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Restructuring of an Air Classifier Rotor by Finite Element Analysis

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Abstract— The air classifier is an equipment that finds its major applications in mineral processing plants. One of its utility is to separate the finer particles from the coarse particles to provide the required size of particle output. Air classifiers use various principles of operation depending on their prerequisite and usage. Various running parameters of the air classifiers can be varied to obtain the desired output particle sizes which are governed by specific particle size distribution curves. The air classifier considered in the paper is based on a 'deflector wheel' principle. These deflector wheels are the rotors. During a trial run on a dynamic balancing machine the rotors failed structurally resulting in their permanent plastic deformation. This indicated a fault in the design; thus requiring failure analysis and restructuring of the same. This research paper points out an alternative to the current design keeping in mind the constraints of the system. The authors have employed the procedure of calculating the failure manually by an analytical approach. Later on, the design was examined and verified using a well-known Finite Element Analysis software followed by the development of conceptual designs and then choosing the most optimum one. The optimized design offered structural integrity to the rotor with minimal reduction in area, and hence the performance.

Keywords— Air Classifier, rotor, failure diagnosis,simulation, Finite Element analysis,Von-Mises stress, stiffener ring

1. INTRODUCTION

An air classifier is an industrial machine which sorts materials by a combination of size, shape, and density. It works by injecting the material stream to be sorted into a chamber which contains a column of rising air. Inside the separation chamber, air drag on the objects supplies an upward force which counteracts the force of gravity and lifts the material to be sorted up into the air. Due to the dependence of air drag on object size and shape, the objects in the moving air column are sorted vertically and can be separated in this manner. Air classifiers are commonly employed in industrial processes where a large volume of mixed materials with differing physical characteristics need to be sorted quickly and efficiently. [1]

(a) (b) (c)

Fig. 1. (a) Typical layout of mineral processing plant (b) fully assembled air classifier

(c) Air Classifier sectional view with particle flow.

Air Classifiers use various principles of operation depending on their need and use. Various running parameters are varied to obtain the desired output particle sizes. These outputs are governed by specific particle size distribution curves. The air classifier considered in the paper is based on a 'deflector wheel' principle. These deflector wheels are the rotors. They are responsible for the screening action that is required for particle separation. [5]Product fineness is controlled by the speed of the classifier wheel which is adjusted using a frequency controller. Classification in the range of 150 to 2.5 micrometers is possible. The material, after coming out from the outlet of the ball mill, is sucked upwards by an induced draught fan provided at the top of the system. This ground material flows through the classifier rotor. The size of the particle that can be allowed to pass through is governed by the speed of the rotor.

The range of particles that are obtained from the process mills vary over a range of few micrometers to a hundred micrometers. But, the output required varies from 6 to 20 micrometers. This can be obtained depending on the speed of the rotor. [4]

2. Component Failure

The rotor after being manufactured was tested on a dynamic balancing machine at 3000 rpm. While under test, it suddenly failed resulting in permanent plastic deformation of the blades. It was suspected that the centrifugal force acting on the blades was responsible for this failure. It pulled out the blades and the end plate under that action moved into the rotor as shown in figure 2(a). The horizontal length of the damaged component was found to be approximately 150 mm. The weld of few blades also broke.

(a) (b)

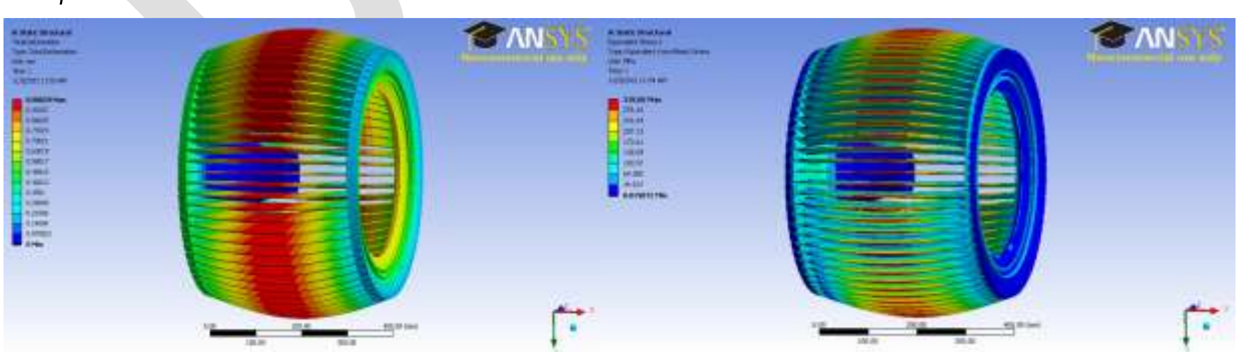
Fig. 2. (a) CAD model of existing component (b) Failure of Component

2.1. Failure Analysis

The reason for failure, as mentioned earlier was suspected to be the centrifugal force. The conceptapplied to show the failure by manual calculations was by treating one blade as a simply supported beam. From the results of manual calculations it was found that the stress generated in the rotor was 337.523 MPa.[2] The material used in the rotor is Plain carbon steel (Code: IS 2062), having a yield strength of 240 MPa. Hence, the stress generated due to the centrifugal force acting on the blades causes the plastic deformation which causes the failure. To verify the result, structural analysis of the rotor in FEA software ANSYS® 13.0 was performed.

3. SIMULATION

3.1. Air Classifier Rotor



(a) (b)

Fig. 3. (a) Static structural total deformation (b) Static structural equivalent stress developed in old rotor

The maximum deformation in static structural analysis of the air classifier rotor was observed to be 0.98 mm. The maximum occurs at the middle of the blades as shown in figure 3 (a).

The maximum von-mises stress was also observed to be at center of blade as shown in figure 3 (b) having magnitude of 310.68 MPa. The minimum safety factor of the design was found to be 0.7826. This clearly indicated an easy cause of failure in the rotor. The simulation resultsalso concurred with the manual calculations as well.

4. RESTRUCTURING OF THE ROTOR

4.1. Variables of the system

Following variables can be provided to the rotor without affecting the performance to a large extent.

- A slight reduction in the area of inlet of the rotor. But beyond a limit the performance of the air classifier as a whole can be affected.
- The material of construction of the vanes can be changed. However, as the vanes are welded onto the hubs, the weld between the two dissimilar metals should be strong enough to not give away during high speed rotation. Also, the cost, strength of the new material should be taken into consideration.
- Thickness of the blade can be marginally increased by a maximum of 1mm. Further increase in the thickness may have implications regarding weight of the rotor, which always effect the motor bearing as the entire rotor is mounted on motor without any other support. In addition to the above, higher thickness will provide lesser area for movement of material across the classifier. This results into performance issues of the classifier.

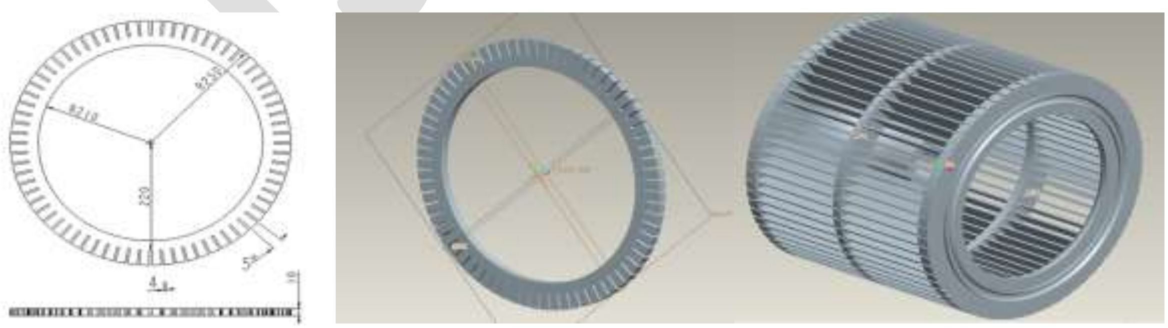
4.2. Restrictions of the system

- The system has been designed to give an output of different quantity based on different particle size requirement of the material.
- Reducing the length of the rotor reduces the rotor area, which also results into lower output. So, the change of rotor length was prohibited.
- Reducing diameter reduces the output of the system; an increase is not possible due to space restrictions. Hence, the diameter also cannot be changed.
- Higher weight will have effect on the motor bearing and the torque to be transmitted will get higher, which may result into higher motor size. Higher motor size will have mounting problems and the system will be very bulky.
- A change in the particle size distribution is undesirable as it would affect all the other parameters up & down the mineral processing plant.[3]

Keeping in mind the constraints and variables of the system, it was found desirable to come up with the following alternatives/improvements to the current design of the rotor:

- Provide a stiffening ring.
- Change in the material.
- Change in the dimensions of the blades.

Amongst the above mentioned alternatives the best option found was to provide a stiffening ring at the point where maximum bending moment is acting.



(a) (b) (c)

Fig. 4. (a) Stiffener specifications (b) CAD model of stiffener ring (c) CAD model of modified rotor

The material of make would be IS 2062 Grade B. The stiffener has to be in the shape of a circular annulus ring. Just like the hub and the end plate, there are 72 axis symmetric slots provided on its outer periphery where the blades can be welded.

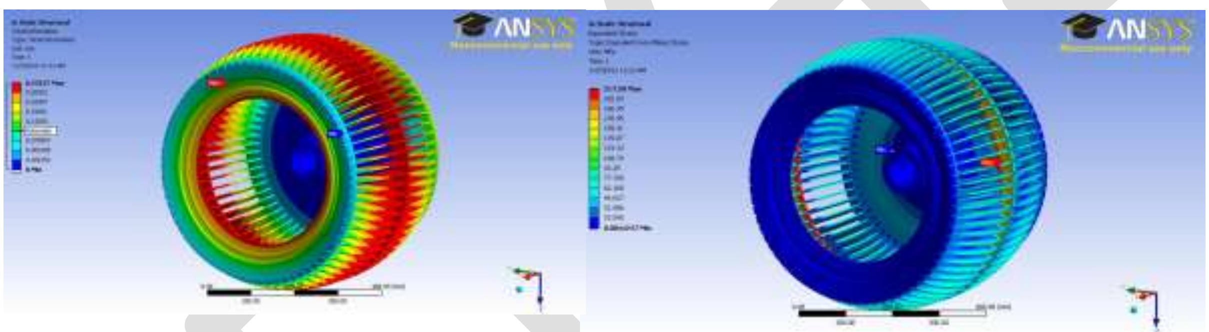
Table 1. Stiffener design specifications

UTS	TYS	CYS	Outer radius	Inner radius	Thickness
410 MPa	240 MPa	240 MPa	250 mm	210 mm	10 mm

The figure 4 (c) shows the modification that has been made to the original design of the rotor. As obtained in the failure calculations and ANSYS® results, the bending moment is found to be maximum at the center of the blade. Hence the stiffener ring has been placed exactly at the middle of the blade.

From analytical calculations, the maximum stress and deformation developed on the blades was found to be 80.145 MPa and 0.05113mm respectively. Although the stress on the blades is reduced, the stress on the stiffener might be higher. Hence, ANSYS analysis of the modified rotor design was done.

4.3. Analysis of Modified rotor

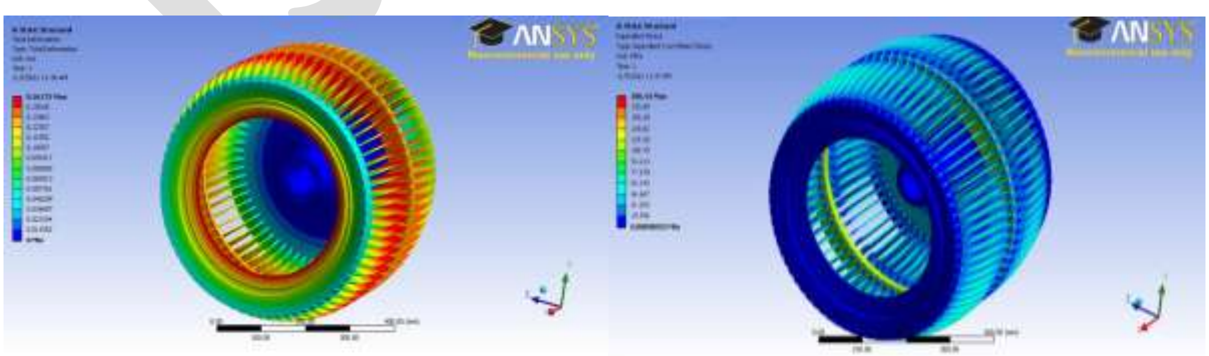


(a) (b)

Fig. 5. (a) Static structural total deformation (b) Static structural equivalent stress developed in modified rotor

The maximum deformation in ANSYS® static structural analysis of the modified rotor was observed to be 0.235 mm. The maximum von-mises stress in ANSYS® static structural analysis of the modified rotor was observed to be 217.58 MPa. The maximum stress was found to be on the stiffener ring and not on the blades. Hence,a drastic improvement in the model.
Upon performing transient analysis, the stress was still found to be higher than the yield strength of the material. Hence the new design, although more stable, needed to be optimized for a better safety factor and stability.

4.4. Analysis of Optimized rotor



(a) (b)

Fig. 6. (a) Static structural total deformation (b) Static structural equivalent stress developed in optimized rotor

On decreasing the rotor inner radius from 210 mm to 200 mm& increasing thickness form 10 mm to 15 mm, the maximum deformation was found to be reduced to 0.16 mm as shown in figure 6(a). The maximum stress of the optimized rotor was found to be 186.43 MPa, at the stiffener ring as shown in figure 6(b). The factor of safety of the optimized rotor was found to be a minimum of 1.341, which was acceptable, as 3000 rpm was the maximum operational speed of the rotor in the system. Hence there won't be a chance of an increase in stress.

5. CONCLUSION

While under test, the rotor of the air classifier suddenly failed resulting in permanent plastic deformation of the blades. It was suspected that the centrifugal force acting on the blades was responsible for this failure resulting in pulling out the blades and the end plate under that action moved into the rotor.The authors employed the procedure of calculating the failure manually by an analytical approach. Later on, the design was examined and verified using a well-known Finite Element Analysis software followed by the development of conceptual designs and then choosing the most optimum one. During static structural analysis of this rotor, the maximum deformation and von-mises stress were observed to be 0.98 mm and 310.68 MPa respectively occurring at the middle of the blades. The minimum safety factor being 0.7826.
Hence, in order to overcome this problem it was found desirable to redesign the rotor by providing a stiffening ring at the point where maximum bending moment was acting.The stress and deflection were brought down to 217.58 MPa and0.235 mm in the redesigned rotor. But, upon performing transient analysis, the stress wasstill higher than the yield strength of the material. Hence the new design, although more stable, needed to be optimized for a better safety factor and stability. On modifying the rotor, by decreasing its inner radius & increasing thickness, the stress and deflection were brought down to 186.4 MPa and0.16 mm respectively, in the optimized rotor. Due to a reduction of about 4.1667% in the area through which particles can pass through, the performance of the rotor was reduced slightly within permissible limits.

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