparing axial flow fans with centrifugal fans, it has been established that the use of axial flow fans will permit selection of fewer fans for a specific ventilation system. With reference to the above example, one or two axial flow fans are needed compared with three centrifugal fans. With this finding, the first objective of reducing the number of fans is met. (It may be of interest to note that all highway tunnels in Austria are equipped with one fresh air supply fan and one or two exhaust air fans, all of which are of the variable pitch axial flow design.)

Regarding the physical size difference between centrifugal and axial flow fans, the following simplified explanation may provide a better understanding. Comparing a single-width single-inlet (SWSI) centrifugal fan with a single-rotor axial flow fan designed for the same conditions (volume flow and pressure) and the assumption that the flow velocity approaching the centrifugal fan rotor is essentially equal to that approaching the axial flow fan rotor, we can conclude that the total inlet area of the centrifugal fan rotor must be equal to the ring area between the rotor tip and the rotor hub diameter of the axial flow fan.

Because low-pressure axial flow fans need especially small rotor hub diameters, the increase in rotor tip diameter is small to compensate for the lost rotor hub area. Therefore, it will be found that the rotor tip diameter of an axial flow fan will almost fit into the centrifugal rotor inlet dimensions.

Ignoring the influence of rotor blade shape, number of blades, and blade arrangement in a centrifugal fan rotor, it can be stated that the pressure created by the centrifugal fan rotor is proportional to its tip speed. The tip speed depends on rotor outer diameter and rotor speed, whereby the diameter has a reverse proportional relation to the speed.

Since the inlet diameter is set because of volume flow and maximum flow velocity and the outer diameter must be larger than the inlet diameter, a maximum speed will result. We now have to consider that the centrifugal fan rotor is placed in a fan housing that has a spiral-shaped silhouette. This housing acts as a collecting device for the volume flow exiting the fan rotor. Although the housing depth parallel to the rotor axis can be maximized, the housing width and height perpendicular to the rotor axis will still be 80-100 percent larger than the rotor outer diameter. In comparison with the axial flow fan, the centrifugal fan is larger by a factor of 1.8-2.0.

One method of reducing the fan size is to use DWDI centrifugal fans, where only 50 percent of the volume flow approaches each rotor inlet. The inlet area can, therefore, be reduced by 50 percent; however, the inlet diameter of the rotor and its housing will only change by 2, or a factor of 1.414.

Again, by using the example where we identified three DWDI centrifugal fans compared with two axial flow fans, each centrifugal fan will supply only two-thirds of the volume flow each axial flow fan has to move. Consequently, the centrifugal fan inlet area can be reduced by a factor of 0.67, which will reduce the inlet diameter or the housing dimensions by a factor of only 1.23.

Combining the two size-reducing factors by calculating the product, the total factor is 1.74. This result, when compared with the original size difference between axial flow fans and centrifugal fans (factor 1.8-2.0), will lead to the conclusion that each individual centrifugal fan will still be larger than each individual axial flow fan. With this result, the second objective of minimizing the physical dimension of the fans is met.

VPAFFs REDUCE OPERATING COSTS

For the purpose of proving that the use of VPAFFs instead of centrifugal fans will reduce operating costs, it is essential to first discuss the typical performance characteristics of both fan designs in relation to the ventilation system.

Figure 2. Typical performance field of VPAFF.

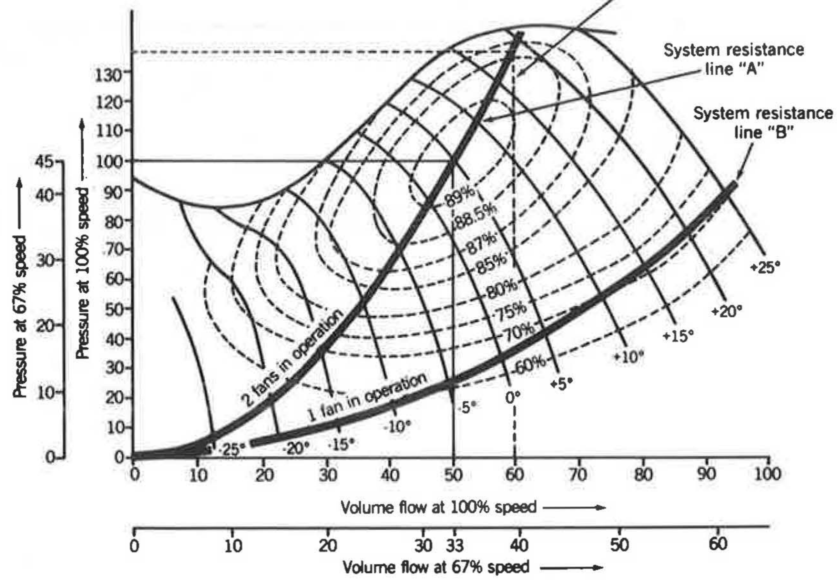
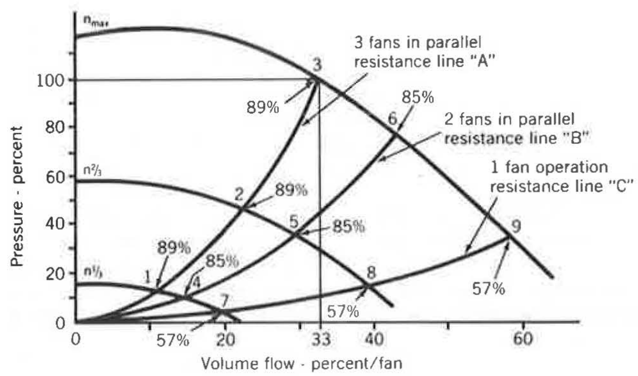


Figure 3. Typical performance field for three-speed radial fan.



VPAFF Performance Characteristics

Figure 2 shows a typical performance field of a VPAFF in an arrangement where two fans operate in parallel. Tunnel ventilation system resistance line A is representative of two fans operating in parallel; each provides 50 percent of the total volume flow. System resistance line B is applicable when one fan is operating. From the performance field, the following observations can be made:

1. Areas of constant efficiency have their maximum extension essentially parallel to the tunnel resistance line, which permits changes in operating conditions over a wide range with high efficiencies.
2. Although each fan is designed to meet 50 percent of the total system volume flow at highest efficiency (resistance line A), 20 percent excess capacity is available.
3. One fan operating alone (resistance line B) can provide 85-100 percent of total volume flow, provided the motor (or motors) is properly sized for the larger volume flow and the reduced fan efficiency.
4. Any volume flow required by the tunnel ventilation system can be met directly by means of rotor blade angle adjustment (intersection points of fan curves for any blade setting with the tunnel resistance line).
5. To increase the extension of high fan efficiencies, and also to maintain high drive motor efficiencies, a second speed, which is normally 67 percent of full speed, is used. (Typically, most ventilation systems equipped with VPAFFs operate with two speed motors or two motors in tandem.)

Centrifugal Fan Performance Characteristics

Figure 3 shows the performance curves of a three-speed centrifugal fan operating in parallel in a three-fan arrangement. Once the maximum 100 percent fan speed (fan design speed) is determined, the two additional fan speeds will normally be selected at 67 and 33 percent of the maximum speed. These conditions are normally met by selecting a single-speed motor for 100 percent fan speed and one two-speed motor for 67 and 33 percent fan speed.

Superimposed on the diagram in Figure 3 are three resistance lines:

1. Tunnel resistance line A is representative for three fans operating in parallel, each providing 33 percent of total volume flow;
2. Tunnel resistance line B is applicable when two of three fans operate in parallel (one fan being shutdown); and
3. Tunnel resistance line C applies when one of three fans operate (two fans being shutdown).

Referring to Figure 3, the following observations can be made:

1. There are nine discrete operating points that can be met (intersection points between the three resistance lines and the three fan speeds) by varying the number of fans in operation and fan speeds;
2. Operating points 1, 2, and 3 are met with maximum fan efficiency;
3. Operating points 4, 5, and 6 are met with a fan efficiency approximately 10 points below maximum;
4. Operating points 7, 8, and 9 are met with an efficiency approximately 25 points below maximum; and
5. In the event of a single fan failure, the

Figure 4. Power savings.

Basis for diagram

$$N^* = \frac{V^* \times \Delta P^*}{102 \times \eta}$$

V^* = Volume flow %

ΔP^* = Total pressure increase %

η = Fan efficiency

N^* = Power consumption

Actual use of diagram

At $N^* = .011$

$$kW = (.011)(900{,}000\ \frac{ft^3}{min})(2\ \text{inch WG})\,(25.4\ \frac{mm}{inch})\,(\frac{1 m^3}{35.31\ ft^3})\,(\frac{1\ min}{60\ sec}) =$$

$= 237$ kW

At $N^* = .002$

$$kW = (.002)\,(900{,}000)\,(2)\,(25.4)\,(\frac{1}{35.31})\,(\frac{1}{60}) = 43\ kW$$

Operating point 1

Power savings 59 [kW]

Power cost savings

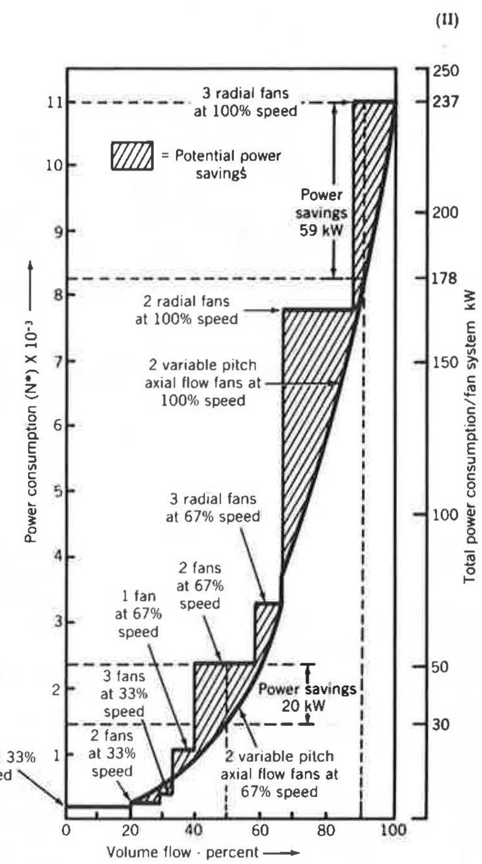
59 x 2 (fan systems) x 1690 (hours)
= 199540 [kWh] x .02 [$/kWh] = 3988 $

Operating point 2

Power savings 20 [kW]

Power cost savings

20 x 2 x 3666 (hours) = 146640
[kWh] x .02 [$/kWh] = 2933 $



remaining two fans operating at 100 percent speed will produce approximately 85 percent of the total volume flow.

With the above description, the basis is established to compare the two systems with regard to their total power consumptions, i.e., three centrifugal fans with three speeds versus two VPAFFs with two speeds.

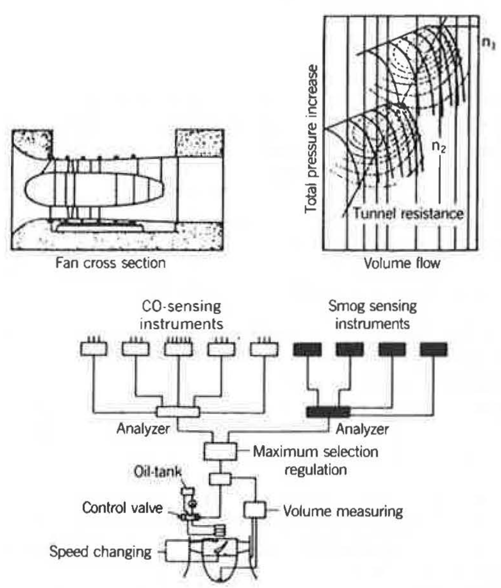
Power Consumption Savings

By using Figures 2 and 3, the total power consumption for each fan system can be calculated. Instead of calculating with actual numbers for volumes and pressures, the calculations were conducted on a percentage basis by using the numbers in Figures 2 and 3. The results are shown in Figure 4.

By using this approach, Figure 4 represents a general tool to conduct power consumption comparisons of the systems; the equations stated in Figure 4 are also important in doing the comparison. The following explanations may be helpful in providing a better understanding of the figure:

1. The diagram shows the difference between a step-control and a continuous-control system (see also Figure 5, which is a schematic of a typical control system for continuous control),
2. The vertical difference between the curve for axial flow fans and the various steps for centrifugal fans for any given volume flow represent the difference in power consumed by the two systems, and
3. The power difference multiplied with the operating hours spent at the various load points and subsequently multiplied with the cost per kilowatt hour results in the operating cost savings that can be realized.

Figure 5. Schematic of fan and tunnel ventilation control system.



To show an order of magnitude for such operating cost savings, the example previously used in this paper was again used to satisfy the equations stated

Figure 6. Cross section of axial flow fan.

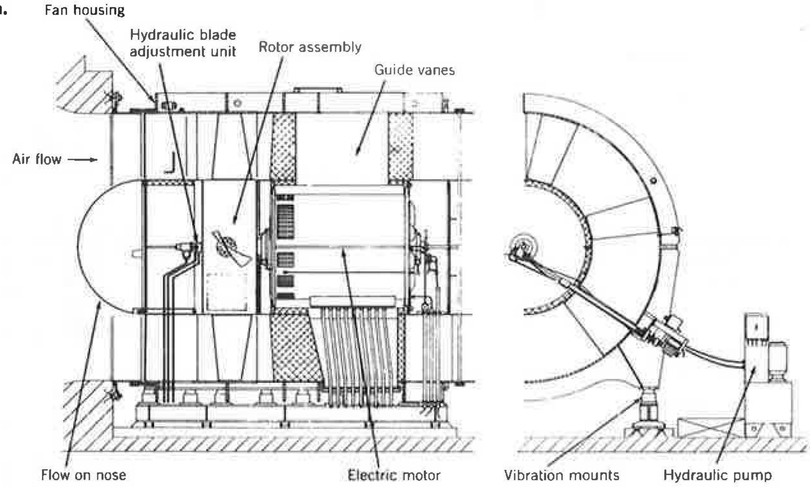
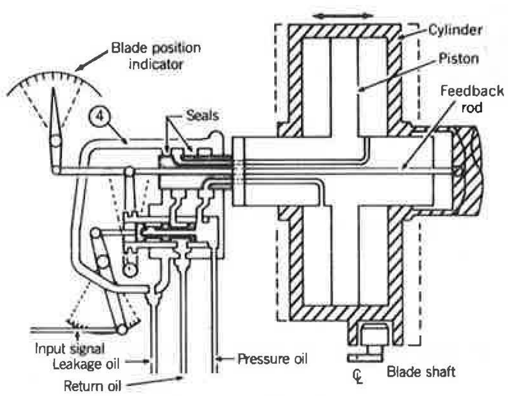


Figure 7. Blade-adjustment mechanism.



in Figure 4. Because the conditions (volume and pressure) meet closely those of a tunnel in a western state that is currently under study, two operating points anticipated in this study were used to compute total operating hours per year.

To identify the overall total for the entire tunnel project, it must be understood that three ventilation sections that consist of six fan systems, as described in the example, are required:

1. Operating point 1: volume flow = 85-100 percent (90 percent) and annual operating hours for a six-fan system = 1690 h; and
2. Operating point 2: volume flow = 39-59 percent (50 percent) and annual operating hours for a six-fan system = 3660 h.

Considering \$0.02/kW•h, the annual operating cost savings by using VPAFFs amounts to \$6920, considering two possible operating points only. Because two-fan systems in the example serve one tunnel tube ventilation section, it must be recognized that the above operating hours represent only 20 percent of the total annual operating hours of three tunnel tubes.

DESIGN FEATURES

After having discussed aerodynamic and performance comparative characteristics of axial and centrifugal fans and the resulting economic differences in both equipment and operating costs, it is appropriate to address design features as they relate to reliability, maintenance, noise, and, most controversial, vulnerability to fire or high temperatures.

Reliability and Availability

Operating records of centrifugal fans in the United States and VPAFFs in Europe show availabilities, on average, in excess of 99.9 percent. To prefer the one design over the other based on availability is, therefore, not practical. It is also of theoretical value only to state that the probability of a single fan failure increases with the number of operating fans. The probability of a single fan failure when using centrifugal fans is, therefore, theoretically higher than experiencing a single fan failure when using axial flow fans, due to the fact that more centrifugal fans are needed for a ventilation system.

For practical purposes, considering that the average availability per fan is less than 100 percent, a single fan failure is probable and will occur. Considering, however, that two of three centrifugals fans or one of two axial flow fans are sufficient to satisfy tunnel ventilation requirements, the reliability of the entire ventilation system is maintained at an extraordinary high level.

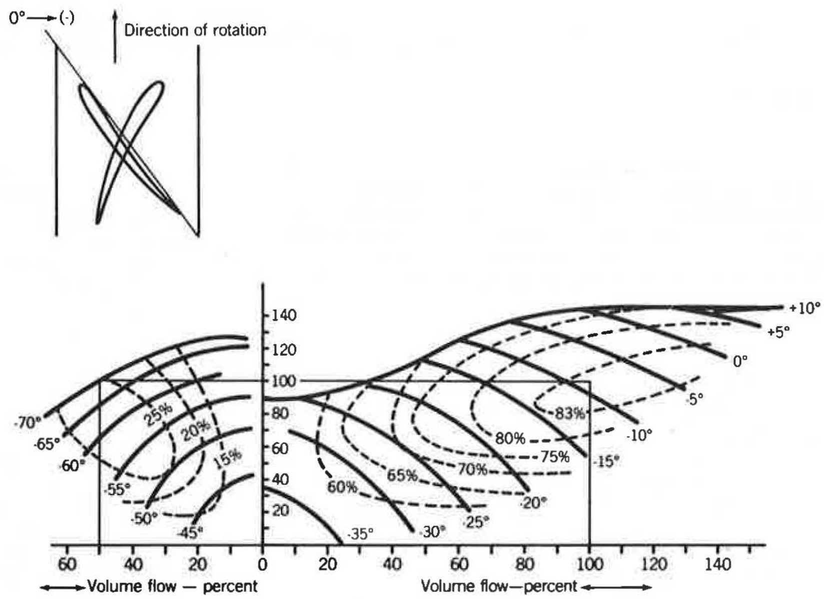
Maintenance

Routine maintenance is required for all fans, regardless of their design. On centrifugal fans, maintenance is normally limited to main bearings, motors, and belt drives. On axial flow fans, as shown in Figure 6, maintenance is required for the motors, blade shaft bearings, and the hydraulic blade-adjustment mechanism (Figure 7), as well as the hydraulic oil supply units.

Well-designed axial flow fans have split fan housings that have a section for easy and fast removal to provide access to all interior components. The interior components are designed for quick removal. A spare unit, which consists of a rotor, motor, and hydraulic blade-adjustment mechanism, should be maintained in stock so that, in case of a premature failure, a fast exchange can be made.

Maintenance cost on axial flow fans will be somewhat higher than on centrifugal fans. This factor should be considered in the overall evaluation.

Figure 8. Reverse flow through blade overturning.



Noise

Emitted fan noise is essentially a function of fan rotor tip speed, pressure developed by the fans, power consumption, and number of fans operating in parallel. The highest sound pressure level will be found at the blade-passing frequency of the fan rotor.

Although centrifugal fans for tunnel ventilation systems usually have eight blades, well-designed axial flow fans will have four to six blades. Despite the higher operating speed of axial flow fans, their blade-passing frequency will not exceed that of centrifugal fans.

The disadvantage of an axial flow fan remains with its higher tip speed; e.g., on a one-to-one comparison, the emitted noise level of an axial flow fan will be higher than that of a centrifugal fan. This disadvantage, however, is eliminated when a system of three centrifugal fans operating in parallel are compared with a system of two axial flow fans operating in parallel. In conclusion, it can be stated that neither fan design will offer an advantage.

Vulnerability to Tunnel Fires

To discuss a fan's resistance to high temperature, it is necessary to first identify all critical fan components that will be exposed to such high temperatures.

Regarding centrifugal fans, the main shaft and the complete rotor will be exposed. To design these components for high-temperature resistance will require expensive special alloy steels that will substantially increase the fan costs, especially when one considers the affected component sizes and the number of fans.

On axial flow fans, the only directly affected components are the fan blades. To build the blades from special alloy steel is easily possible. Although the fan cost will increase substantially, it will be disproportionately less than for centrifugal fans, considering that only four to six blades per fan are required and fewer fans are needed.

All other components of the fan are well protected in the fan hub center. The fan hub can be heat insulated and purged with cooling air.

Another beneficial feature inherent to VPAFFs is that the blade pitch can be reversed, thereby causing reversed flow (see Figure 8). This feature may provide an added tool to control tunnel fire. In conclusion, it is safe to say the VPAFFs have a greater flexibility to meet upset tunnel ventilation conditions.

SUMMARY

VPAFFs have been used for tunnel ventilation applications in Europe for the past decade. Today's state-of-the-art design is the end product of consistent fan component design improvements based on operating experience. The reliability and recorded availabilities easily match those of conservatively designed centrifugal fans. The basic advantages of power savings, which result in reduced operating costs, and small physical size and lower number of operating equipment, which reduce capital investment requirements, make this fan a viable alternative to centrifugal fans for tunnel ventilation systems.