INTRODUCTION

The safe and efficient operation of ventilation systems is important for mining activities. The core components of a ventilation system are fans, including centrifugal, axial, and counter-rotating fans. Fans consume a significant amount of electricity in the system and need to work continuously and safely for a long time. Mine counter-rotating fans (MCRFs) are widely used for ventilation in mining and tunnel construction. An MCRF is composed of two rotors (R1 and R2) rotating in opposite directions. An MCRF has higher pressure rise and efficiency than a common two-stage rotating axial-flow fan under design conditions. However, the overall performance is compromised under off-design conditions. When the MCRF is connected to a ventilation pipe, the operating point of the system is the intersection between the pipe network resistance curve and the aerodynamic performance curve of the MCRF. The actual operating flow rate of the MCRF decreases with an increase in the resistance of the ventilation pipe network. In the early stages of operation, the resistance of the ventilation pipe network is low, and the actual operating flow rate is thus higher than the design flow rate, resulting in a sharp decrease in its pressure rise and efficiency. In the later stages of operation, the resistance of the ventilation pipe network is high, and the actual operating flow rate is thus lower than the

1 | INTRODUCTION

Variablespeed method for improving the performance of a mine counter-rotating fan

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Abstract
Existing methods of improving the performance of mine counter-rotating fans (MCRFs) through speed matching of the rotors have certain drawbacks. This study proposes a variable-speed method for the two rotors of an MCRF to improve its performance, including the blocking margin, operational safety, high-efficiency working range (HER), and stability margin. Experimental and numerical methods were used to study the pneumatic performance of the MCRF. The results show that the second rotor (R2) is the critical stage affecting the safety of the motor, HER, and blocking margin. A condition-adaptive speed matching method was proposed for R2 to avoid overloading the second motor at low-flow rates and to improve the blocking margin and HER. The first rotor (R1) is the critical stage determining the stability margin of the MCRF. Reducing \( n_1 \) and increasing \( n_2 \) at low-flow rates helped extend the stability margin.

KEYWORDS
blocking margin, mine counter-rotating fan, operation efficiency, stability margin, variable speed
design flow rate. With a further decrease in the flow rate, the second-stage motor is overloaded, and surging of the fan is observed. To explore these issues, the performance of the MCRF under off-design conditions should be studied.

Studies have shown that the performance of the MCRF under off-design conditions can be improved by adjusting the tip clearance of the two rotors. The total pressure ratio and efficiency decrease with increasing tip gap, and the rotor peak efficiency is linear with increasing tip clearance. The peak efficiency and stall margin are also influenced by the tip clearance of the two rotors. The intensified tip leakage flow in the front rotor (R1) due to the increased tip clearance leads to an increase in the incoming incidence angle near the tip of the rear rotor (R2). An increase in the double leakage flow range plays a significant role in the sensitivity of the efficiency to the tip clearance, and this range gradually extends with increasing tip clearance. Additionally, the fluctuating strength near the tip of R2 significantly increases. The axial spacing of two-stage contra-rotating blade rows has a significant impact on the aerodynamic performance and noise of the MCRF. With an increase in this axial spacing, the rotor-rotor interactions weaken, while the unsteady features of the two-stage rotor blades tend to be consistent. The performance of an MCRF can also be improved by using an unequal power design for the two rotors, adjusting the blade installation angle, or adjusting the turning and twisting angles. Optimizing the shape of the blades based on aerodynamic, structural, and aeroacoustic criteria could serve as a new multiobjective optimization technique for improving the performance. However, these methods are generally suitable for improving the aerodynamic performance of counter-rotating fans under specific conditions and are difficult to adapt to changing flow conditions in real time. In addition, an improper adjustment of the blade installation angle under variable working conditions may lead to a significant deterioration in the aerodynamic performance. Varying the rotational speed of the rotors to meet the demands of flow rate change is a convenient method to adjust the operating performance of the MCRF because this method does not require any change in the structure of the fan. Studies have shown that the power matching and efficiency of the two rotors vary when the speeds are varied. The stall performance and flow features of the rotors will also change depending on the speed matching of the two-stage rotors, and the stable operating range of the MCRF can be extended. It is necessary to develop a method to change the speed of the rotors quantitatively, so that the speed matching of the rotors can adapt to the demands of actual operation conditions automatically.

The rest of this study is organized as follows. Section 2 presents experimental and numerical methods for analyzing the performance of the MCRF. In Section 3.1, we discuss the causes of overload of the second motor under low-flow conditions and rapid reductions in the pressure and efficiency under high-flow conditions. A condition-adaptive speed matching (CASM) strategy is proposed for the second rotor to improve the blocking margin, high-efficiency working range (HER), and operation safety. The cause of instability in the flow and the critical stage affecting the stability margin of the MCRF are discussed in Section 3.2, and the influence of varying the speed of the two-stage rotors on the flow is discussed. Finally, we present a variable-speed method for improving the stability margin of the MCRF.

### Highlights

1. The second rotor is the critical stage affecting motor safety, high-efficiency working range, and blocking margin of the MCRF.
2. The first rotor R1 mainly determines the stability margin of the MCRF.
3. The MCRF performance is improved owing to condition-adaptive speed matching of R2.
4. Reducing \( n_1 \) and increasing \( n_2 \) can extend the stability margin.

### 2 | RESEARCH METHODS

#### 2.1 | Prototype fan and test rig

This study considered a small counter-rotating fan typically used for mine ventilation. The design pressure of this MCRF is 2800 Pa, and the design flow rate is 4.25 kg/s. Table 1 lists the parameters of the two rotors of this MCRF. A test rig for the performance testing of the MCRF was set up based on the C-type test device using standardized airways (ISO5801-1997). Figure 1 shows a diagram of the test device. Here, \( D_1 = 560 \text{ mm}, D_2 = 560 \text{ mm}, D_3 = 740 \text{ mm}, \) and \( D_4 = 542 \text{ mm}. \) The primary performance parameters were measured to evaluate the MCRF performance, including the differential pressure of the inlet bell mouth (\( \Delta p \)), inlet static pressure (\( p_2 \)), and input power of the two motors (\( N_{1m} \) and \( N_{2m} \)).

Figure 2 shows the MCRF used in the experimental test and its rotors. Each motor should be connected to a frequency converter to change the speed of the rotor. The rated power of the frequency converter is 11 kW, and the rated efficiency is 96%.

Figure 3 shows the efficiencies of the motor and frequency converter. The shaft power of the rotor, \( N \), is derived from the input power to the motor (\( N_{im} \)), motor efficiency (\( \eta_m \)), and efficiency of the frequency converter (\( \eta_f \)). The experimental efficiency of the MCRF is expressed as:
Here, $p_{out}$ is the total pressure at the outlet, $p_{in}$ is the total pressure at the inlet, $N$ is the shaft power (kW), and $q_v$ is the inlet volume flow (m$^3$/s).

$$n_{exp} = \frac{q_v (p_{out} - p_{in})}{\sum N} \times 100\% = \frac{q_v (p_{out} - p_{in})}{\sum N_m n_m \eta_f} \times 100\%. \quad (1)$$

Figure 4 shows the model and local grids of a single passage in the MCRF. The simulation model was divided into five parts: inlet domain, R1 domain, stage gap domain, R2 domain, and outlet domain. A conical diffuser was employed.
in the outlet domain. The R1 and R2 domains rotate about the z-axis, while the others are stationary. The full grid of the MCRF was generated by duplicating the single-passage grids of all the domains. Structured and local grid refinements were employed in the meshing of the single passage to improve the quality of the grids. The grids around the blade surface and wall are as shown in Figure 4.

The numerical investigation was based on the CFX software, and the fluid model was “air ideal gas.” It was assumed that the internal fluid was a compressible gas, heat transfer effects were considered, and gravitational effects were neglected. The renormalization group (RNG) two-equation k-epsilon turbulence model was employed for the steady-flow numerical calculations in rotating flows. Two types of frame change/mixing models are available in ANSYS CFX for steady-state simulations: Frozen rotor model and stage model (also known as the mixed plane method). Compared with the frozen rotor model, the stage model can obtain a higher calculation accuracy, especially for high-speed rotation, but requires more computational effort to converge. The MCRF used in this study is a low-speed fan, and the number of simulation grids on the machine is very high. A comparison between the numerical simulation results and the experimental results shows that the accuracy of the numerical simulations using the frozen rotor model meets the requirements. Therefore, considering the numerical calculation requirements and equipment conditions in this study, the frozen rotor model was applied to the mixing plane between the rotating and static domains. Wall boundary conditions were applied to the surfaces of the fans, and a standard wall function was utilized near the wall. The total pressure, flow direction, and static temperature were specified as the inlet boundary conditions, and the inlet flow was assumed to be uniform. The mass flow rate was specified as the outlet boundary condition. The residual target was set to $1 \times 10^{-6}$. The physical

![FIGURE 3](image-url) Efficiencies of the motor and frequency converter

![FIGURE 4](image-url) Simulation model and single-passage grids
timescale was derived from the speed of the rotor and was equal to $1/\omega$ ($\omega$: angular velocity).

Five group grids, as listed in Table 2, were used to carry out the grid independence verification. The grid independence verification results showed that the relative change rate of the outlet pressure is $<0.5\%$ when the total number of grids is greater than 6,296,992. Therefore, the fourth group of grids was selected for the numerical calculations.

The shaft power of the rotors, $N_1$ and $N_2$, can be expressed as.

$$N_i = T_i \omega \ (i = 1, 2).$$

Here, $T$ is the rotor shaft torque ($\text{N} \cdot \text{m}$), and $\omega$ is the angular velocity of the rotor ($\text{rad/s}$).

The total efficiency of the rotor, $\eta_1$ and $\eta_2$, can be expressed as.

$$\eta_i = \frac{q \cdot \rho \cdot \omega}{N_i} \ (i = 1, 2).$$

The total efficiency of the MCRF, $\eta$, can be expressed as.

$$\eta = \frac{q \cdot \rho \cdot \omega}{N_1 + N_2}.$$  

The flow coefficient of the MCRF, $\phi$, can be expressed as.

$$\phi = \frac{q \cdot \rho \cdot \omega}{\pi \cdot d^2 / 4}.$$  

Here, $d$ is the tip diameter ($\text{m}$), and $u$ is the circumferential velocity of the outer edge of the rotor ($\text{m/s}$).

The total pressure coefficient of the MCRF, $\psi$, can be expressed as.

$$\psi = \frac{p_1}{\frac{1}{2} \rho u^2}.$$  

### Table 2  Grid independence verification

| Group | Number of grids | Outlet Pressure (Pa) | Relative change rate (%) |
|-------|-----------------|----------------------|------------------------|
| 1     | 725,948         | 2888.93              |                        |
| 2     | 1,598,084       | 2801.06              | 3.04                   |
| 3     | 3,244,960       | 2780.89              | 0.72                   |
| 4     | 6,296,992       | 2768.61              | 0.442                  |
| 5     | 11,392,288      | 2766.32              | 0.083                  |

### Figure 5  Total pressure coefficient and total efficiency curves of the MCRF during 2930-2930 rpm operation
the second motor under low-flow conditions, and this effect restricts the safety of the MCRF.

The performance of R2 under off-design conditions reduces the safety, HER, and blocking margin of the MCRF. In short, the second rotor is the critical stage affecting the safety of the motor, HER, and blocking margin. In view of the performance stability of the two rotors under off-design conditions, the performance of R2 could be improved by reducing the speed under low-flow conditions and increasing the speed under high-flow conditions. To this end, a CASM for R2 was proposed, whereby the speed of R2, \( n_2 \), is adjusted in real time until the power of R2 is equal to that of R1 when the operating conditions change. Thus, the performance of R2 could be improved to meet the requirements of different working conditions.

Figure 7 shows the real-time adjustment method and process for the speed control of R2. The input power of the two rotors was measured using power sensors in real time and was inputted to a computer for analysis. When the power difference between R1 and R2 (\( \Delta N \)) was greater than a limit error (\( \Delta N > \varepsilon \)), an instruction to increase the frequency was inputted to the frequency converter to increase \( n_2 \). When \( \Delta N < -\varepsilon \), an instruction to decrease the frequency was inputted to the frequency converter to decrease \( n_2 \). When \( -\varepsilon < \Delta N < \varepsilon \), the frequency remained constant.

Figure 8 shows the total pressure coefficient and total efficiency curves of the MCRF, R1, and R2 determined by CFD calculation. The CASM operation improved the pressure rise and efficiency of R2 under high-flow rate conditions. Thus, the efficiency of the MCRF under high-flow rate conditions was increased. The pressure rise of R2 decreased under low-flow rates during the CASM operation, and this effect helped reduce the load on the second-stage motor and avoid the overloading problem.

The HER of the MCRF, where the efficiency is higher than 90% of \( \eta_{\text{des}} \) (design efficiency), could be extended during the CASM operation. The HER of the MCRF can be calculated using Equation 7:

\[
\text{HER} = \frac{\varphi_{0.9\eta_{\text{des}}} - \varphi_{\text{UP}}}{\varphi_{\text{des}}} \times 100\%.
\]  

(7)

Here, \( \varphi_{0.9\eta_{\text{des}}} \) is the flow coefficient under the condition where the efficiency is equal to 0.9\( \eta_{\text{des}} \), \( \varphi_{\text{UP}} \) is the flow coefficient under the unstable point (UP) condition, and \( \varphi_{\text{des}} \) is the design flow coefficient.

Figure 9 shows a comparison of the HERs based on the efficiency curves obtained by CFD. According to Equation 7 and Figure 9, the HER of the MCRF during 2930-2930 rpm operation is equal to 32.57%; it is increased to 37.37% through the CASM operation. The CASM operation increased the HER of the MCRF by 4.8%.

The blocking margin of the MCRF could be extended during the CASM operation. The blocking margin, BM, can be calculated using Equation 8:

\[
\text{BM} = \frac{\varphi_B - \varphi_{\text{des}}}{\varphi_{\text{des}}} \times 100\%.
\]  

(8)

**FIGURE 6** Performance curves of the two rotors during 2930-2930 rpm operation

**FIGURE 7** Real-time adjustment method and process for the speed control of R2
Here, $\phi_B$ is the flow coefficient under the blocking condition where the total pressure of the MCRF can only be used to overcome the flow loss and the flow of the MCRF reaches the maximum value.

Figure 10 shows a comparison of the blocking margins based on the experimental pressure coefficient curves. According to Equation 8 and Figure 10, the blocking margin of the MCRF during 2930-2930 rpm operation is equal to 32.8%; it is increased to 44.44% through the CASM operation. The CASM operation increased the blocking margin of the MCRF by 11.64%.

### 3.2 Influence of variable speed of rotors on stability margin

In this section, the steady-state simulation is presented. This simulation can be used to obtain the flow characteristics only when the flow reaches a steady state; the generation and development process of the flow separation cannot be reflected. With a decrease in the flow rate, when the pressure rise begins to decrease, the MCRF is said to have entered an unstable state. The cause of instability in the flow and the critical stage affecting the stability margin of the MCRF are discussed. The performance of the MCRF under different speed combinations is studied to find ways of improving the stability margin of the MCRF.

The working conditions of the MCRF can be divided into two parts: stable condition (normal working area) and unstable condition (saddle area). The demarcation point between the stable and unstable conditions is called the UP, at which the total pressure of the MCRF begins to decrease as the mass...
**FIGURE 11**  Mach number distribution at 50% span for 2930-2930 rpm operation

**FIGURE 12**  Mach number distribution at 5% span for 2930-2930 rpm operation
flow decreases. The stability margin of the MCRF, SM, is calculated using Equation 9:

$$SM = \frac{\phi_{\text{des}} - \phi_{\text{UP}}}{\phi_{\text{des}}} \times 100\%.$$  \hspace{1cm} (9)

Here, SM is the stability margin, and $\phi_{\text{UP}}$ is the flow coefficient under the UP condition.

The relative Mach number in the blade-to-blade planes of the MCRF during 2930-2930 rpm operation was used to explore the cause of instability in the flow and the critical stage affecting the stability margin of the MCRF. Figure 11 shows the Mach number distribution at 50% span of the rotors. The contours show that the flow around the median diameter is maintained well, and the flow coefficient decreases to $\phi_{\text{UP}}$. Even under the UP condition, there is no flow separation around the blades of R1 and R2 at 50% span. Figure 12 shows the Mach number distribution at 5% span of the rotors. Under low-flow conditions, there is a narrow range of flow separation on the suction surface near the trailing edge of the R1 blade root, but a wider range of flow separation on the suction surface near the trailing edge of the R2 blade root. The flow separation of the R2 blades increases with a decrease in the flow coefficient. Figure 12 also shows that when the flow coefficient decreases to 0.16 (far away from the UP), the flow separation appears on the suction surface near the trailing edge of the R2 blade root. Therefore, the flow separation near the R2 blade root does not directly lead to flow instability.

Figure 13 shows the Mach number distribution at 95% span of the rotors. Under stable working conditions and even near the UP, there is no obvious flow separation on the suction surface near the trailing edge of the R1 blade tip. However, when the flow coefficient decreases to 0.152 (UP), a wide range of flow separation appears on the suction surface near the trailing edge of the R1 blade tip. It subsequently spreads to the pressure surface near the leading edge of the next blade and fills the entire cascade passage (as indicated by the red ellipse in Figure 13). This process directly led to flow instability. The serious flow deterioration on the suction surface...
near the trailing edge of the R1 blade tip is the critical reason for the instability of the MCRF. The critical stage affecting the stability margin of the MCRF is the first rotor (R1).

Figure 14 shows the velocity triangles of the rotors. Here, \( u_1 \) and \( u_2 \) are the median diameter circumferential speeds of R1 and R2, respectively, where \( u = 2\pi n r_m \). For R1, \( c_{11} \) and \( c_{12} \) are the absolute velocities of the inlet and outlet, respectively; \( w_{11} \) and \( w_{12} \) are the relative velocities of the inlet and outlet, respectively; \( \beta_{11} \) and \( \beta_{12} \) are the relative angles of the inlet and outlet, respectively; and \( \Delta c_{1u} \) is the twisting velocity. For R2, the above parameters are written by replacing the first subscript 1 with 2. The parameters under low-flow conditions are distinguished by the superscript “\(^{\prime}\)”. Because the first rotor is the critical stage affecting the stability margin of the MCRF, the inlet relative angle of R1, \( \beta_{11} \), decreases with decreasing flow rate, leading to flow separation on the suction surface near the trailing edge of the R1 blade tip. A decrease in \( u_1 \) under low-flow conditions can increase \( \beta_{11} \) and is expected to improve the flow separation in R1. Additionally, to ensure the pressure capacity of the machine, \( u_1 \) could be increased appropriately. Based on this idea, we carried out the following research.

The performance of the MCRF in the speed of 2780-3080 and 3080-2780 rpm was studied (the average speeds of the two rotors remain unchanged). Figure 15 shows the stability margin of the MCRF while operating in the speed of 2780-3080 rpm. The experimental and CFD results show that the stability margin in the speed of 2780-3080 rpm is greater than that in the speed of 2930-2930 rpm. The CFD results show that the flow coefficient of the UP decreases from 0.1524 to 0.10.
0.145 when the speed combination of the MCRF is changed from 2930-2930 rpm to 2780-3080 rpm, which increases the stability margin from 12.91% to 17.14%.

Figure 16 shows the stability margin of the MCRF in the speed of 3080-2780 rpm. The experimental and CFD results show that the stability margin in the speed of 3080-2780 rpm is lower than that in the speed of 2930-2930 rpm. The CFD results show that the flow coefficient of the UP increases from 0.1524 to 0.1598 when the speed combination of the MCRF is changed from 2930-2930 to 3080-2780 rpm, which decreased the stability margin from 12.91% to 8.69%.

Figure 17 shows a comparison of the Mach number distributions at $\phi = 0.1524$ while operating in the speed of 2930-2930 and 2780-3080 rpm. In the latter case, the flow around the blade roots of R1 and R2 is slightly improved, whereas the flow around the blade tip of R1 is remarkably improved. The flow separation on the suction surface near the trailing edge of the R1 blade tip was eliminated.

Figure 18 shows that when the MCRF is operating at 2930-2930 rpm, a significant flow separation occurs in the trailing edge of the blade tip of R1 when the flow coefficient is reduced to 0.1524, causing the fan to enter an unstable state. When the MCRF is operating in the speed of 3080-2780 rpm, the flow separation occurs in the trailing edge of the blade tip of R1 when the flow coefficient is reduced to 0.1598, causing the fan to enter an unstable state earlier and reducing the stability margin. However, when the MCRF is operating in the speed of 2780-3080 rpm, the flow separation occurs in the trailing edge of the blade tip of R1 when the flow coefficient is reduced to 0.145. Therefore, the MCRF enters the unstable state much later, and the stability margin is increased. In summary, operating in the speed of 2780-3080 rpm helped improve the flow near the blade tip of R1 under low-flow rates, and the occurrence of unstable condition was delayed from $\phi = 0.1524$ to $\phi = 0.145$. Conversely, operating in the speed of 3080-2780 rpm deteriorated the flow near the blade tip of R1 under low-flow rates. The flow coefficient of the unstable condition increased from $\phi = 0.1524$ to $\phi = 0.1598$.

The unstable working lines (UWLs) of the MCRF under the three speed combination operating modes (including operations when $n_1 < n_2$, $n_1 = n_2$, and $n_1 > n_2$) were analyzed. Figure 19 shows the CFD total pressure curves of the MCRF with the three speed combinations. The UWLs of the MCRF when $n_2 - n_1 = 150$ rpm, $n_1 = n_2$, and $n_1 - n_2 = 150$ rpm are indicated by the red dotted line, black dotted line, and blue dotted line, respectively. The results show that compared with the
conventional operating mode \((n_1 = n_2)\), operating with \(n_1 < n_2\) shifts the position of the UWL leftward and extends the stability margin. In contrast, operating with \(n_1 > n_2\) shifts the position of the UWL rightward and decreases the stability margin.

Figure 20 shows the experimental total pressure curves of the MCRF with the three speed combinations. The conclusions regarding the UWLs from the experimental results are consistent with those from the CFD calculations. The tested UWLs indicate that the stability margin with the speed combination of \(n_2 - n_1 = 150\) rpm is greater than that with the speed combination of \(n_1 = n_2\), and the stability margin with the speed combination of \(n_1 - n_2 = 150\) rpm is lower than that with the speed combination of \(n_1 = n_2\). In other words, reducing \(n_1\) and increasing \(n_2\) can increase the stability margin on the basis of ensuring pressure rise.

## 4. CONCLUSIONS

This study proposes a variable-speed method for the two rotors of an MCRF. The method helped improve the performance of the MCRF, including the stability margin, operational safety, HER, and blocking margin. The following conclusions can be drawn from the study:

The second rotor \(R_2\) is the critical stage affecting the safety of the motor, HER, and blocking margin of the MCRF. The pressure coefficient and efficiency of \(R_2\) decrease rapidly under high-flow conditions, thus reducing the total efficiency and pressure rise of the MCRF and restricting the HER and blocking margin. The shaft power of \(R_2\) increases rapidly under low-flow conditions, thereby overloading the second motor.
The first rotor R1 is the critical stage that determines the stability margin. The inlet relative angle of R1 decreases with decreasing flow rate, leading to flow separation on the suction surface near the trailing edge of the R1 blade tip. This serious flow deterioration is the main reason for the instability of the MCRF. When a wide range of flow separation appears on the suction surface near the trailing edge of the R1 blade tip with a decrease in the mass flow under low-flow conditions, the MCRF will operate in an unstable condition.

A CASM was proposed for R2 to overcome the overload- ing of the second motor under low-flow conditions and to improve the HER and blocking margin. Through this method, the speed of R2 is adjusted in real time until the power of R2 is equal to that of R1 when the operating conditions change. The CASM could help reduce the load on the second-stage motor and avoid the overloading problem under low-flow conditions. It also effectively improved the HER and blocking margin.

When the speed of R1 is reduced under low-flow conditions, the positive attack angle of the inlet of R1 decreases, and the cascade flow near the blade tip of R1 improves. Thus, under low-flow rates, operating with decreased $n_1$ and increased $n_2$ can shift the position of the UWL leftward and extends the stability margin on the basis of ensuring pressure rise.

Considering that a higher simulation accuracy can be obtained using the “stage” model, this model will be adopted in the next research work, and the simulation results will be compared with those obtained in this study.

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**NOMENCLATURE**

**Notation**

- R1: The first rotor, Rotor 1
- R2: The second rotor, Rotor 2
- MCRF: Mine counter-rotating fan
- CFD: Computational fluid dynamics
- $\omega$: Angular velocity
- $\Delta p$: Differential pressure of imported current collector
- $p_3$: Inlet static pressure
- $N_{in}$: Input power of motor
- $\eta_m$: Motor efficiency
- $\eta_f$: Frequency converter efficiency
- $N$: Shaft power of rotor
- $T$: Rotor shaft torque
- $\phi$: Flow coefficient
- $\psi$: Total pressure coefficient
- $\rho$: Air density
- $u$: Median diameter circumferential speed of rotor
- $n$: Speed of rotor
- $c$: Absolute velocity
- $w$: Relative velocity
- $\beta$: Relative angle
- HER: High-efficiency working range
- UP: Unstable point
- BM: Blocking margin
- SM: Stability margin
- UWL: Unstable working line

**Subscript and superscript**

- 1st 1: The first rotor
- 1st 2: The second rotor

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**FIGURE 19** Unstable working lines obtained by CFD. CFD, computational fluid dynamics

**FIGURE 20** UWLs obtained by experiments. UWL, unstable working lines
In the speed matching operation

Under non-design conditions

In the speed matching operation

REFERENCES

1. Yueze L, Akhtar S, Sasmito AP, Kurnia JC. Prediction of air flow, methane, and coal dust dispersion in a room and pillar mining face. *Int J Min Sci Technol*. 2017;27(4):657-662.
2. Rezaeian M, Montazeri H, Loonen R. Science foresight using life-cycle analysis, text mining and clustering: a case study on natural ventilation. *Technol Forecast Soc Change*. 2017;118:270-280.
3. Geng F, Luo G, Wang Y, et al. Dust dispersion in a coal roadway driven by a hybrid ventilation system: a numerical study. *Process Saf Environ Protect*. 2018;113:388-400.
4. Kursunoglu N, Onder M. Selection of an appropriate fan for an underground coal mine using the analytic hierarchy process. *Tunn Undergr Sp Tech*. 2015;48:101-109.
5. Odyjas P, Moczko P, Zawiślak M, et al. Investigation of influence of unevenly spaced blades onto working parameters of centrifugal fans impellers used in mine ventilation. *1st Renewable Energy Sources-Research and Business (RESRB-2016), June 22–24 2016, Wroclaw, Poland*. Cham: Springer; 2016:389-398.
6. Radionov A. Magnetic fluid sealing complexes for bearing assemblies of mine main ventilation fans. *Magnetohydrodynamics* (0024-998X). 2018;54:109-114.
7. Nouri H, Ravelet F, Bakir F, et al. Design and experimental validation of a ducted counter-rotating axial-flow fans system. *J Fluids Eng*. 2012;134(10):104504.
8. Lesley EP. Experiments with a counter-propeller. National Advisory Committee for Aeronautics, Technical Report No. 453, 1933.
9. Mistry C, Pradeep AM. Influence of circumferential inflow distortion on the performance of a low speed, high aspect ratio contra-rotating axial fan. *J Turbomach*. 2014;136(7):071009.
10. Shigemitsu T, Takeshima Y, Oyama K, et al. Performance and internal flow of contra-rotating small-sized cooling fan. *Open J Fluid Dyn*. 2018;8(2):181.
11. Toke TD, Pradeep AM. Experimental investigation of stall inception of a low speed contra rotating axial flow fan under circumferential distorted flow condition. *Aerosp Sci Technol*. 2017;70:534-548.
12. Mao X, Liu B, Zhao H. Effects of tip clearance size on the unsteady flow behaviors and performance in a counter-rotating axial flow compressor. *Proc Inst Mech Eng G-J Aerosp Eng*. 2019;233(3):1059-1070.
13. Luan H, Weng L, Luan Y, et al. Numerical study on aerodynamic noise performances of axial spacing in a contra-rotating axial fan. *J Vibroeng*. 2016;18(8):5605-5618.
14. Mistry C, Pradeep AM. Effect of variation in axial spacing and rotor speed combinations on the performance of a high aspect ratio contra-rotating axial fan stage. *Proc Inst Mech Eng A-J Power Energy*. 2013;227(2):138-146.
15. Luan H, Weng L, Luan Y. Numerical simulation of unsteady aerodynamic interactions of contra-rotating axial fan. *PLoS ONE*. 2018;13(7):e0200510.
16. Luan H, Weng L, Liu R, et al. Axial spacing effects on rotor-rotor interaction noise and vibration in a contra-rotating fan. *Int J Aerosp Eng*. 2019;2019:1–15.
17. Sun WL, Fang XJ. Influence of blade structure optimization on the aerodynamic characteristics for counter-rotating fan. *J Eng Thermophys*. 2018;6:12.
18. Grasso G, Moreau S, Christophe J, et al. Multi-disciplinary optimization of a contra-rotating fan. *Int J Aeroacoust*. 2018;17(6-8):655-686.
19. Sun X, Meng D, Liu B, Wang Q. Numerical investigation of differential speed operation of two impellers of contra-rotating axial-flow fan. *Adv Mech Eng*. 2017;9(10):168781401772008.
20. Meng DW, Sun X, Zhang Z. Study on deformation of the two rotors of contra-rotating fan under variable speeds. *2015 2nd International Conference on Machinery, Materials Engineering, Chemical Engineering and Biotechnology*. Paris: Atlantis Press; 2015.
21. Yakhot V, Orszag SA. Renormalization group analysis of turbulence. I. Basic theory. *J Sci Comput*. 1986;1(1):3-51.