Flow performance of highly loaded axial fan with bowed rotor blades

L Chen, X J Liu, A L Yang and R Dai

School of Energy and Power Engineering, University of Shanghai for Science and Technology, 516 Jungong Road, Shanghai, 200093, China

chen_liu@usst.edu.cn

Abstract. In this paper, a partial bowed rotor blade was proposed for a newly designed high loaded axial fan. The blade was positively bowed 30 degrees from hub to 30 percent spanwise position. Flows of radial blade and bowed blade fans were numerically compared for various operation conditions. Results show that the fan’s performance is improved. At the designed condition with flow coefficient of 0.52, the efficiency of the bowed blade fan is increased 1.44% and the static pressure rise is increased 11%. Comparing the flow structures, it can be found that the separated flow in the bowed fan is reduced and confined within 20 percent span, which is less than the 35 percent in the radial fan. It means that the bowed blade generates negative blade force and counteracts partial centrifugal force. It is alleviates the radial movements of boundary layers in fan’s hub region. Flow losses due to 3D mixing are reduced in the rotor. Inlet flow to downstream stator is also improved.

1. Introduction

Air cooling is an important technique in large electric motors and power generators to remove the inside heating from electric-magnetic stator. It is important to keep generator inside temperature under limitation for safety and efficiency. Cooling air is usually supplied with one multi-stage axial flow fan which can provide required pressure rise capacity with flexible combination of stages. It is motor designer’s wish to reduce the number of fan stages and then to reduce motor shaft length for shaft stiffness and vibration considerations. This can be achieved by either increasing the blade speed or by increasing the stage load coefficient. Since fan is fixed with motor shaft, the blade speed is limited by the motor rotation speed which is usually 3000 or 1500 rotations per minute (RPM). A higher pressure rise will require increased stage loading.

Designing of highly loaded and efficient axial flow fan stage has been a long time research effort. Two possible ways are to increase fan’s rotational speed and flow turning through the blade row. Boundary layers along the blade airfoil are thicken by the increased flow diffusion and tend to separate at blade trailing edge. Rotating centrifugal force drives the low momentum fluids in hub region toward mid channel to mix with main flow. They accumulate and form the hub separation region at the rotor exit, which becomes the major flow loss resource in the highly loaded fan.

Emmerson[1] and Calvert[2] have carried out a high load transonic compressor test, and designed an new airfoil with the turning angle of root reaching 60 °. The numerical simulation results showed that the performance basically reached the requirements. Friedrichs [3] has tested the performance of a high load low speed axial compressor, whose the turning angle of stator airfoil exceeds 60 °. The
results showed that curved vanes can effectively delay the separation in bottom corner on suction side and make the compressor flutter margin increased, as well as the efficiency increased by about 10%.

All the research of the compressor cascade shows that the high loaded van can improve the structure of flow field and improve compressor efficiency. However, in order to enhance the level of the compressor pressure ratio, increased blade loading, the high loads design for the compressor impeller is needed. As Compressor Cascade diffuser flow characteristics, with cascades load increases, the adverse pressure gradient enhancement, blade suction side and end walls prone dimensional separation, secondary flow loss increases, the efficiency decreases. In order to effectively improve the compressor load, while able to guarantee constant efficiency, the introduction of curved blade technology [4].Bogod [5], tested six different forms of exit guide vanes curved grating used in a subsonic typical multi-stage compressor stages and results showed that all five curved blade level have improved the overall characteristics. Former research publications have revealed that the flow through high-loaded fan is strongly three dimensional and suffered from the local hub separation. Alternative complex 3D optimization methods have integrated sweep, bowing and controlled diffusion concepts to design highly loaded fan rotor, but it is computational expensive to reach design target.

In this paper, partial bowed rotor blade was proposed for a newly designed high loaded axial fan. The numerical simulation is carried out with NUMECA code and the influence of bowed technology on the performance of compressor is studied, which can afford the basic support for the high load axial fan with curved blades rotor technology

2. Numerical Model
On the base of NACA-65 series leaf type basis, the single arc in the arc design method is adopted to design the bowed airfoil with the power efficient is 0.55 and the diffuser factor of the impeller and vane is 0.54 and 0.5 respectively. The model of airfoil is showed in Figure 1.

![Blade Airfoils for NLF and HLF](image)

**Figure 1.** Blade airfoils for NLF and HLF

Comparison the performance with different bending height of prototype compressor under the design speed through numerical simulation, 30% of blade height is chosen since the best efficiency. Traditional bending blade design methods is employed, by offsetting airfoil along circumference and bowing the root parts of impeller blade. The distribution of stacking line is composition of parabolic and straight line in radial direction with the bowed angle is 30°. Its model is shown in Figure 2.
3. Numerical Method

Numerical simulation is carried out with software. H-type grids is used in the fan inlet and outlet and O-grid is employed along the blade channel. The grid distribution of impeller channel and blades along circumferential and spanwise and streamwise is: $41 \times 61 \times 105$ and $17 \times 61 \times 161$. In van channel and blade, the grid distribution is $41 \times 61 \times 101$ and $17 \times 61 \times 153$. There are 1.9mm clearance on the top of vane blade. The butterfly type is used with 13 layer in radial distribution. The independent verification is made with the grid number between 650000-950000, and the 820000 is adopted. The same grid topology is used in both straight and bowed stages, shown in Figure 3. All the solid wall insulation is given the no-slip boundary condition; and the temperature, total pressure and axial intake direction is fixed inlet with the given mass flow in outlet.

![Figure 2. Schematic of curved rotor blades and blade shape stacking line](image)

4. Result and Discussion

The distribution of performance parameters is shown in Figures 4 and 5 respectively. It’s obviously that the bowed stages is superior in efficiency and power ability for 1.44% is increased in efficiency and 11% in static pressure head. It’s illustrated that the flow condition is improved, and the flow loss is reduced by using bowed blades.

![Figure 3. Grid of straight stage and bowed stage](image)
The limiting streamlines of impeller and vane blade are showed in figure 6 respectively. The adverse pressure gradient is reduced along blade suction side, which slowed the development of boundary layer and pushed the separation point back, as well as weakened the impact on the radial flow. The process in vane is good. On the contrary, the wake from rotor made the boundary layer thicker in the van, and the separation flow area is enlarged about 30% of blade height, which caused the more flow loss in straight stages.

Figure 4. Efficiency vs. mass

Figure 5. Static pressure head vs. mass

Figure 6. The streamline in suction surface

Figure 7 shows contour map in S3. There are large area of low velocity in the root parts of straight stages, especially in the vane area, the velocity is almost zero, which cause higher losses. In the bowed stages, the area of low velocity is obviously small, low energy fluid can’t accumulate at the suction side of the root easily, thereby reducing losses.
Figure 7. Contour map of velocity in S3

Figure 8 shows the parameters distribution along the outlet blade height in design conditions. All the parameters are mass-weighted average value along circumference. It’s easily proved that bowed impeller blade can improve the flow dramatically at the root part. The crosswise pressure gradient is weakened, and the end wall boundary layer is postponed, which reduced the accumulation of low energy fluid at the corner of suction surface. The efficiency is increased under 40% of blade height by using bowed blade.

Figure 8. The parameters distribution in spanwise

Definition of the energy loss coefficient in impeller is

$$\bar{\omega} = \Delta I \left( \rho U_{mid}^2 \right)^{-1}$$

Since the presence of the tip clearance, the leakage flow caused more losses. The bowed impeller has a slight advantage than straight stage for the delayed separation at the root area. And it’s almost the same in other parts, as shown in Figure 9.
Definition of the total pressure loss coefficient in vane is

\[
\sigma_p = \frac{p_{in}^* - p_{out}^*}{\rho U_{mid}^2}
\]

As shown in Figure 10, the total pressure loss is maximum at the roots because of the clearance between impeller and shroud. For the better condition in bowed stage, the flow in vane has much lower pressure loss in the range of 10% ~ 40% of blade height than straight stages.

Definition of the static pressure coefficient is

\[
C_p = \frac{P_L - P_m}{P_m^* - P_m}
\]

Figures 11 showed the static pressure distribution along the different section (10%, 50%, 90% of blade height) of the impeller and vane blade. The area size reflect the blade loading. The lager adverse pressure gradient along the streamwise, the faster boundary layer develop, the more easily separated, which cause more wake momentum loss and pressure loss.

In the whole axial chord area, adverse pressure gradient in pressure side is lower than suction side, thus the boundary layer developed faster in suction. And in different section, the variation of static pressure is more significant, which indicated that the flow near the suction side affected the loading.
larger. The general rule of loading is maximum in 50% section, minimum in root. In bowed impeller, the loading is reduced, which can avoid the low energy fluid accumulation, high loss and separation.

Combined with figure 7, in bowed stage, the area of low velocity at vane root is reduced, which suggested the higher flow capacity and larger blade loading. Though the loading is the same in middle section and top section, the each pressure line is moved up, indicating the larger static pressure rise.

![Graphs showing pressure coefficient distribution](image)

**Figure 11.** Distribution of pressure coefficient along impeller and vane
5. Conclusion
Based on the numerical simulation of straight and bowed impeller stage of fans, the following conclusions can be obtained:

(1) The performance of stage can be improved obviously with bowed impeller. In the design condition, the efficiency can be increased by 1.44% and the static pressure rise is increased by 11%.

(2) The separation strength at root is weakened and the area of low velocity is reduced in bowed impeller, so that the low energy fluid cannot accumulate at the root suction of vane, reducing the loss.

(3) The bowed impeller can improved the flow at the root of the high-loaded fan, and lower the energy losses in the passages.

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