

Article

Vibration and Aerodynamic Analysis and Optimization Design of a Test Centrifuge

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Abstract: Taking a type of test centrifuge as the research object, the finite element model of the test centrifuge was established, the vibration characteristics and aerodynamic performance of the test centrifuge were analyzed, and a structural optimization design of the test centrifuge was carried out. In this paper, the load was applied according to the actual working condition of a type of test centrifuge. The vibration of the mounting seat of the test centrifuge was analyzed, and the structure of the mounting seat was improved. After improvement, the vibration of the mounting seat was 77.38% lower than that of the original mounting seat. Then, the aerodynamic analysis of the test centrifuge was carried out. The analysis results show that the test centrifuge moved more smoothly under the whole-package shell and the fairing, the resistance decreased, and the shaft load decreased. Finally, the fairing of the test centrifuge was optimized. The analysis shows that an increase in the width of the fairing can reduce the resistance coefficient, which is helpful to the stability of the test centrifuge during operation and reduces the unbalanced response of the system caused by air resistance.

Keywords: test centrifuge; numerical simulation; vibration analysis; pneumatic analysis; structural optimization design



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1. Introduction

It is very important for equipment and people to be in a high-gravity environment in ultra-high-altitude flight. At present, centrifuges are commonly used for scientific experiments to simulate high-gravity environments [1]. Centrifuges for scientific tests include geotechnical centrifuges [2], manned centrifuges [3], and precision test centrifuges. Centrifuges for scientific tests are a type of equipment that uses a high rotational speed rotating arm to generate a centrifugal force more than several times or even more than ten times that of gravity to simulate the high-gravity environment to which equipment or people are subjected [4,5]. Precision test centrifuges are usually high-precision inertial navigation test equipment for inertial device verification, calibration, and testing [6]. The precision test centrifuge equipment provides acceleration by rotating the central shaft. The greater the rotation radius, the greater the acceleration provided. Therefore, a pod is generally set at the end of the equipment to place various test equipment [7,8].

Significant research has been carried out on precision test centrifuges. Ling et al. [9] established three types of acceleration measurement uncertainty evaluation models, and the acceleration measurement uncertainty of precision centrifuge is calculated and verified. Wang et al. [10] carried out a study on the influence of structural errors of precision test centrifuges on the motion accuracy of the equipment itself and obtained the variation law between the structural error parameters and equipment motion parameters. Cheng et al. [11] mainly studied the deformation of the turntable and the rotating arm during the operation of the precision test centrifuge. The results showed that the increase in centrifugal load and temperature led to an increase in the deformation of the turntable and the rotating arm. Ren et al. [12] proposed a new method for the calibration of high-precision

centrifuge accelerometers in order to further improve the calibration accuracy of the higher-order error model coefficients, and the calibration accuracy was greatly improved after applying the method. Liu et al. [13] studied the effect of temperature on the uncertainty of the static radius measurement of a precision test centrifuge. Huang et al. [14] gave the calculation formula of the wind resistance power during the operation of the centrifuge, which provided a theoretical basis for the engineering design of the centrifuge.

Most of the current research is directed toward the accuracy of centrifuge motion; however, the stability problem during the operation of the precision test centrifuge is very critical [15,16]. In order to improve the design efficiency of the centrifuge, the structure of the centrifuge was analyzed and optimized using CFD. In this paper, the numerical simulation analysis of the vibration response and aerodynamic performance of the centrifuge structure was carried out for a type of test centrifuge. At present, the centrifuge is still under design. In order to ensure sufficient safety margin, the external vibration response and the aerodynamic characteristics of the equipment flow field at the installation seat of the test centrifuge were analyzed emphatically. The equipment structure was improved and optimized from the two aspects of vibration and aerodynamics [17–20]. After improvement and optimization, the vibration response of the key components and the aerodynamic resistance of the centrifuge are reduced, and the performance of the test centrifuge is improved.

2. Numerical Model of Test Centrifuge

The test centrifuge studied in this paper is shown in Figure 1. The test centrifuge consisted of a rotating arm, a rotating shaft, a pod, and a counterweight area. The counterweight area can be trimmed by placing blocks of different weights to reduce the dynamic unbalance of the centrifuge [21]. The centrifuge was a whole rotary arm centrifuge, and the test piece in the pod was heavy. The force condition of the test centrifuge is shown in Figure 1. It can be seen that the centrifuge system was affected by two centrifugal forces in the opposite direction. When the two forces were balanced, the load of the shaft could be reduced. The three-dimensional model of the test centrifuge was established, and the vibration response of the centrifuge was analyzed using the static simulation module. The static calculation model of the test centrifuge is shown in Figure 2.

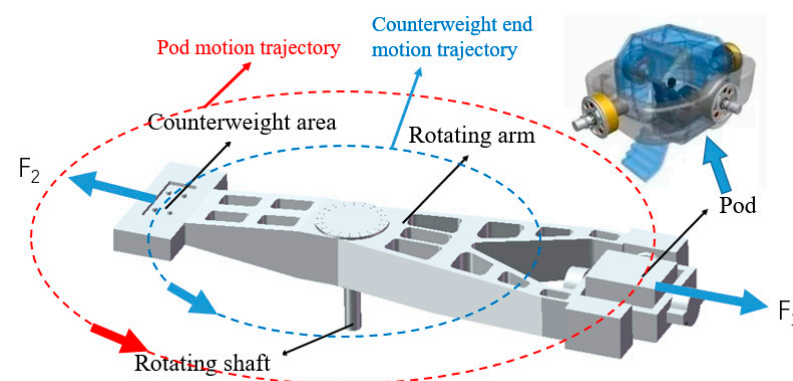


Figure 1. The structure and motion trajectory of the test centrifuge.

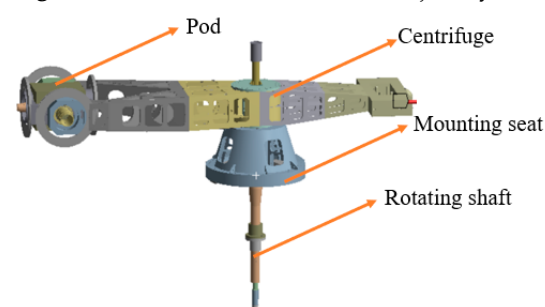


Figure 2. Statics calculation model of the test centrifuge.

3. Vibration Numerical Simulation and Structural Improvement Design of the Test Centrifuge

3.1. Vibration of the Test Centrifuge

During the operation of the test centrifuge, the vibration at the mounting seat comes from the imbalance of the whole machine, the external vibration at the pod, and the vibration at the bearing. The vibration problem of the centrifuge has a great influence on the performance of the centrifuge. It is necessary to control the vibration value of the mounting seat within a reasonable range to minimize the vibration and provide the necessary technical basis for the good operation of the centrifuge. In order to ensure the safety and stability of the centrifuge, it is necessary to control the vibration at the mounting seat within the safety line.

There are many reasons for the imbalance of equipment in rotating machinery. The imbalance will make the rotating shaft bear excessive bending and torsion, which will lead to instability. In order to solve the above problems, it is necessary to identify and eliminate the imbalance of equipment [22–24]. The FEM model is shown in Figure 3, using a tetrahedral mesh. The number of elements is 1.44 million. The first load is the G40 dynamic unbalance.

$$m = 9549 \times \frac{MG}{rn} \quad (1)$$

$$F = mr\omega^2 \quad (2)$$

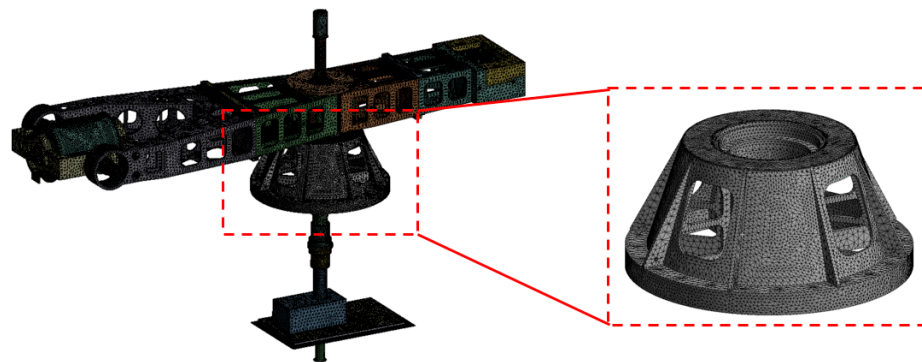


Figure 3. FEM model.

In the Equation [25], M is the rotating arm mass, G is the unbalanced precision grade, r is the correction radius, n is the working rotating speed of the centrifuge, and m is the unbalanced qualified quantity. The dynamic unbalance was simulated by applying force on the counterweight end face. According to Equation (1) [25], the dynamic unbalance G was 40, and the centrifuge speed was 38 r/min. It was calculated that the counterweight end should apply 200 kg of weight. According to Equation (2), it was calculated that a force of 19,077 N should be applied along the Y-axis direction at the counterweight end, as shown in Figure 4a. According to the spindle speed of 38 r/min, the harmonic response analysis frequency was 0~0.6333 Hz. The second load was the external vibration of the test piece in the pod, and its value was 1 g.

$$F = ma \quad (3)$$

The acceleration excitation was converted into force by Equation (3). According to the mass of the test piece in the external vibration pod, the calculated excitation size was 7840 N. Therefore, the force was applied in all directions of the test piece, as shown in Figure 4b. According to the test piece speed in 2400–17,000 r/min, the harmonic response analysis frequency was 40~290 Hz.

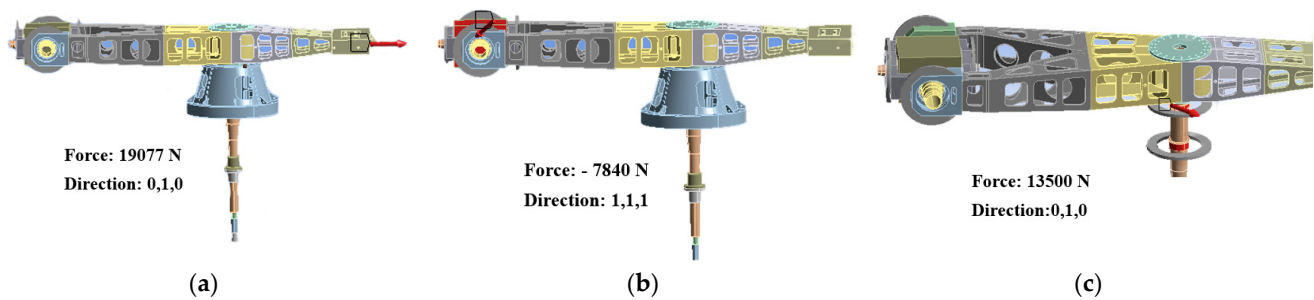


Figure 4. Vibration analysis results: (a) Unbalanced excitation load; (b) External vibration load; (c) Vibration load of bearing.

The last one is the bearing vibration. The excessive clearance of the bearing itself and the interaction between the bearings in the rotating machinery will cause an excessive vibration response of the equipment. To solve the problems, the appropriate bearing should be selected in the design [26–28]. In order to simulate the vibration excitation of the bearing, the excitation force generated by the residual unbalanced mass was applied at the upper and lower spindle bearings, that is, a force of 19,077 N along the angular bisector of the X and Y axes, as shown in Figure 4c. The bearing characteristic frequencies of radial bearing and thrust bearing were calculated, including the frequency f_{bpf_i} of the rolling element passing through the inner ring, the rotation frequency f_{bsf} of the rolling element, and selecting the maximum value as the harmonic response analysis frequency to simulate the bearing excitation. According to Equations (4)–(6), the frequency of the rolling part passing through the inner ring and the rotation frequency of the rolling part were calculated.

$$D = \frac{D_i + D_o}{2} \quad (4)$$

$$f_{bpf_i} = \left(1 + \frac{d}{D} \cos \alpha\right) \frac{Z f_i}{2} \quad (5)$$

$$f_{bsf} = \frac{f_i D}{2 d} \left(1 - \left(\frac{d}{D} \cos \alpha\right)^2\right) \quad (6)$$

In these equations, D is the bearing pitch diameter, D_i is the bearing inner ring raceway diameter, D_o is the bearing outer ring raceway diameter, d is the rolling element diameter, α is the rolling element contact angle, Z is the number of rollers, and f_i is the rotation frequency of the inner ring around the center of the circle. The radial bearing parameters and thrust bearing parameters are shown in Table 1. The frequency of the rolling element of the radial bearing through the inner ring was calculated to be 11.57 Hz by Equation (5), and the frequency of the rolling element of the radial bearing through the inner ring was calculated to be 12.74 Hz. The rolling element rotation frequency of the radial bearing was calculated to be 4.16 Hz by Equation (6), and the rolling element rotation frequency of the radial bearing was calculated to be 5.33 Hz.

Table 1. Bearing parameters.

Parameter	Radial Bearing	Thrust Bearing
D /mm	685	800
D_i /mm	770	1060
D_o /mm	727.5	930
d /mm	55	55
α /°	0	40
Z	34	38
f_i /Hz	0.633	0.633

3.2. Harmonic Response Analysis Results

The rotational speed of the test centrifuge was 38 r/min. Three excitations of unbalance, external vibration, and bearing vibration were applied in the harmonic response analysis. At present, the centrifuge is still under design. In order to ensure sufficient safety margin, the vibration response of the support seat was linearly superimposed to analyze the maximum vibration response of the support seat. The static analysis adopted harmonic response analysis. The mesh is generated by the mesh module. The constraint setting in the analysis corresponded to the actual situation. A total of 1 fixed constraint, nine spring connections, and eight bearing connections were applied. Radial bearing stiffness K_1 and thrust bearing stiffness K_2 were calculated according to Equation (7) [29].

$$K = \left[56 \times 10^{-7} \frac{1}{Zl\cos\beta} \left(l g \frac{7.6 \times 10^6 d_3 l z \cos\beta}{R} - l g e \right) \right]^{-1} \quad (7)$$

In this Equation, K is the bearing stiffness, Z is the number of rollers, β is the contact angle of the roller, l is the effective length of the roller, d_3 is the diameter of the roller, and R is the radial load.

The first one is the vibration response under unbalanced excitation. According to the spindle rotating speed, 0~0.6333 Hz was set to simulate the harmonic response analysis frequency of G40 dynamic unbalance. The force was 19,077 N along the Y axis. As shown in Figure 5a, the maximum vibration velocity response of the mounting seat under unbalanced excitation was 0.023 mm/s. The second one is the vibration response of the test piece in the pod under external vibration excitation. According to the rotational speed of the test piece, the harmonic response analysis frequency of the external vibration was set to 40~290 Hz. The force applied to the bottom of the test piece was 7840 N along the vertical direction, the radial direction of the centrifuge, and the tangential negative direction of the centrifuge. As shown in Figure 5b, the maximum vibration velocity response of the mounting seat was 2.4998 mm/s under the external vibration excitation. The last one is the vibration response of bearing under vibration. The harmonic response analysis frequency was 0~22 Hz to simulate the harmonic response analysis frequency of the bearing vibration. The force was set to 13,500 N along the X and Y axes. As shown in Figure 5c, the maximum vibration velocity response of the mounting seat under bearing vibration excitation was 5.2242 mm/s.

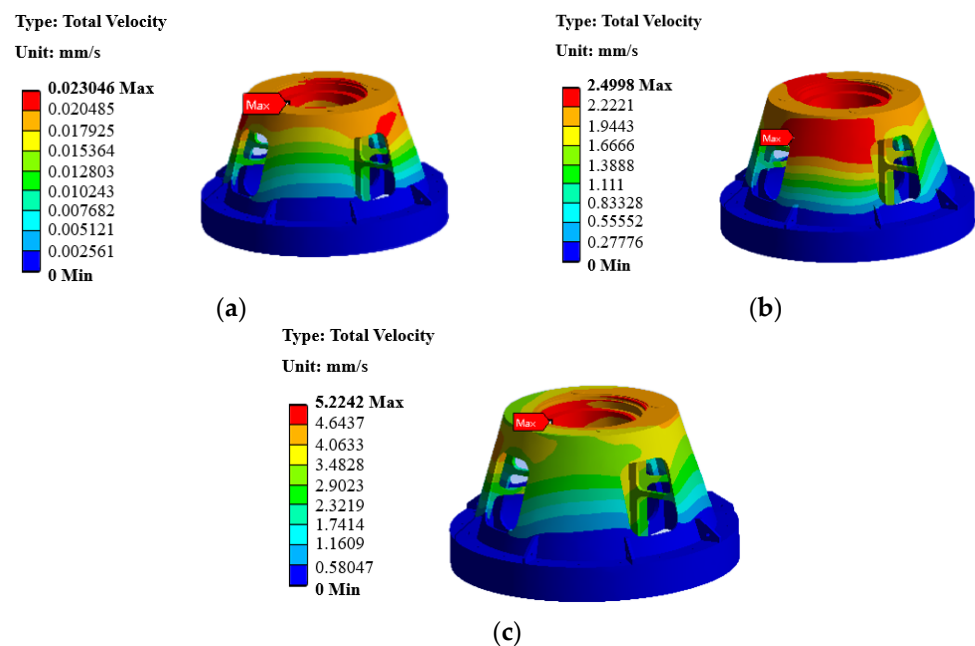


Figure 5. Vibration analysis results: (a) Vibration velocity response of the mounting seat under unbalanced excitation; (b) Vibration velocity response of mounting base under external vibration excitation; (c) Vibration velocity response of mounting seat under bearing vibration excitation.

According to the analysis results, the maximum vibration velocity response of the mounting seat was 7.747 mm/s under the combined action of external vibration excitation, unbalanced excitation, and bearing vibration excitation. The specific data are shown in Figure 6 and Table 2.

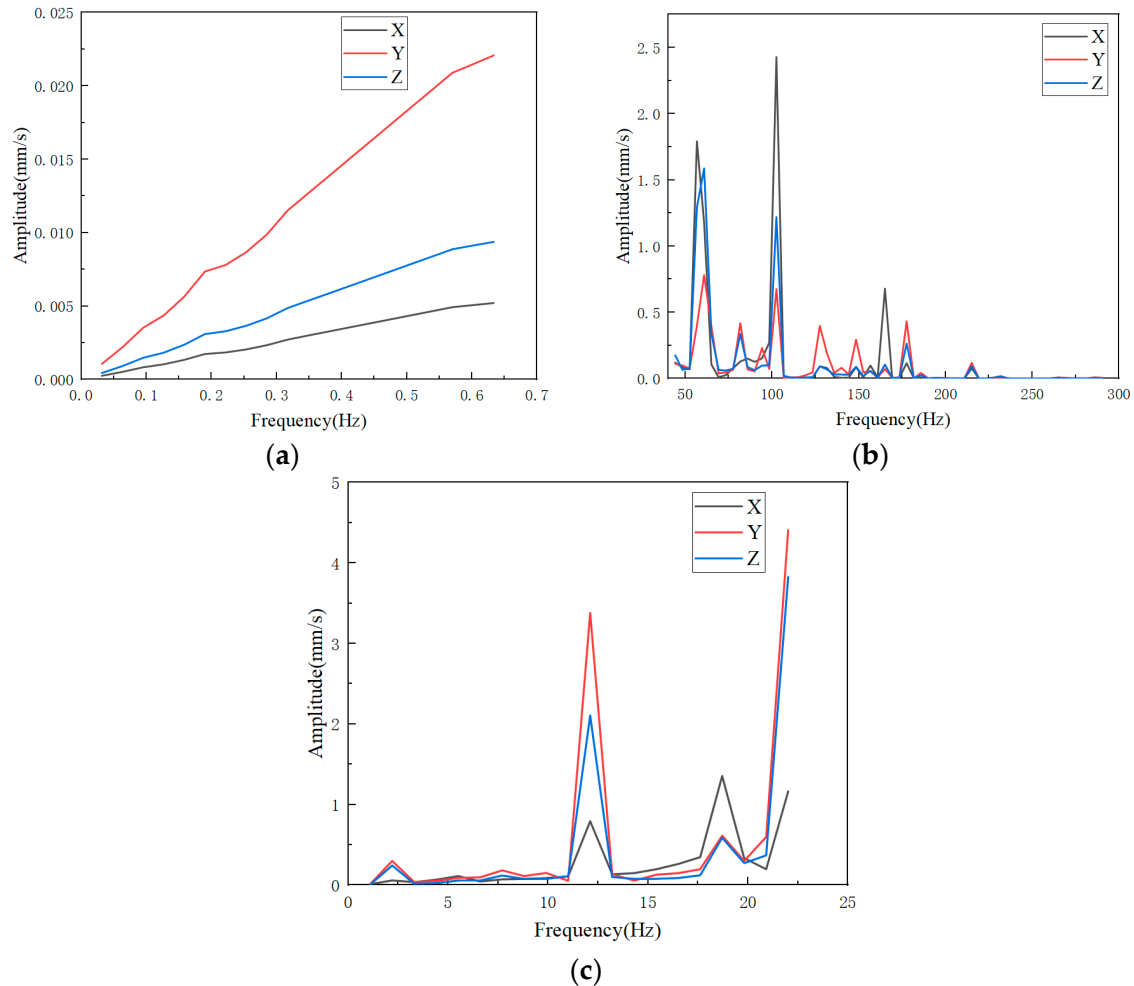


Figure 6. Vibration response spectrum: (a) Unbalanced excitation; (b) External vibration excitation; (c) Bearing vibration excitation.

Table 2. Harmonic response analysis results.

Incentive Type	Mounting Seat Vibration (mm/s)
Unbalanced excitation	2.4998
External vibration excitation	0.0230
Bearing vibration	5.2242
Total	7.747

3.3. Improvement Design of Mounting Seat

In order to reduce the vibration response of the experimental centrifuge mounting seat, the mounting seat model was optimized. The optimized model applied annular stiffeners at the top of the mounting seat and extended the original stiffeners. The wall thickness of the mounting seat was increased from 26 mm to 30 mm, as shown in Figure 7. The modified model was simulated again and compared with the original model.

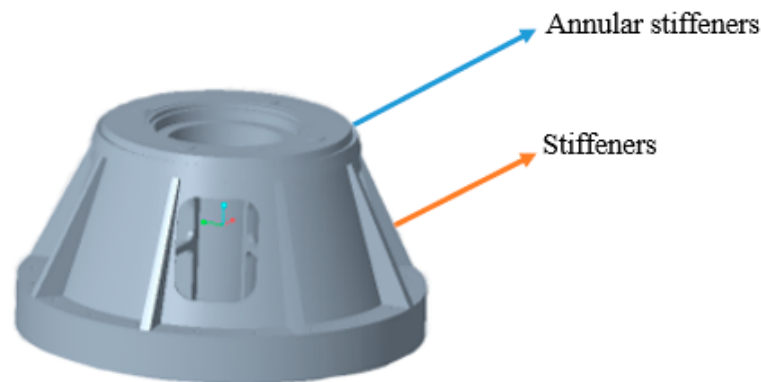


Figure 7. Improved mounting seat.

The load and constraint application were consistent with the model of the mounting seat before improvement. According to the analysis results, the maximum vibration velocity response of the improved mounting seat under the combined action of external vibration excitation, unbalanced excitation, and bearing vibration excitation was 1.754 mm/s, which was 77.36% lower than that before the improvement. The specific data are shown in Figures 8 and 9 and Table 3.

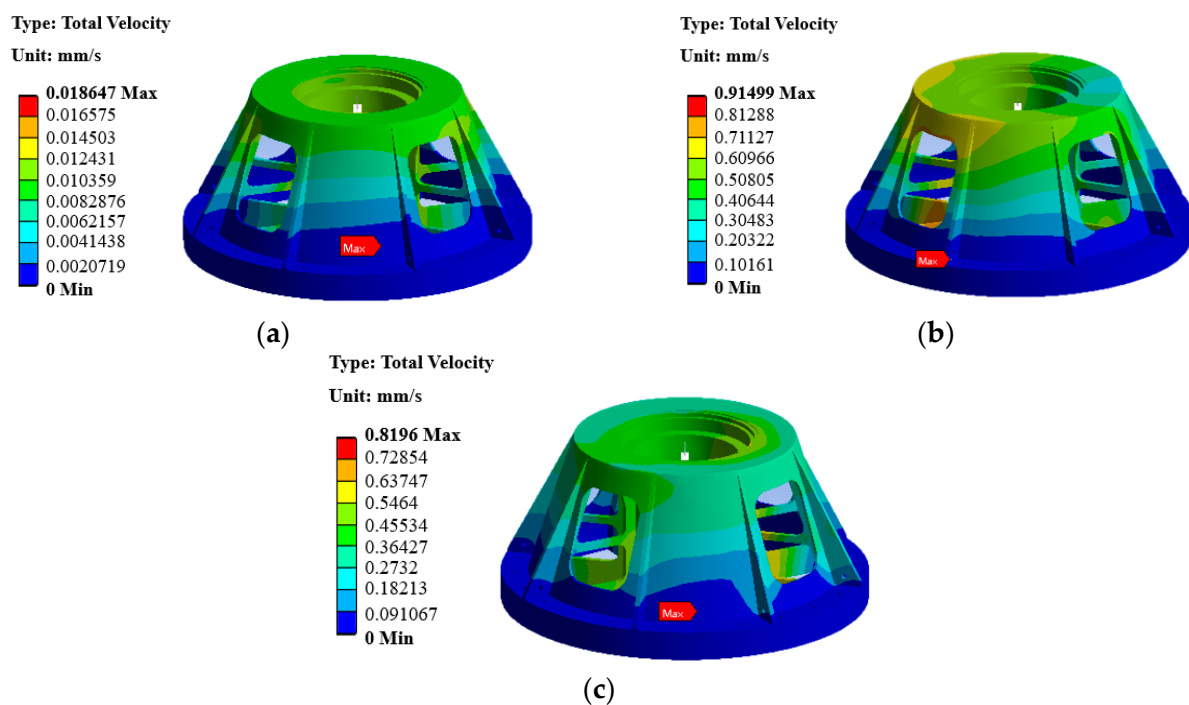


Figure 8. The improved vibration analysis results: (a) Vibration velocity response of the mounting seat under unbalanced excitation; (b) Vibration velocity response of mounting base under external vibration excitation; (c) Vibration velocity response of mounting seat under bearing vibration excitation.

Table 3. The harmonic response analysis results of the improved mounting seat.

Incentive Type	Mounting Seat Vibration (mm/s)
Unbalanced excitation	0.019
External vibration excitation	0.915
Bearing vibration	0.820
Total	1.754

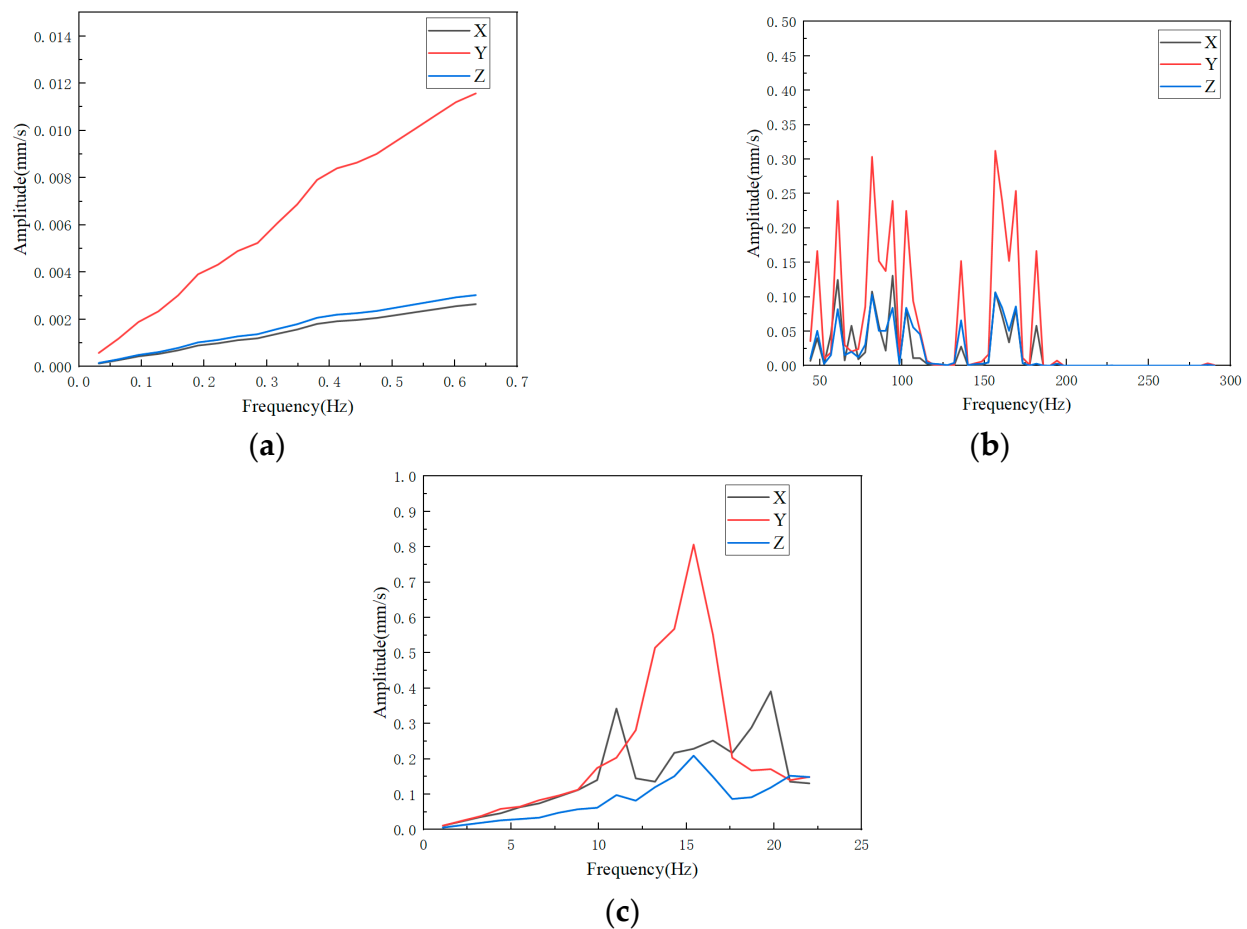


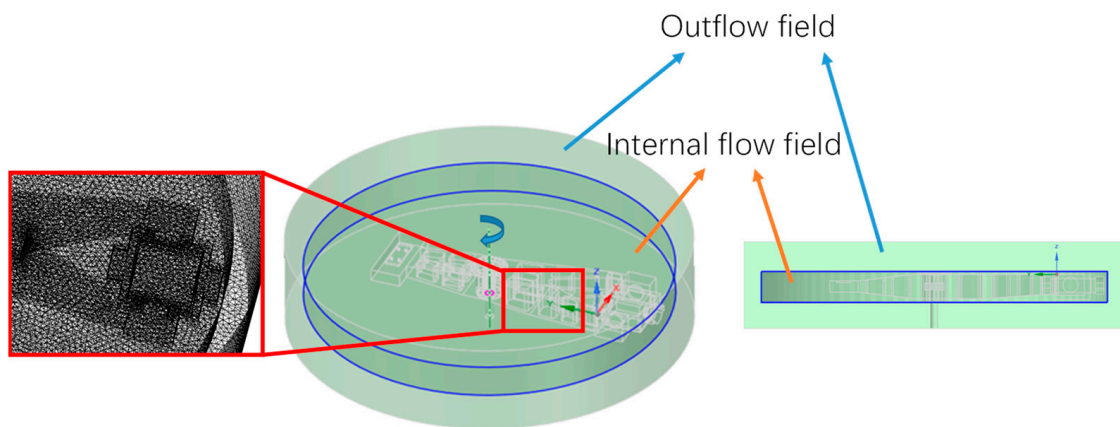
Figure 9. The improved vibration analysis results: (a) Vibration response spectrum of the mounting seat under unbalanced excitation; (b) Vibration response spectrum of the mounting base under external vibration excitation; (c) Vibration response spectrum of the mounting seat under bearing vibration excitation.

4. Aerodynamic Numerical Simulation and Structural Optimization Design of Test Centrifuge

In order to improve the stability of the test centrifuge during operation, considering the aerodynamic performance of the whole machine, the structure was optimized according to the aerodynamic numerical simulation results. The numerical model of the test centrifuge was imported into the ANSYS SCDM module to establish the flow field calculation model, and the flow field analysis was carried out based on the Fluent flow field calculation module. The operation and environmental parameters of the test centrifuge are shown in Table 4. The rotation direction of the test centrifuge was clockwise, and its flow field calculation model is shown in Figure 10. Numerical computational grids were generated for the computational domain of the centrifuge flow field, and the tetrahedral grids were used for the computational solution. The grids were refined near the walls to capture the boundary layers, and The first cell height (adjacent to the airfoil surfaces) was set to ensure a y^+ value lower than 1 [30]. In order to ensure the accuracy of the numerical calculations, the authors carried out a grid-independent verification. The authors selected three grids for numerical calculations to analyze the results of the drag coefficient calculation of the centrifuge model with different numbers of grids and compared the drag coefficient with other grids; it can be seen that as the number of grids is further increased, the change of drag coefficient value is less than 1%, as shown in Table 5. Therefore, the authors believe that the flow field is no longer sensitive to the number of grids. So, the authors selected grid 2 (1.24 million elements) as the final grid model for numerical calculations. The schematic diagram of the adopted grid is shown in Figure 10.

Table 4. Test centrifuge geometric parameters and operating conditions.

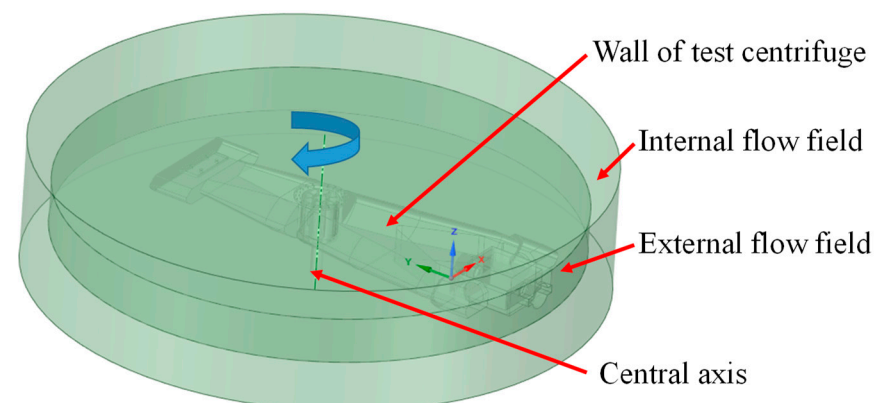
Incentive Type	Mounting Seat Vibration
Test centrifuge running space diameter/m	22
Test centrifuge running space height/m	5
air density/(kg/m ³)	1.205
temperature/°C	20
Test centrifuge rotating speed/rpm	38

**Figure 10.** Flow field calculation model of the test centrifuge.**Table 5.** Grid independence verification.

Type	Grids Number (Million)	Drag Coefficient
Grid 1	0.82	0.4916
Grid 2	1.24	0.4823
Grid 3	1.67	0.4797

4.1. Aerodynamic Numerical Simulation Method

The numerical simulation of the flow field uses the relative reference frame method to simulate the movement of the test centrifuge system. The wall of the test centrifuge was set to be static, the internal flow field was set to rotate around the central axis, and the speed was 38 rpm. The external flow field was relatively static; the external flow field contained the central axis wall. The absolute motion of the central axis wall was set, and the speed was 38 rpm, as shown in Figure 11.

**Figure 11.** Boundary conditions schematic diagram.

All simulations are carried out with Ansys Fluent. For the RANS approach with a Steady Realizable k-epsilon turbulence model, a SIMPLEC scheme is adopted to solve the coupling between the velocity and pressure fields. Convective terms are discretized with a second-order upwind scheme. Viscous terms are discretized with a second-order scheme [30,31]. The numerical calculation model and numerical calculation method in this paper were verified, and the reference paper is [32–34] a simulation of geotechnical centrifuges.

When the air is the flowing medium, the drag (lift) of the object moving in the air can be calculated by Equation (8). In the Equation, Y is the drag (lift) of the object or system moving in the air, and ρ is the density of the flow medium at a specific temperature. The flow medium analyzed in this paper is the air with a density of 1.169 kg/m^3 at 25°C , C is the drag coefficient, S is the windward area, and v is the flow velocity of the medium. According to Equation (8), the drag coefficient (lift coefficient) calculation Equation (9) is obtained.

$$Y = \frac{1}{2} \rho C S v^2 \quad (8)$$

$$C = \frac{2}{v^2 S \rho} Y \quad (9)$$

4.2. The Influence of the Whole-Package Shell of the Test Centrifuge on the Aerodynamic Resistance

The finite volume aerodynamic analysis of the flow field model of the test centrifuge shown in Figure 12a was carried out, and the vortex core distribution diagram obtained by the analysis was derived. The vortex core is a dense area of internal vortices. The vortex will produce vortex vibration, as will the vortex core. Vortex vibration is an important factor affecting the vibration of the rotating platform of the test centrifuge [35]. From the vortex core area map, it can be seen that the vortices at the rotating shaft and the pod were dense, as shown in Figure 13a. In order to analyze the influence of the whole-package shell on the aerodynamic drag, all openings of the model were filled in; that is, the influence of the whole-package shell on the surrounding flow field was simulated in the test centrifuge. The test centrifuge model after adding the whole-package shell is shown in Figure 12b. The aerodynamic analysis of the test centrifuge model after adding the whole-package shell was carried out, and the vortex core distribution map obtained by the analysis was derived. It was found that the overall vortex core quantity was reduced after the structure was changed, indicating that the whole-package shell structure suppressed the occurrence of vortex vibration, as shown in Figure 13b.

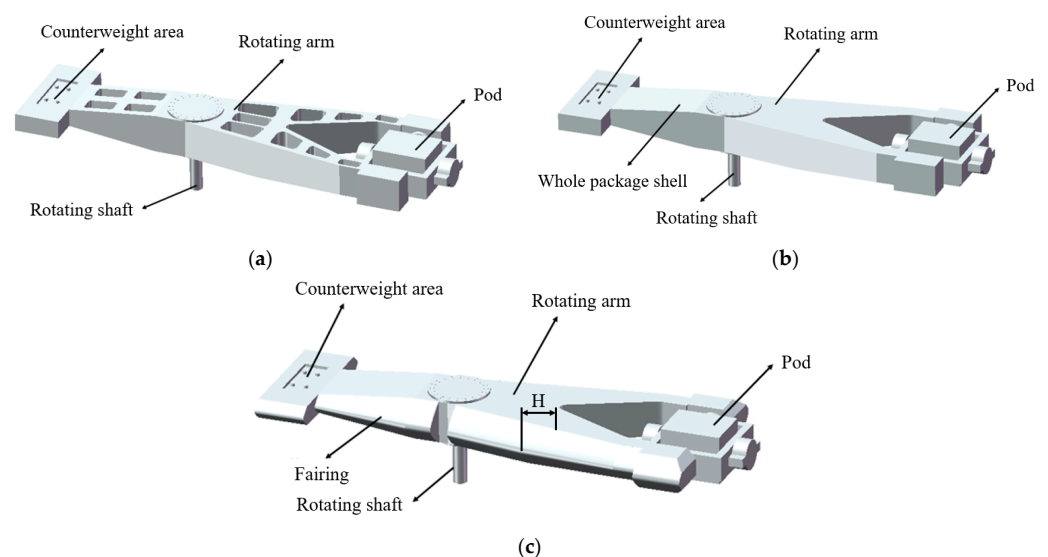


Figure 12. Illustration of model: (a) Test centrifuge; (b) Test centrifuge with whole package shell; (c) Test centrifuge with a fairing.

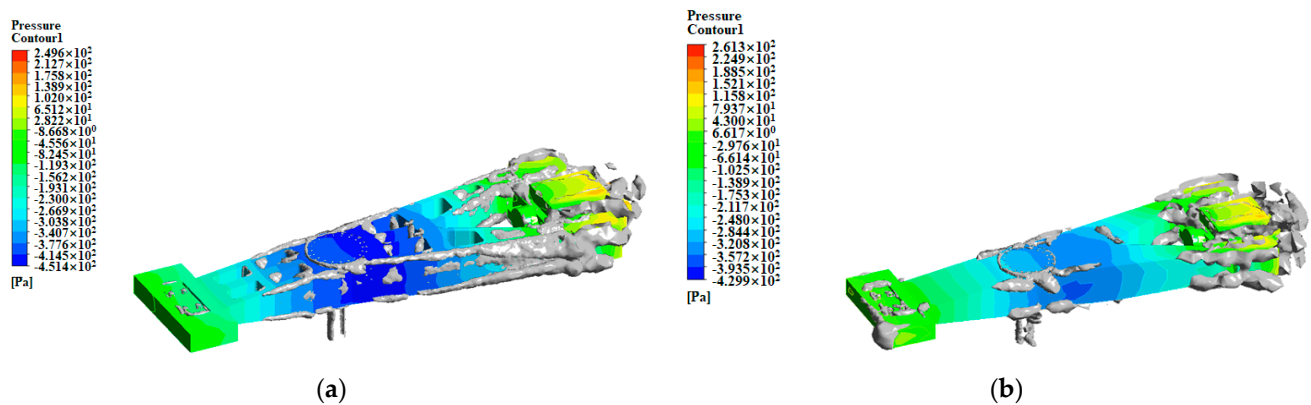


Figure 13. Vortex core diagram: (a) Schematic diagram of the vortex core area; (b) Schematic diagram of the vortex core area after adding the whole-package shell.

Using the analysis, it was concluded that the amount of vortex core was less in the state of the whole-package shell, and the decrease in the amount of vortex core means the decrease in the amount of vortex. During the operation process of the test centrifuge, the vortex shedding near the vortex will lead to vortex vibration. The generation of vortex vibration will cause the test centrifuge to be excited by pulsation, resulting in unstable motion and greater resistance during rotation. Therefore, the motion of the test centrifuge with the whole package is more stable, and the resistance is smaller than that without the whole package. Then, the structural improvement in the whole-package shell for the test centrifuge model was proposed.

4.3. The Influence of Fairing on the Aerodynamic Performance of Centrifuge

The test centrifuge model adds a whole package of casing, and the test centrifuge model is shown in Figure 12b. After the whole-package shell was added to the test centrifuge, its stability was improved, but the resistance of the centrifuge system was very large, especially on the windward side. Therefore, on the basis of the original structure, a fairing was added, as shown in Figure 12c. After increasing the fairing, the impact of air on the centrifuge itself can be reduced so that the airflow can smoothly cross the centrifuge. Due to the existence of the fairing on the leeward side, the phenomenon of vortex shedding also decreased to a certain extent. As shown in Figure 14, when the fluid flows through the centrifuge without fairing, it produces a larger impact on the centrifuge wall and generates gas reflux, which not only leads to an increase in the wind resistance of the centrifuge equipment but also generates fluid excitation by the reflux gas on both sides. When the fluid flows through the centrifuge with fairing, there is less impact on the centrifuge wall and no gas reflux, which helps to reduce the wind resistance of the centrifuge equipment. It is clear that fairing reduces the impact of the fluid impact on the centrifuge. The flow field model of the test centrifuge after adding the fairing was established, and the relevant parameters in the flow field remained unchanged.

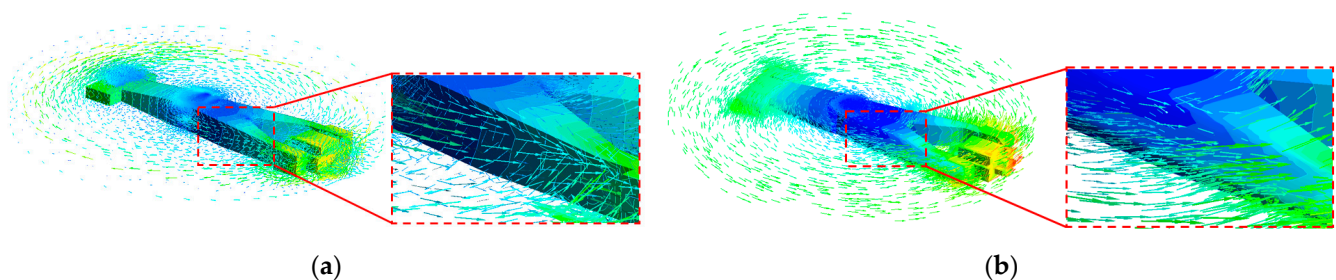


Figure 14. Velocity vector diagram: (a) Test centrifuge with whole package shell; (b) Test centrifuge with fairing.

The flow field analysis of the above two test centrifuge models was carried out, and the average flow velocity, drag coefficient, and lift coefficient of the two cases were obtained using CFD post-processing software. The relevant data are shown in Tables 6–9.

Table 6. Average flow rate under two models.

Flow Analysis Model	Average Flow Velocity (m/s)
Whole-package shell + fairing	16.67
Whole-package shell	17.02

Table 7. Drag and lift forces under two models.

Flow Analysis Model	Drag (N)	Lift (N)
Whole-package shell + fairing	1469.67	96.61
Whole-package shell	1885.19	204.30

Table 8. Windward area under two models (m²).

Flow Analysis Model	X-Direction	Z-Direction
Whole-package shell + fairing	23.83	66.84
Whole-package shell	23.60	47.79

Table 9. Drag and lift coefficients under two models.

Flow Analysis Model	Drag Coefficient	Lift Coefficient
Whole-package shell + fairing	0.3796	0.0089
Whole-package shell	0.4717	0.0252

4.4. Optimization Design of Fairing

The stability of the test centrifuge during rotation is particularly important. As mentioned above, the test centrifuge with a steady-flow shell had a good aerodynamic performance, and the resistance and resistance coefficient were reduced. In order to further improve the stability of the test centrifuge during operation and explore the influence of the streamline of the steady-flow shell on the test centrifuge, five different streamline models of the steady-flow shell were established to optimize the steady-flow shell of the test centrifuge model. The streamlined improvement position of the fairing was mainly in the front wing and the rear wing, as shown in Figure 12c.

The change of streamline was mainly to change the width, H, of the fairing extending to the side, as shown in Figure 13b. The side extension width of the front wing and the rear wing was adjusted, and the design width was 250 mm, 350 mm, 500 mm, 700 mm, and 950 mm. The aerodynamic analyses of five fairing models were carried out. The drag coefficient is shown in Figure 15 and Table 10.

The analysis results show that the higher the extension width of the front and rear wings, the lower the drag coefficient, and the maximum decrease was 11.4%. The aerodynamic performance of the test centrifuge was improved, and the stability of the equipment during operation was guaranteed.

The drag and lift, drag coefficient, and lift coefficient were obtained under two conditions: with and without the fairing. The above analysis shows that the resistance of the test centrifuge with the fairing was smaller than that without the fairing, and the decrease was 22.0%. The resistance coefficient with the fairing was 24.6% lower than that without the fairing. The lift coefficient with the fairing was 64.7% lower than that without the fairing, and the lift with the fairing was 52.7% lower than that without the fairing.

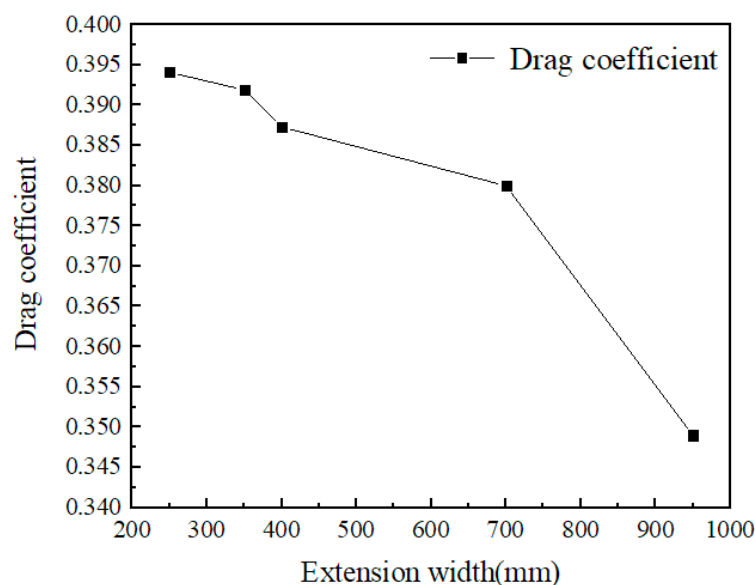


Figure 15. Drag coefficients of different extension widths.

Table 10. Drag coefficients under five models.

Extension Width (mm)	Drag Coefficient
250	0.3941
350	0.3919
500	0.3873
700	0.38
950	0.349

5. Conclusions

In this paper, the vibration and aerodynamic performance of a certain type of test centrifuge were studied. The vibration and aerodynamic numerical simulation of centrifuges with different structures was carried out. The changes in the vibration and aerodynamic performance of the test centrifuges before and after optimization were analyzed. The main conclusions are as follows.

The strengthened mounting seat can effectively reduce the vibration response of the equipment. The overall vibration response of the equipment was reduced from 7.747 mm/s to 1.754 mm/s, with a decrease of 77.36%.

The whole-package shell can effectively improve the stability of the test centrifuge system during operation. The steady-flow shell can significantly reduce the resistance and lift of the test centrifuge during operation. The resistance decreased by 22.0%, the resistance coefficient decreased by 24.6%, the lift decreased by 64.7%, and the lift coefficient decreased by 52.7%.

For a certain type of test centrifuge, the application of a strengthened mounting seat and steady-flow shell structure can improve the stability of the equipment during operation. At the same time, the optimization design in this paper can improve the design efficiency of test centrifuge equipment.

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