Powder Technology 292 (2016) 46–53

Contents lists available at ScienceDirect

Powder Technology

journal homepage: www.elsevier.com/locate/powtec

Design of a rotor cage with non-radial arc blades for turbo air classifiers

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ARTICLE INFO

Article history:
Received 4 September 2015
Received in revised form 11 January 2016
Accepted 16 January 2016
Available online 18 January 2016

Keyword:
Turbo air classifier
Rotor cage
Non-radial arc blade
Classification accuracy
Fine powder yield

ABSTRACT

As an important dynamic classifier, turbo air classifiers are widely used in various fields. To improve the classification performance of turbo air classifiers, a novel rotor cage with non-radial arc blades is designed by analyzing the influence of the rotor blade profile and the installed angle on the flow field in a turbo air classifier. Numerical simulations by ANSYS-FLUENT 14.5, as well as material classification experiments, are implemented to verify the new design. Simulation results indicate the significant improvement of flow field distribution in the rotor cage with non-radial arc blades. The incidence angle at the inlet of the rotor cage decreases significantly. Airflow streamlines match the profile of the non-radial arc rotor blade perfectly, and no air vortex is present in the channels of the rotor cage. The material classification experiment results demonstrate that the classification accuracy increases by 10.6%–40.8%, and the fine powder yield increases by 12.5%–40.1%, with an almost changeless cut size. The experimental results agree with the simulation results, thus verifying the feasibility of the modified rotor blades in practice. This design provides a theoretical guidance for the structure improvement of different types of classifiers with a rotor cage.

1. Introduction

Turbo air classifiers are mainstream dynamic classifiers that have simple structures and controllable product granularity. Turbo air classifiers are widely used in various fields, including minerals engineering, fine chemical industry and medicine [1–3]. Given the surge in the demand for ultrafine powder, the requirements of classification performance (e.g., small cut size, high classification accuracy and narrow particle–size distribution) have been increasing gradually. Considering that rotor cages directly affect the interior flow field of turbo air classifiers and classification performance, many researchers have focused on rotor cages. Gao [4] analyzed the effect of rotor cage rotary speed on classification accuracy by using FLUENT software and obtained a reasonable parameter combination for classification. Xing [5] measured and analyzed the vortex swirling between rotor blades by using Particle Image Velocimetry (PIV) technique. They found changes in the regulation of classification efficiency and cut size and obtained optimized operating parameters to obtain the minimum cut size. Ito [6] found that most of the pressure loss in a classifier is produced on the inside of the rotor. To minimize pressure loss and achieve a well-distributed velocity field, the inclined blade was designed.

To improve the flow field distribution in a rotor cage, some researchers made structure improvements [7–10]. Xu [7] designed backward crooked elbow rotor blades and simulated the particle tracks between the rotor blades. Huang [8] simulated and compared the flow field distribution in turbo air classifiers equipped with positively bowed, negatively bowed, and straight guide vanes. He concluded that the inertia rotating vortex and radial velocity fluctuations in rotor cages can be decreased by using positively bowed guide vanes.

Most previous of studies are limited to a given device or configuration and lack a systemic design method. In the present study, the influence of rotor blade configuration on flow field is analyzed thoroughly by referring to the theories of turbo-machinery. A rotor cage with non-radial arc blades is designed to realize a favorable flow field for classification. The rotor blade profile and installed angle are obtained by calculating the airflow velocity and analyzing flow field distribution. ANSYS-FLUENT 14.5 is used to simulate the flow field and compare the rotor cages with non-radial arc blades and straight blades. The experimental data are validated against simulation results to demonstrate that the rotor cage with non-radial arc blades can improve the classification performance of turbo air classifiers.

2. Design of the non-radial arc rotor blade

2.1. Classification principle of the turbo air classifier

The structural scheme of the turbo air classifier used in the present study is shown in Fig. 1-(a). The main geometric dimensions and symbols of the turbo air classifier are as follows: (1) the air inlet is 96 mm in height and 62 mm in width (labeled as a and b, respectively); (2) the outer and inner boundary radii of the rotor cage, R_1 and R_2 are 105 mm and 75 mm, respectively, with 24 blades radially installed and evenly distributed across the circumference of the rotor cage. The

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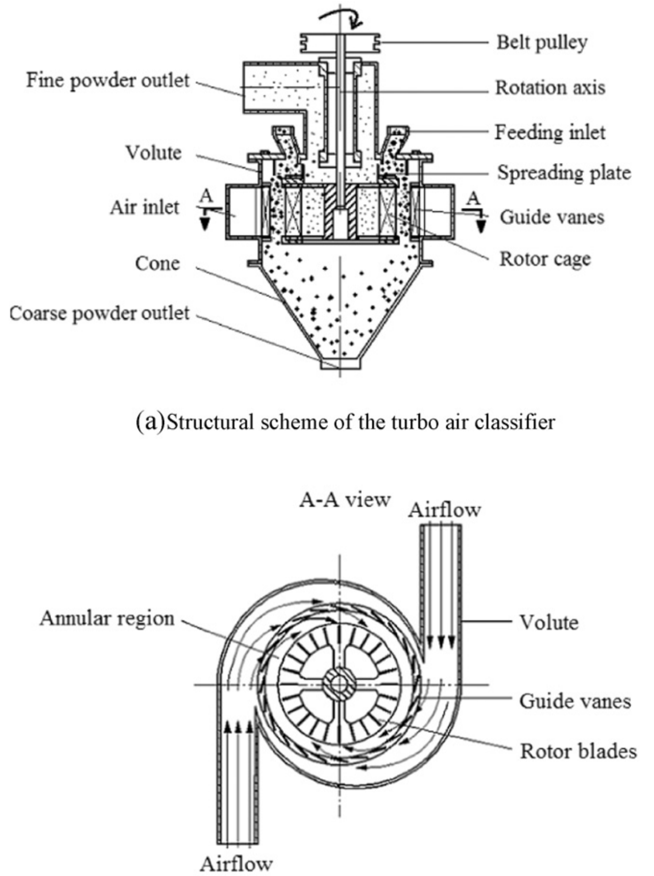
http://dx.doi.org/10.1016/j.powtec.2016.01.024
0032-5910/



(a)Structural scheme of the turbo air classifier

(b) Vertical view of the turbo air classifier and sketch of airflow track

Fig. 1. Schematic of the turbo air classifier.

dimensions of these blades are 30 mm in length, 2 mm in thickness (δ) and 96 mm in height; (3) 24 guide vanes with the same structural dimensions are distributed uniformly along the circumference of a circle with a radius of 136 mm and oriented at an angle of 15° from the tangent at the point on the circle coincident with the vane's centroid.

The rotor cage rotates clockwise and is driven by a belt pulley through the connection of a rotation axis. Under central negative pressure, the airflow enters from two symmetrical air inlets into an annular region through the guide vanes, shown in Fig. 1-(b). The annular region is a cylindrical space between the outer boundary of the rotor cage and the inner boundary of the guide vanes. The powder to be classified enters through the feeding inlet and falls to the spreading plate, which is rapidly rotating together with the rotor cage. The particles are thrown outward into the annular region. Among the many forces acting on the particles in this annular region, the air drag force, centrifugal force and gravity dominate the behavior of the particles [11]. Subject to these forces, small particles with little centrifugal force are drawn through the fine powder outlet, whereas coarse particles with large centrifugal force are thrown out, hit the guide vanes and fall down along the cone, where they eventually fall through the coarse powder outlet.

2.2. Dynamic analysis of a fluid element in the rotor cage

To improve the blade profile according to the flow field characteristics, a fluid element between two blades in the rotor cage is analyzed. On one hand, airflow rotates with the rotor cage. On the other hand, airflow moves inward under the force of the central negative pressure [12]. For a fluid element at any time, its absolute velocity **V** can be divided into relative velocity **W** and transport velocity **U** [13]. A local Cartesian coordinate system consisting of axis **x′** and axis **y′** is set for a fluid element, shown in Fig. 2. The axis **x′** is in the direction of the relative velocity **W**. Axis **y′** is perpendicular to axis **x′**.

The mechanics equilibrium equation in the direction of **y′** can be expressed as Eq. (1) on the basis of the principles of relative motion [14]:

$$P_1 - P_2 = \frac{dP}{dy'} = \rho\left(\omega^2 R_f \cos\beta + \frac{W^2}{R_c} + 2\omega W\right) \tag{1}$$

where,

dP/dy' is the differential incremental pressure along the axis **y′**, and is the pressure difference of P_1 and P_2 acting on the fluid element;
$\Omega^2 R_f \cos\beta$ is the transport acceleration, in which Ω is the rotor cage rotational speed, where β is the relative velocity angle between the relative velocity and the negative direction of transport velocity; R_f is the radius of circle in rotor cage where the fluid element is located;
W^2/R_c is the relative acceleration, where R_c is the curvature radius of streamline where the fluid element is located;
$2\Omega W$ is the Coriolis acceleration;
ρ is the density of air.

The differential form of composite Bernoulli equation in the direction of **y′** is expressed as follows:

$$\frac{dP}{dy'} = \rho\left(U\frac{\partial U}{\partial y'} - W\frac{\partial W}{\partial y'}\right). \tag{2}$$

By substituting $U = \Omega R_f$ and $\cos\beta = dR_f/dy'$ into Eqs. (1) and (2), the linear first-order differential equation of W is derived as follows:

$$\frac{\partial W}{\partial y'} = -2\omega - \frac{W}{R_c}. \tag{3}$$

Eq. (3) indicates that the distribution of relative velocity is only related to the curvature radius of streamline R_c and the rotor cage rotational speed Ω. When the operating parameters are fixed and Ω is constant, R_c becomes the only factor affecting the distribution of relative velocity. The discussions on R_c are made as follows:

1) For straight rotor blades, $R_c = \infty$, the Eq. (3) can be solved as follows:

$$W = -2\omega y' + C_1. \tag{4}$$

2) For the arc rotor blades, the Eq. (3) can be solved as follows:

$$W = e^{\frac{-y'}{R_c}}(C_2 \pm 2\omega R_c) - 2\omega R_c. \tag{5}$$

Using Taylor to expand e^{-y'/R_c}, the Eq. (5) can be substituted with Eq. (6).

$$W = \left(\pm\frac{C_2}{R_c} - 2\omega\right)y' + C_2 \tag{6}$$

where, C_1 and C_2 are positive constants.

Eq. (4) indicates that for the straight rotor blade, the speed variation rate of W is 2 Ω in the direction of axis **y′**. For the arc rotor blade with concave pressure side shown in Fig. 2, the sign ($\pm$) in Eq. (6) is positive (+), and the speed variation rate of W is $|-2\omega + C_2/R_c|$. However, for the arc rotor blade with convex pressure side the sign ($\pm$) in Eq. (6) is negative ($-$) and the speed variation rate of W is $|2\omega + C_2/R_c|$. The speed variation rate of W in the direction of axis **y′** is smallest when the