Study of Axial Flow Fan Components and its Failure Affecting the Performance

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Abstract---Ventilation is a critical task in underground mining operation. Ventilation controls the air movement, its amount, and direction which provides adequate fresh air to miners for a safe and comfortable working environment. Fans, which provide air flow to different faces of a mine, have a great role of impact in ventilation systems. In this paper study has been carried out on the different components of an axial fan and their failures which is affecting its performance. For that Blade failure, Shaft failure, Bearing holding U-bolts, pitching angle, Blade chord angle, Tip clearance, Erosion and corrosion parameters have been considered.

Keywords: axial fan; failure analysis; blades; hub; shaft; Bearing; Tip clearance; Erosion and corrosion

I. INTRODUCTION

In ventilation systems, fans have major role. Fans (ventilators) are major devices in ventilation networks that utilizes the mechanical energy of a rotating impeller to produce both movement of the air and an increase in its total pressure and provide airflow to different parts of a mine. Fans are made of a wheel confidant (i.e., a wheel which covers by blades). Fans work in two modes as exhaust mode and blower mode. An axial flow fan moves air or gas parallel to the axis of rotation by the change in velocity of the air passing over the impeller blade By comparison, a centrifugal or radial flow fan moves air perpendicular to the axis of rotation. Axial flow fans are better suited for low-resistance, high-flow applications, whereas centrifugal flow fans apply to high-pressure resistance, low-flow conditions. Axial fans are divided into three groups - propeller, tube axial, and vane axial, whereas Centrifugal fan are classified as Airfoil blade, Radial fans, forward curved fans and backward inclined/curved fan. This paper includes the affects on the performance of axial fan due to the failure of different parts.

Failure of fan major classified as:

- Catastrophic, structural failure.
- Major disabling, mechanical or electrical failure.
- Failures requiring immediate but temporary shut-down.

Fortunately catastrophic failures are rare but they do happen. Lesser failures may be precursors of catastrophic failure and the root cause can be similar. Result of catastrophic failure can be devastating the energy stored in large = 500kW centrifugal fan at full speed is equivalent to about 2-3kg of TNT and is already in kinetic form. Many failures are non-catastrophic only because protection systems shut the fan down before the critical point was reached. Some potential causes are:

- Inadequately designed: replacement impeller or repairs or modifications to impeller or shaft.
- Incorrect operation: Poor commissioning procedures, Over-speed, Cancelling alarms or over-riding trips.
- Design: Poor impeller structural design, Poor shaft or foundations design (critical speed close to running speed), incorrect impeller material selection like yield strength too low or material too hard (brittle fracture).
- Manufacture: Weld failures due to incorrect consumables or procedure and Poor quality assurance.

Fan stall resulting from too high system resistance (incorrect initial estimate or increase over time). Fan stall resulting from under-performing fans (wear etc.). Fan instability in parallel fan systems. This is usually also due to high system resistance and/or inadequate fan margins. Fan damage resulting from explosion. Adequate explosion doors should be installed at the shaft top. Avoid mounting fans over the shaft.

Parts failure affecting the performance of axial fan to be considered are listed below.

A. Air Fan Blades
   1) Pitching Angle
   2) Blade chord angle
B. Hub Size
C. Fan Shaft
D. Bearing
E. Holding U-Bolts
F. Tip Clearance
G. Erosion and Corrosion

A. Air Fan Blade

(1)There is an investigation has been carried out for the failure analysis of blades in the Xinji colliery, test has carried out on a special type of fan consists of 12 blades (cast from an aluminium-silicon alloy) installed by a locking handle to a center spindle. And, it found failed during service when it supplied under the conditions of a 3000 r/min turbine rotation with a power output of 118 MW.

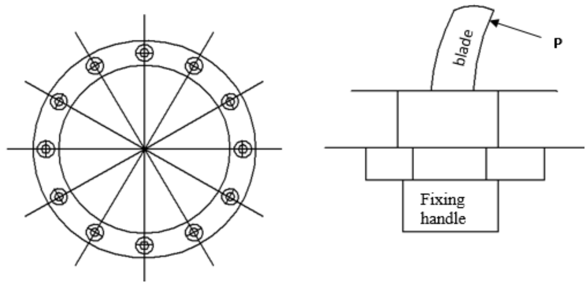

Fig. 1: Segments and locking of blades

Just after a week of running fan it results in to the fractures were at the root of each blade. As well, the failure took place at the locking handle of blade No. 10.some analysis has been carried out to understand the real picture of the failure that are:

1) *Metallurgical analyses:*

It consist of macroscopic and microstructural observations(e.g. Visual inspection, Fractography and microstructure)

a) *Visual inspection:*

Two kind of failure were found on blade No. 10: one at the locking handle of the blade and the other at the root of the blade. And others blade found to be broken only at the root of the blades.

b) *Fractography:*

Fractography Analysis of the cracked locking handle of blade No.10 shows that the brittle fracture area with a smooth surface and shallow depth was very large, about 90% of the entire fracture surface. Derived from one point of the blade edge where there might have been a large number of shrinkage holes, the crack initiation was around the edge of the blade in an approximate semi-elliptical area and ended at the central part of the locking handle. A marked directionality of the microstructure at the beginning fracture surface, oriented perpendicular to the axis. The features of the entire fracture indicate that brittleness could have played a strong role in the propagation of the crack.

2) *Microstructure:*

Metallography samples were prepared by using standard metallographic techniques and etched correctly. Some samples from blades were used for analysis by an OLYMPUS optical microscope with magnification factors 4, 8, 20, 50, 100…. which showed that the microstructure had marked defects such as coarse grains, a large quantity of shrinkage holes and a loose structure. Striations of cleavage fractures or quasi-cleavage fractures, which accounted for the brittle fracture, were very distinct.

3) *Mechanical examinations:*

It consist of Impact toughness and hardness performance tests.

a) *Hardness test:*

Hardness was measured at three different points of each sample from blades number 10, 2 and 11, using a Brinel hardness testing machine. The results show that each hardness value exceeded 120 HBS, far beyond the recommended value of 70 HBS for cast aluminium-silicon alloys, indicating that the blade material was unusual (Table.1).

Sample	Point 1	Point 2	Point 3	Average	Overall average
From blade 10	128	123	149	133	129
From blade 2	142	122	131	132	
From blade 11	134	122	112	123	

Table. 1: Experimental results of hardness test (HBS)

b) *Impact toughness test:*

Impact toughness performance tests were carried out in which four clear impact test pieces from blade No.10 were prepared in standard dimensions of 10 mm□10 mm□50 mm. The average toughness was as low as 3.37 J/cm2, far below the recommended value of 30 J/cm2 for cast aluminium-silicon alloys, which further provided evidence of the unusual brittleness of the blade material (Table. 2).

Sample	Toughness (J/cm2)	Impact energy (kg(J))
1	3.92	3.92
2	2.94	2.94
3	3.18	3.43
4	3.43	3.62
Average	3.37	

Table. 2: Experimental results of impact toughness test

(1) Other than these tests some more inspection has been carried out to clear the picture of the failure of blades e.g. examining of fitting and tightening of the blade and found no fracture after defect.

(2) Dust analysis of the lubricant taken from the center of the spindle and result showed that striations on the outer surface of the handle of blade No.10 were caused by vibration and the spin off because of improper conjunction between the locking handle and the center spindle.

4) *Pitching Angle*

The variation pitch angles of the impeller improving flow and static pressure of were analyzed to improve the flow rate and the static pressure of a turbo fan. After we analyzed effect of the pitch angle variation on the flow rate and static pressure, we reached the following conclusion.

(1) By comparing the flow rate of 1,175 CMH, 1,270 CMH,1,340 CMH, and 800 CMH of the each pitch angle of 44°, 54°, 59°, and 64° respectively, the largest flow rate can be obtained by the pitch angle of 59°.

(2) The static pressure difference between the impeller by the pitch angle variations of 44°, 54°, 59°, and 64° were, 120 Pa, 214 Pa, 242 Pa, and 60 Pa, respectively. The pitch angle 59° showed the highest static pressure.

(3) In order to increase the flow rate and the static pressure of the axial flow turbo fan, the 59° of pitch angle should be adopted.

5) *Blade Chord Angle:*

If a fan continues to operate outside of the stall region of its performance curve, air flow will continue to increase as the chord angle is increased from about 20° to 60°. Many existing fan systems are operating in unknown areas of their performance curve and a change in chord angle gives unpredictable results. As shown in the figure it is clear that on increasing the chord angle the flow rate also increases.

For large commercial machines, the upwind, three-bladed rotor is the industry-accepted configuration. Virtually all large machines installed during the last several years are of this configuration. The three-bladed rotor offers the following advantages over the two-bladed configuration. Although the upwind choice is based largely on noise considerations, it also results in lower blade fatigue. Tower-shadow noise and impulsive blade loading for an upwind

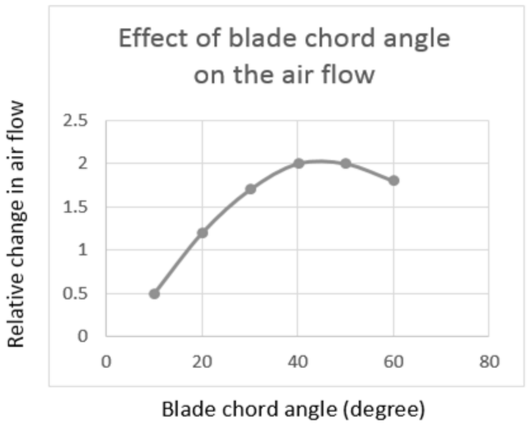

Fig. 2 Numbers of Blade Affecting the Blower Design

rotor are less than for a downwind rotor that passes through the tower wake. For an upwind rotor, the blade-number choice is then a balance among blade stiffness for tower clearance, aerodynamic efficiency, and tower-shadow impulsive noise. The three-bladed rotor configuration appears to provide the best balance. For a given radius and airfoil thickness, more blades result in lower blade flap stiffness. With three blades, adequate flap stiffness is still achievable to avoid tower strikes and the blade loading is low enough to avoid annoying impulsive noise. Aerodynamic efficiency increases with increasing blade number 2 with diminishing return. Increasing the number of blades from one to two results in a six-percent increase in aerodynamic efficiency, whereas increasing the number from two to three yields only an additional three-percent. Further increases in blade number sacrifice too much blade stiffness for a minimal increase in aerodynamic efficiency. For small machines, the aerodynamic-efficiency increase resulting from more blades for a constant solidity rotor is diminished somewhat by the lower Reynolds numbers.

B. Hub Size and Effect

The question of how large a fan hub should be is commonly asked. The answer is simple: A hub must be large enough to pick up where the blades are no longer able to carry the load. As the radius is reduced and the center of the fan approached, the reduction in the speed of the blade section reduces the potential work which may be accomplished by the blade and increases the mean blade angle. Blade width will begin to increase abnormally and the blade angle will rise sharply until additional width and angle are no longer practical. At this point the hub must begin.

The hub serves two major (aerodynamic) purposes. It allows termination of the blades at a point where they would cease to function efficiently and it prevents back flow of air through the center. If the hub is too large for the required performance, the result will be an increase in velocity pressure, due to the smaller net opening, and subsequent waste of power. If the hub is too small for the required performance, the result will be deterioration of the flow near the hub, possibly even a reversal of flow in this area. Reference to the drawing on the following page will readily show the necessity of a hub of some proportion. As the centerline of the wheel is approached, the width of the blade becomes infinite. For practical reasons, it is evident that the hub in this example should start at or near the "RADIUS = 2" point to avoid excessive blade depth.

The fan hub must, of course, also serve a structural function in connecting the blades and imparting rotation to them. The hub size required by the aerodynamic considerations discussed in the preceding paragraphs would result in an extremely heavy (and expensive) structural member. For this reason, fan manufacturers usually provide a hub that is inadequate from an aerodynamic point of view. Moore solves this problem by providing two hub designs for each series of fan: A smaller structural hub and a properly proportioned aerodynamic hub referred to as the Air Seal.

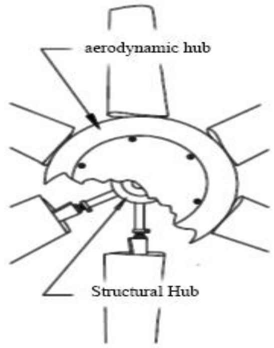

Fig. 3: Fan Hub

C. Bearing Life

Fan applications demand quality bearings. Rolling element (or anti-friction) bearings are good selections for fan applications because they offer low friction and are reasonably priced.

Under laboratory conditions with controlled loads and lubrication, bearings fail due to fatigue. Rolling elements and bearing races eventually start to fail from the repetitive loading and unloading. In theory, all bearings have a finite life and will eventually fail. Figure 8 shows that as with light bulbs and humans, as time goes on the probability for the end of life increases. Just as statistical methods can estimate human life expectancy, they can estimate bearing fatigue life. The Anti-Friction Bearing Manufacturers Association (AFBMA), and the International Organization for Standardization (ISO) specify the method used in determining the expected life of a bearing. They define L-10 life as the number of hours 90% of a group of identical and identically loaded bearings will survive without failing. This term, also called “minimum” life, is a function of bearing type, load, and speed. Note that the number of hours (or revolutions) and the load conditions must be specified. For example, saying that a bearing must have an L-10 life does not mean anything, while specifying an L-10 life of 40,000 hours at a fan’s maximum design condition is distinct.

CLASS OF MACHINE	**L-10 LIFE (HRS.)**
Machines requiring extreme reliability for use 24 hrs./day: Large electric machinery, power Station machinery, mine pumps, mine ventilator fans, tunnel shaft bearings for ocean-going vessels, paper making machinery.	≈100000

Table. 3: class of machine

L-10 life is related to fatigue failure. From a practical standpoint, specifying an L-10 life over 200,000 hours offers very little advantage since other factors such as relubrication frequency, lubricant breakdown, or lack of cleanliness will cause failure before fatigue does. In fact, specifying an L-10 that is too high can result in a bearing with not enough load. With too little load, rolling elements can skid against the races instead of rolling properly. In time, this will lead to a bearing failure. Frequently, factors other than fatigue have the most influence on bearing life. As an example, a bearing on a fan operating in a hot, moist, and extremely dirty environment with an L-10 life of 45,000 hours is likely to have its life limited by seal failure, lubricant breakdown, Rust, or dirty lubricant. This may occur before 45, 000 hours.
The equation for L-10 life in hours is:
$L\text{-}10 = (16666/n)*(C/P)^{p}$ ………..eqn. 1
In this equation, n is the speed in RPM, C is the dynamic load rating, P is the equivalent radial load, and p is 3 for ball bearings and 10/3 for spherical roller bearings.

D. Tip clearance

Tip clearance between the blade and inside of the fan cylinder or fan ring is critical to proper axial flow fan performance. Hudson fan curves are derived from tests conducted at Texas A&M University's Engineering Laboratory,

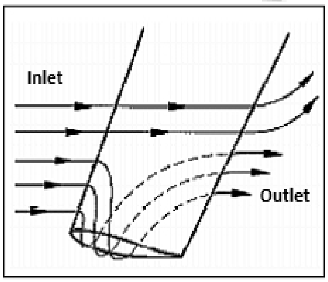

Fig. 4 : Tip Vortex (Leakage)

based on very close tip clearances with optimum inlet conditions. If close tip clearances are not maintained, a loss of performance will be noted, in pressure capability and airflow. The primary reason for close tip clearance is to minimize air loss or leakage around the tip, known as a tip vortex. (See Fig. 4). Hudson recommends the use of safety factors to correct for the environment in which a fan is placed.

Exit air will be at a higher pressure than incoming air due to the work expended by the fan. Note that the blade performs most of the work in the outer portion of the airfoil. If a leakage path (or clearance) exists, the air will seek the path of least resistance and bypass the tip, causing a vortex and loss of performance. Tip clearances can be reduced with Hudson accessories. Fig. 6 shows the API recommended tip clearances for smaller (16ft and less) diameter fans. Also included in this figure are the nominal tip clearances for larger diameter fans.

Fan Diameter	Minimum	Maximum
3ft through 9ft	1/4 in.	1/2 in.
>9ft through 11ft	1/4 in.	5/8 in.
>11ft through 16ft	1/4 in.	3/4 in.
18ft through 40ft	1/2 in.	1 in.

Table. 4: Tip Clearance

E. Erosion and Corrosion of Blade:

By the practical observations and by the customer complains it is found that after some time due to the moisture effect the surface of the blade get affected due to the corrosion. Corrosion causes the surface rough in the nature. As the air flow strikes to the rough surface the flow of air will be deflected, this deflected air will not follow the aerodynamic shape of the blade and will strike to the casing. This striking of the air causes the vibration and that will affect the design as well.

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