

Article

Thermal Characteristics and Dynamic Behavior of Auxiliary Bearings in a Vertical Magnetic Suspension System

Xiaoxu Pang ^{1,2,*}, Chongfeng Jiang ¹, Zhixin Shen ³, Dingkang Zhu ^{4,5,*}, Aosha Wang ⁶ and Kaili Wang ⁷

¹ School of Mechanical and Electrical Engineering, Henan University of Science and Technology, Luoyang 471003, China; jiangchongfeng@stu.haust.edu.cn

² High-Level Bearing Henan Synergy Innovation Center, Luoyang 471003, China

³ Luoyang Bearing Research Institute Co., Ltd., Luoyang 471003, China; syjkbfb@zys.com.cn

⁴ State Key Laboratory of Tribology in Advanced Equipment, Tsinghua University, Beijing 100084, China

⁵ School of Mechanical Engineering, Tsinghua University, Beijing 100084, China

⁶ College of Physics Engineering, Henan University of Science and Technology, Luoyang 471003, China; 240320090822@stu.haust.edu.cn

⁷ Zhejiang Pinnuo Machinery Co., Ltd., Shaoxing 312580, China; wkl@pace-china.com

* Correspondence: pxx8308@haust.edu.cn (X.P.); zdkkong@mail.tsinghua.edu.cn (D.Z.)

Abstract

Auxiliary bearings in vertical magnetic suspension systems can suffer thermal damage and impact-induced failure during rotor drop events caused by instability. This study aims to clarify the coupled effects of collision, frictional heating, and transient heat transfer on auxiliary bearing response. Dynamic, thermodynamic, and finite element models were established to analyze impact behavior, frictional heating, and temperature-field evolution, and were validated using rotor-drop measurements of impact force, rotor displacement, and outer-ring temperature together with post-test damage observations. The results show that severe impact and friction rapidly convert rotor kinetic energy into thermal energy, producing a non-uniform temperature field in the auxiliary bearings. The highest temperature occurs in the inner ring, followed by the rolling elements and outer ring, with peak temperatures of 169.59 °C, 154.66 °C, and 94.79 °C, respectively. Owing to gravity, gyroscopic motion, and rotor inclination during drop, the upper auxiliary bearing experiences greater impact loads, a faster speed increase, and a higher peak temperature rise than the lower bearing. Experimental evidence, including thermal discoloration, wear positions, and component damage, agrees with the simulated high-temperature regions. These results support thermal-shock-resistant design, structural optimization, and operational safety assessment of auxiliary bearings.

Keywords: vertical magnetic suspension system; rotor drop; auxiliary bearing; thermal characteristics; dynamics



Academic Editor: He Liang

Received: 9 May 2026

Revised: 14 June 2026

Accepted: 24 June 2026

Published: 30 June 2026

Copyright: © 2026 by the authors.

Licensee MDPI, Basel, Switzerland.

This article is an open access article distributed under the terms and conditions of the [Creative Commons Attribution \(CC BY\)](https://creativecommons.org/licenses/by/4.0/) license.

1. Introduction

Active magnetic bearings (AMBs) offer advantages such as high-speed capability, frictionless operation, and environmental friendliness, and can effectively eliminate the mechanical friction associated with conventional rolling bearings [1]. Therefore, they have been widely applied in fields such as aerospace and industrial machinery [2]. The auxiliary bearing is one of the indispensable key components of a magnetic suspension system and serves as an important safety boundary of the system. It withstands the severe impact loading and thermal shock caused by rotor instability.

For high-speed rotating machinery, common bearing options include rolling element bearings, oil-film bearings, journal bearings, gas bearings, air foil bearings, active magnetic bearings, passive magnetic bearings, superconducting magnetic bearings, and hybrid gas-magnetic bearings. Magnetic bearings are generally adopted when conventional lubricated bearings or gas bearings cannot meet the requirements of clean operation, high-vacuum operation, avoidance of oil contamination, corrosive or toxic gas environments, very high rotational speed, low vibration, long service life, or active rotor-dynamic control [1,3–6]. Active magnetic bearings are widely used in high-speed compressors, turbomachinery, helium blowers, turbomolecular pumps, and flywheel energy-storage systems [7,8]. Passive magnetic bearings are mainly used for load reduction or axial support in hybrid suspension systems, whereas superconducting magnetic bearings are attractive for low-loss flywheel energy-storage applications [9,10].

Touchdown conditions differ among applications. In some horizontal-axis turbomachinery or machines with repeated start–stop operation, touchdown may occur during start-up, shutdown, or transient control states, where wear resistance, repeatability, and contact stability are important. In contrast, in flywheel energy-storage systems, turbomolecular pumps, and vertical magnetic suspension systems, touchdown is usually an emergency event caused by power loss, control failure, or rotor instability. In such cases, auxiliary bearings, also referred to as backup bearings, touchdown bearings, or emergency bearings, are mostly implemented as deep-groove ball bearings or angular-contact ball bearings, and must withstand short-duration impact, rapid spin-up, severe frictional heating, and transient thermal shock. Failure analysis of auxiliary bearings has shown that most auxiliary bearings do not fail primarily because of direct impact damage, but rather because they are thermally damaged by the high temperatures generated within a short period of time [11]. These features make the emergency rotor-drop behavior of auxiliary bearings in vertical magnetic suspension systems an important issue for dynamic and thermal reliability analysis.

Magnetic suspension systems can be configured with either horizontal-axis or vertical-axis rotors, and the rotor touchdown mechanisms in these two configurations are considerably different [3–7]. In horizontal-axis systems, rotor touchdown is usually dominated by gravity-induced radial sag, radial rubbing, rebound, and whirl motion [4–8]. By contrast, in vertical-axis systems, the direction of gravity is aligned with the rotor axis. Therefore, magnetic suspension failure first induces axial rotor drop and axial impact on the upper auxiliary bearing, followed by radial contact, rotor inclination, gyroscopic motion, inner-ring spin-up, and asymmetric load distribution between the upper and lower auxiliary bearings [3,9–14]. These characteristics make the coupling among axial and radial contact, dynamic response, friction-induced heat generation, and transient temperature-field evolution particularly pronounced in vertical magnetic suspension systems [3,9–12,15,16]. Therefore, vertical magnetic suspension systems represent an important and distinct class of applications for investigating the dynamic–thermal coupling behavior of auxiliary bearings under emergency rotor-drop conditions.

Extensive studies have been conducted on the thermal characteristics of auxiliary bearings under the severe frictional heating and complex contact conditions induced by rotor drop in magnetic suspension systems. Keogh et al. [12] proposed an improved transient thermal Green’s function method to predict highly localized contact temperatures caused by bouncing and friction, and analyzed the thermal responses of auxiliary bearings with a fixed bushing and a rotatable inner ring. Salehi et al. [13] investigated the dynamic and thermal performance of zero-clearance auxiliary bearings using a thermal resistance network model, and validated the predicted temperature rise experimentally. Kong et al. [14] analyzed the thermal behavior of auxiliary bearings in a vertical

magnetic suspension system using both the thermal network method and the finite element method, and clarified the temperature distribution among internal components. Liu et al. [15] analyzed the frictional heat generation power of oil-free auxiliary bearings during rotor drop and showed, through free-drop experiments, that ball spin heating and differential sliding heating are the dominant heat sources. For high-temperature gas-cooled reactor magnetic suspension systems, Zhao et al. [11,16–18] systematically investigated the dynamics, thermal characteristics, and failure behavior of auxiliary bearings in HTR-10 and HTR-PM systems. Their results showed that transient heat generation at the rotor–auxiliary bearing interface is the primary heat source, and verified that dry-lubricated ceramic auxiliary bearings in vertical systems can withstand severe transient thermal shock during rotor drop. Neisi et al. [19] further demonstrated that high-amplitude surface waviness significantly increases internal contact force and frictional power loss, thereby intensifying thermal expansion and temperature rise of the inner ring. Li et al. [20–22] established coupled dynamic–thermal models for vertical rotor drop and, together with experimental validation, clarified the evolution of dry-friction whirl and dry-friction whip, as well as the influence of friction parameters, rotor eccentricity, and protective clearance on impact force and temperature rise. Nie [23] and Li et al. [24] showed through simulations and experiments that solid-lubricant coatings prepared by magnetron sputtering can effectively reduce friction, suppress temperature rise, and improve the service reliability of auxiliary bearings.

With regard to dynamic response and failure mechanisms, existing studies have mainly focused on the impact characteristics, nonlinear vibration behavior, and coupled effects of multiple factors in auxiliary bearings during rotor drop. Ebrahimi et al. [25] developed a 10-degree-of-freedom dynamic model of a coaxial rotor–active magnetic bearing system and analyzed the influence of speed, speed ratio, and gravity on nonlinear responses such as chaos and quasiperiodic vibration, together with the corresponding rotor–auxiliary bearing contact forces. Halminen et al. [26] established a detailed two-dimensional multibody model with realistic elastic contact to compare the dynamic response and contact force of auxiliary bearings with and without a cage. The presence or absence of a cage has an important influence on the dynamic and thermal behavior of auxiliary bearings during rotor drop. In a caged bearing, the cage separates and guides the rolling elements in the circumferential direction, thereby reducing direct rolling element–rolling element contact, collision, and differential sliding. This improves high-speed kinematic stability and reduces the risk of inter-ball frictional heating and seizure [26–28]. Zhu et al. [27,28] further showed that a caged structure with an optimized ball filling number can reduce radial impact force and improve rotational stability during rotor drop. However, because the cage occupies part of the internal space, the number of rolling elements is usually smaller than that in a full-complement bearing, and the instantaneous load-carrying capacity may therefore be lower. In addition, the cage itself may suffer wear, deformation, or failure under severe impact, high-speed friction, cage slip, and temperature rise [29,30]. In contrast, a cageless or full-complement auxiliary bearing contains more rolling elements and therefore provides higher transient load-carrying capacity, which is beneficial under emergency touchdown or high-impact rotor-drop conditions. However, due to the absence of cage guidance, direct rolling element–rolling element contact, mutual squeezing, differential sliding, and local sliding friction are more likely to occur. These effects may intensify frictional heat generation and increase the risk of rolling-element seizure, local burn damage, and material degradation during high-speed rotor drop [31–33]. Jarroux et al. [29,34] investigated rotor drop dynamics after active magnetic bearing failure and rotor–active magnetic bearing interactions under rotor–auxiliary bearing contact while the active magnetic bearing

remained operational. Through numerical and experimental analyses, they showed that support stiffness, damping, and friction strongly influence impact load, rebound trajectory, synchronous-following behavior, nonlinear dynamic response, and transient stability under strong harmonic and impact excitations. Sun et al. [35] developed a high-fidelity ball-bearing model and clarified the effects of friction coefficient, preload, support damping, and side load on auxiliary-bearing performance and backward-whirl stability. Ma et al. [36] built a flexible rotor–auxiliary bearing model using Abaqus and Caba3D, and analyzed the responses of rotational speed, rotor trajectory, rotor–inner ring impact force, ball-to-ball impact force, and ball–inner ring sliding velocity during rotor drop. Kong et al. [37] used ADAMS to study the effects of friction coefficient, axial clearance, rotor mass, and speed on the initial axial impact force in a vertical magnetic suspension system, showing that the maximum impact force increases with axial clearance and rotor mass, decreases with kinetic friction coefficient, and is only weakly affected by rotor speed. Xiao et al. [38] analyzed the load-carrying behavior of auxiliary bearings in the HTR-10 helium blower based on Hertzian contact theory, and showed that while the axial drop impact can be effectively resisted, radial load during the free-running stage is mainly induced by rotor rotation and heat accumulation in the inner ring, which may eventually cause failure. Fonseca et al. [39] demonstrated that system response is strongly influenced by unbalance: low unbalance leads to bottom oscillation, moderate unbalance induces strongly nonlinear chaotic impacts, and high unbalance results in sustained forward whirl. Pang et al. [40,41] systematically analyzed the nonlinear contact dynamics and attitude evolution during eccentric rotor drop, and further revealed the mechanism by which active clearance-elimination auxiliary bearings suppress impact loading and improve drop response. Zhao et al. [42] conducted rotor drop experiments on a 10 MW vertical helium blower test rig and, based on a momentum–impulse inversion method, revealed the impact-load distribution and dynamic behavior of upper and lower auxiliary bearings under different speeds and bearing materials. Sun et al. [43] developed a detailed model including nonlinear Hertzian contact, a squeeze-film damper, and a one-dimensional thermal model to analyze rotor drop dynamics and friction-induced thermal growth, and proposed design guidelines based on friction coefficient, axial preload, and damping viscosity. Lai et al. [44] showed through a 24,000 r/min, 1500 h life test that frictional heating is clearly coupled with near-surface material evolution. Lee and Palazzolo [45] combined a thermally expanded nonlinear auxiliary-bearing model with rainflow counting and Miner’s rule, and showed that high-speed backward whirl accelerates cumulative damage and dominates failure evolution. Jia et al. [46] further demonstrated through a multiphysics model and repeated drop tests that auxiliary-bearing failure is essentially a process of burn damage, spalling, and material degradation caused by the combined effects of impact-induced plastic deformation, hard debris scratching, local high temperature, and ball seizure. Recent studies have further extended the investigation of rotor-drop behavior, protective-bearing design, and coupled modeling in magnetic bearing systems. Zhang and Xu [47] evaluated the recontrol feasibility of an active magnetic bearing rotor after falling onto touch-down bearings, showing that appropriate electromagnetic damping can help suppress the post-drop rotor response. Pang et al. [41] investigated the dynamic behavior of a vertical suspension rotor falling on an active clearance-elimination protective bearing and demonstrated, through simulation and drop tests, that reducing the protective clearance can effectively alleviate impact intensity. Zhu et al. [27] further analyzed the influence of structural parameters on the dynamic response of a vertical magnetic suspension rotor dropping onto protective bearings, highlighting the importance of protective-bearing configuration in impact-load control. In addition, Du et al. [48] reviewed recent developments in magnetic bearings and emphasized that multi-field

coupled modeling, intelligent control, reliability-oriented design, and integrated optimization are becoming important research trends. These recent advances indicate that the coupled evolution of collision response, inner-ring spin-up, friction-induced heat generation, and transient temperature-field distribution in vertical magnetic suspension auxiliary bearings remains an important issue for reliability-oriented design.

In summary, previous studies have substantially advanced the understanding of auxiliary-bearing dynamics and friction-induced heating during rotor drop events in magnetic suspension systems, with important progress achieved in dynamic modeling, heat-generation analysis, and experimental validation. Nevertheless, several limitations remain. Existing studies have largely addressed either the dynamic response or the thermal behavior in a relatively separated manner. Although some studies have considered temperature rise, simplified thermal resistance networks, one-dimensional heat-transfer models, or local temperature estimations have commonly been adopted, which are insufficient to fully describe the continuous coupling among collision response, inner-ring spin-up, friction-induced heat generation, and transient temperature-field evolution during rotor drop. In particular, from the perspective of simulation and theoretical modeling, there is still a lack of a unified framework capable of simultaneously capturing axial contact, radial contact, multiple sources of frictional heat generation, the asymmetric responses of the upper and lower auxiliary bearings, and the three-dimensional transient spatial temperature-field distribution.

Based on the above considerations and the identified research gaps, the present study focuses on the auxiliary bearings of a vertical magnetic suspension system and investigates their dynamic and thermal responses under emergency rotor-drop conditions. Dynamic and thermodynamic models of the rotor–auxiliary-bearing system are established, together with a finite element simulation model incorporating actual material properties, contact boundaries, gravity direction, initial rotor speed, and structural constraints. On this basis, the collision-load characteristics, inner-ring speed response, friction-induced heating behavior, and temperature-field evolution during rotor drop are systematically analyzed. Particular emphasis is placed on clarifying the differences between the upper and lower auxiliary bearings in terms of contact sequence, impact load, inner-ring acceleration, temperature-rise rate, and thermal-failure risk in a vertical configuration. The proposed framework reveals the coupling mechanism between the dynamic response and thermal behavior of auxiliary bearings and provides a theoretical basis for thermal-shock-resistant design, structural optimization, and operational safety assessment of auxiliary bearings in vertical magnetic suspension systems.

2. Model Architecture

In the vertical magnetic suspension system, the rotor is arranged along the vertical direction. In this system, both the axial magnetic bearing and the radial magnetic bearings are active magnetic bearings. During normal operation, the axial active magnetic bearing balances the rotor weight and controls the axial position of the rotor, whereas the radial active magnetic bearings control the radial position of the rotor. The upper auxiliary-bearing assembly consists of a pair of angular-contact ball bearings mounted in a face-to-face configuration. Unless otherwise specified, the upper auxiliary bearing and the lower auxiliary bearing mentioned in the following analysis refer to the two angular-contact auxiliary bearings within the upper auxiliary-bearing assembly. These bearings primarily support the radial and axial loads generated during rotor drop events. By contrast, the separate auxiliary bearing located at the lower end of the rotor is referred to as the lower auxiliary-bearing assembly, which is a single deep-groove ball bearing and carries only radial load. A schematic diagram of the vertical magnetic sus-

pension system is shown in Figure 1. The axial and radial protective clearances between the rotor and the auxiliary bearings were both 0.2 mm, as determined by the existing test-rig configuration. These clearance values were used consistently in the finite element model and rotor-drop experiments. The protective clearances allow non-contact operation during normal magnetic suspension and define the axial drop distance and radial displacement before touchdown after magnetic suspension failure. Because the contact mechanics between the upper auxiliary-bearing assembly and the rotor are more complex, the upper auxiliary-bearing assembly is selected in this study as the main object of analysis and discussion.

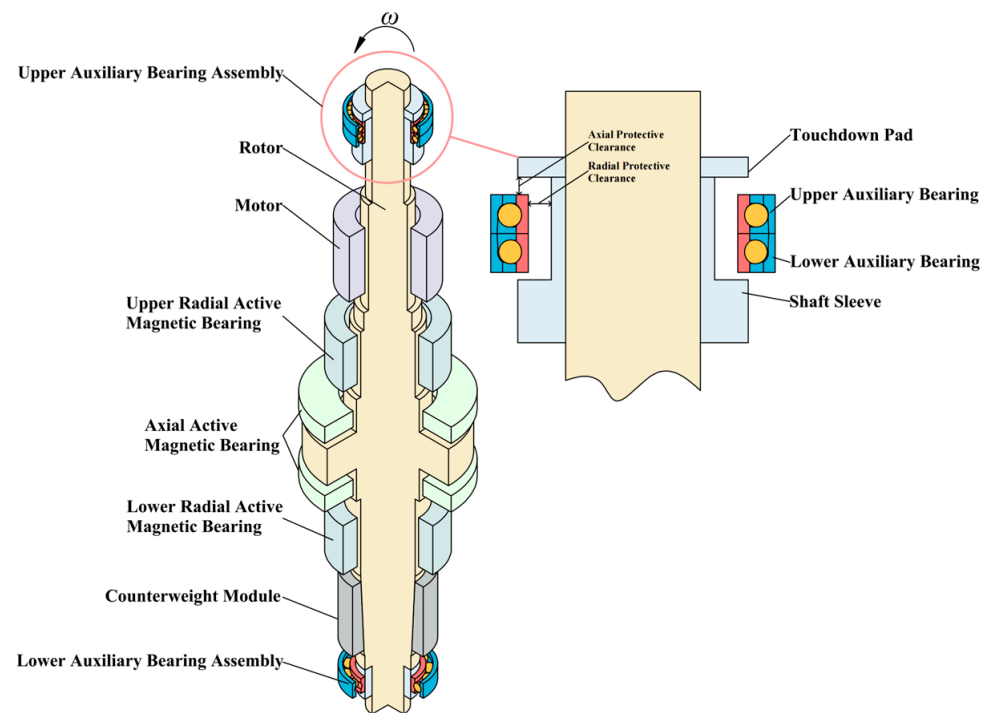


Figure 1. Schematic diagram of the vertical magnetic suspension system.

The vertical magnetic suspension system considered in this study is mainly intended for non-contact support and emergency protection in high-speed vertical rotating equipment, such as magnetically levitated motors, high-speed centrifugal machines, flywheel energy-storage rotors, and helium blowers/circulators. During normal operation, the rotor is supported without mechanical contact by active magnetic bearings. When magnetic suspension control fails or rotor instability occurs, the auxiliary bearings act as emergency support structures to limit rotor displacement and withstand transient impact loads and friction-induced thermal loads. Therefore, the “supported mass” investigated in this study corresponds to the rotor mass supported by the magnetic suspension system and dropped onto the auxiliary bearings after instability. The main operating parameters adopted in the simulations and experiments are as follows: rotor mass of 45 kg, rotor length of 765 mm, mean rotor diameter of 45 mm, axial protective clearance of 0.2 mm, radial protective clearance of 0.2 mm, and initial drop speed of 20,000 r/min, corresponding to a synchronous rotational frequency of approximately 333.3 Hz. It should be noted that this study focuses on the transient rotor-drop process after active magnetic-bearing failure rather than a vibration-isolation problem under steady harmonic excitation. Therefore, no fixed-frequency external harmonic excitation is prescribed in this study. The main external and initial excitations considered include gravity and the initial high-speed rotation of the rotor, while the transient impact and rubbing between the rotor and the auxiliary bearings are analyzed as the primary contact excitations during the rotor-drop process.

2.1. Dynamic Model of the Auxiliary Bearing

The contact interaction between the rotor and the auxiliary bearing during the rotor drop process is illustrated in Figures 2 and 3. In these figures, α , ρ , and γ represent the dynamic characteristics of the rotor. O_b denotes the geometric center of the auxiliary bearing, O_r denotes the geometric center of the rotor, and O_c denotes the center of mass of the rotor. Point A is the axial contact point between the touchdown pad and the inner ring of the auxiliary bearing. α is the angle between O_bO_r and the x -axis, representing the whirl angle of the rotor. θ is the angle between O_bA and the x -axis and is used to describe the gyroscopic rotational motion of the rotor at the next instant of contact. ρ is the relative distance between O_b and O_r , representing the radius of rotor motion in the horizontal plane. $\dot{\alpha}$ denotes the whirl angular velocity of the rotor, e is the rotor eccentricity, R_{b1} denotes the radius of the inner surface of the inner ring, and R_{b2} denotes the radius of the inner-ring shoulder. γ denotes the spin angle of the rotor, $\dot{\gamma}$ denotes the spin angular velocity of the rotor, and $\dot{\theta}_b$ denotes the angular velocity of the inner ring of the auxiliary bearing. i and j are the unit vectors in the polar coordinate system, F_t denotes the tangential contact force, F_r denotes the radial contact force, and r denotes the radial coordinate in the polar coordinate system.

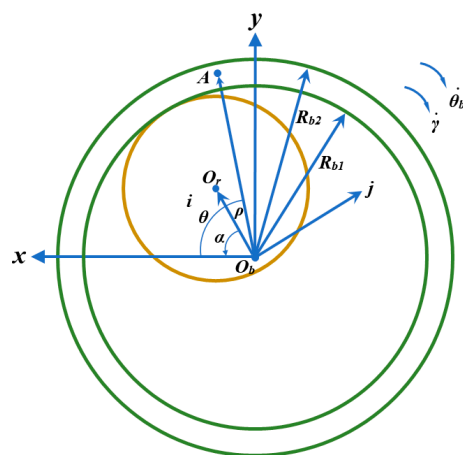


Figure 2. Axial contact interaction.

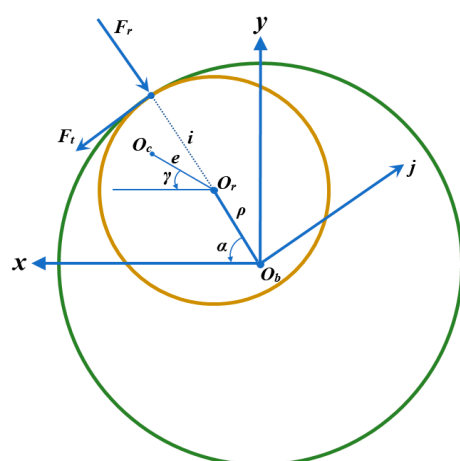


Figure 3. Radial contact interaction.

Contact between the rotor and the auxiliary bearing is determined by calculating their relative displacement. Axial impact occurs when the axial penetration depth is positive, whereas radial impact occurs when the radial penetration depth is positive. Based on the Hertzian contact model, the load–deformation relationship during the rotor drop process

can be obtained. Accordingly, the axial impact force F_a generated by the impact between the rotor and the upper auxiliary bearing can be derived [40], as shown in Equation (1):

$$F_a = \begin{cases} K_a \delta_a^e + C_a \dot{\delta}_a, & \delta_a > 0 \\ 0, & \delta_a \leq 0 \end{cases} \quad (1)$$

In the above equation, K_a denotes the axial contact stiffness, δ_a denotes the axial impact depth, and C_a denotes the axial contact damping. e is a parameter associated with the contact type. Specifically, $e = 3/2$ for point contact, which occurs within the auxiliary bearing; $e = 10/9$ for line contact, which occurs at the radial contact interface; and $e = 1$ for surface contact, which occurs at the axial contact interface between the rotor and the inner ring of the auxiliary bearing.

$$F_{af,i} = -P_a \mu_a \int_{R_{b1}}^{R_{b2}} \int_0^{2\pi} \frac{[\dot{\rho} + (\dot{\gamma} - \dot{\theta}_b)r \sin \theta] \vec{i}}{\sqrt{[\dot{\rho} + (\dot{\gamma} - \dot{\theta}_b)r \sin \theta]^2 + [\dot{\alpha}\rho - (\dot{\theta}_b - \dot{\gamma})r \cos \theta - \rho\dot{\gamma}]^2}} r d\theta dr \quad (2)$$

$$F_{af,j} = -P_a \mu_a \int_{R_{b1}}^{R_{b2}} \int_0^{2\pi} \frac{[\dot{\alpha}\rho + (\dot{\gamma} - \dot{\theta}_b)r \cos \theta - \rho\dot{\gamma}] \vec{j}}{\sqrt{[\dot{\rho} + (\dot{\gamma} - \dot{\theta}_b)r \sin \theta]^2 + [\dot{\alpha}\rho - (\dot{\theta}_b - \dot{\gamma})r \cos \theta - \rho\dot{\gamma}]^2}} r d\theta dr \quad (3)$$

$$T_a = \frac{2\mu_a F_a (R_{b2}^3 - R_{b1}^3)}{3(R_{b2}^3 - R_{b1}^3)} \quad (4)$$

In the above equations, P_a denotes the contact stress, and μ denotes the coefficient of friction at the axial contact interface.

Due to rotor unbalance, eccentricity, and other factors, radial contact impact occurs between the rotor and the inner surface of the inner ring of the auxiliary bearing [47]. The radial contact force can be expressed by Equation (5):

$$F_r = \begin{cases} K_r \delta_r^e + C_r \dot{\delta}_r, & \delta_r > 0 \\ 0, & \delta_r \leq 0 \end{cases} \quad (5)$$

In this equation, K_r denotes the radial contact stiffness, δ_r denotes the radial penetration depth, and C_r denotes the radial contact damping.

The rotational motion of the auxiliary bearing is jointly driven by the axial friction torque and the radial friction torque. Accordingly, the rotational motions of the rotor and the inner rings of the upper and lower auxiliary bearings are respectively given by Equations (6)–(8):

$$\ddot{\gamma} = -\frac{(F_{t1} + F_{t2})R_r + T_a}{I_p} \quad (6)$$

$$\ddot{\theta}_{b1} = -\frac{F_{t1}R_{b1} + T_a - T_{b1}}{I_b} \quad (7)$$

$$\ddot{\theta}_{b2} = -\frac{F_{t2}R_{b1} - T_{b2}}{I_b} \quad (8)$$

In these equations, F_{t1} and F_{t2} represent the tangential friction forces of the upper and lower auxiliary bearings, respectively, whereas T_{b1} and T_{b2} represent the corresponding internal bearing torques, which are calculated using the Palmgren empirical formula:

$$T_{b,j} = T_{0,j} + T_{1,j} \quad j = 1, 2 \quad (9)$$

Here, T_0 is generated by the lubricant frictional resistance under no-load conditions and is influenced by the lubricant, bearing type, and rotational speed:

$$T_0 = \begin{cases} 10^{-7} f_0 (v \dot{\theta}_b)^{2/3} d_m^3 & v n \geq 2000 \\ 160 \times 10^{-7} f_0 d_m^3 & v n \leq 2000 \end{cases} \quad (10)$$

In this equation, f_0 denotes a coefficient related to the bearing type and lubrication method, v denotes the kinematic viscosity of the lubricant at the bearing operating temperature, and d_m denotes the pitch diameter of the bearing.

T_l reflects the friction losses caused by elastic hysteresis and differential sliding, and its calculation formula is given in Equation (11):

$$T_l = f_1 P_1 d_m \quad (11)$$

In this equation, P_1 denotes the equivalent dynamic load of the bearing, and f_1 denotes a coefficient related to the bearing type and the applied load.

When rolling friction takes place at the contact interface of the upper auxiliary bearing, F_{t1} is determined by simultaneously solving Equations (6), (7), and (12), as expressed in Equation (13):

$$R_r \dot{\gamma} = R_{b1} \dot{\theta}_{b1} \quad (12)$$

$$F_{t1} = \frac{-F_{t2} \cdot R_r^2 / I_p - T_a (R_r / I_p + R_{b1} / I_b) + T_{b1} \cdot R_{b1} / I_b}{R_r^2 / I_p + R_{b1}^2 / I_b} \quad (13)$$

In this equation, the tangential friction force of the lower auxiliary bearing is expressed as $F_{t2} = \mu_d F_{r2}$, where μ_d is the coefficient of kinetic friction. I_p denotes the polar moment of inertia of the rotor, and I_b denotes the equivalent rotational inertia of the rotating part of the auxiliary bearing. In this study, I_b includes the mass moment of inertia of the inner race and the effective inertia contribution of the rolling elements associated with the bearing rotational motion.

When rolling friction occurs at the contact interface of the lower auxiliary bearing, F_{t2} can be calculated by solving Equations (6) and (8), as shown in Equation (15).

$$R_r \dot{\gamma} = R_{b1} \dot{\theta}_{b2} \quad (14)$$

$$F_{t2} = \frac{-F_{t1} \cdot \frac{R_r^2}{I_p} - T_a \cdot \frac{R_r}{I_p} + T_{b2} \cdot \frac{R_{b1}}{I_b}}{\frac{R_r^2}{I_p} + \frac{R_{b1}^2}{I_b}} \quad (15)$$

In this equation, the tangential friction force is given by $F_{t1} = \mu_d F_{r1}$.

In the paired auxiliary bearing arrangement, rolling resistance exists at the contact interfaces of the upper and lower auxiliary bearings, which can be determined from Equations (6) and (8):

$$R_r \dot{\gamma} = R_{b1} \dot{\theta}_{b1} = R_{b1} \dot{\theta}_{b2} \quad (16)$$

$$F_{t1} = \frac{-\frac{R_r^2 (T_a + T_{b1} + T_{b2})}{R_{b1} I_p} - \frac{R_{b1} (T_a - T_{b1})}{I_b} + \frac{R_r T_a}{I_p}}{2 R_r^2 / I_p + R_{b1}^2 / I_p} \quad (17)$$

$$F_{t2} = F_{t1} + \frac{T_a - T_{b1} + T_{b2}}{R_{b1}} \quad (18)$$

When $F_{tj} > \mu_s F_{rj}$, sliding friction occurs, and $F_{tj} = \mu_d F_{rj}$; when $F_{tj} < \mu_s F_{rj}$, rolling friction occurs. Here, μ_s is the coefficient of static friction.

At the instant of rotor touchdown, the auxiliary-bearing inner ring is initially stationary. Therefore, the initial tangential interaction between the rotor and the inner ring is governed by a Coulomb stick–slip friction law rather than by the Palmgren running-torque model alone. When the required tangential force is lower than the static-friction limit, sticking is assumed; once this limit is exceeded, sliding occurs and the kinetic friction coefficient is used. The Palmgren friction torque is then applied to describe the internal bearing resistance after incipient rotation of the auxiliary bearing.

2.2. Thermal Model of the Auxiliary Bearing

Palmgren proposed a method for calculating the bearing friction torque [49], as shown in Equation (19):

$$M = M_0 + M_1 \quad (19)$$

In the above equation, M denotes the total friction torque of the bearing. M_0 represents the friction torque generated by the lubricant under no-load conditions, and its magnitude depends on the structural configuration of the auxiliary bearing, the properties of the lubricant, and the rotor speed during the drop process. M_1 represents the friction torque generated under load, mainly arising from elastic hysteresis of the material and frictional losses induced by differential sliding in the contact region.

The calculation formula for M_0 is given in Equation (20):

$$M_0 = \begin{cases} 10^{-7} f_0 (vn)^{2/3} d_m^3, & vn \geq 2000 \\ 160 \times 10^{-7} f_0 d_m^3, & vn < 2000 \end{cases} \quad (20)$$

It should be noted that the Palmgren equation is used here for the internal friction torque of a rolling bearing after the onset of rotation. The starting friction of the initially stationary auxiliary bearing is treated separately through the static-friction breakaway condition at the rotor–inner-ring contact interface. The lubricant kinematic viscosity appears in the no-load torque term M_0 ; therefore, the influence of lubricant temperature is reflected through the viscosity parameter. In the initial stage of rotor touchdown, the viscosity is taken as the value corresponding to the initial temperature of the bearing and lubricant, which reflects the increased viscous resistance of the lubricant during the early start-up stage.

The calculation formula for M_1 is given in Equation (21):

$$M_1 = P f_1 d_m \quad (21)$$

For angular contact ball bearings, the calculation formulas for f_1 and P_1 are given in Equations (22) and (23), respectively:

$$f_1 = z_1 \cdot \left(\frac{P_0}{C_0} \right)^{0.33} \quad (22)$$

$$P_1 = \begin{cases} F_r, & P_1 < F_r \\ F_a - 0.4F_r, & P_1 \geq F_r \end{cases} \quad (23)$$

Here, z_1 ranges from 0.0004 to 0.0006; a larger value is used for heavily loaded bearings, whereas a smaller value is used for lightly loaded bearings. P_0 denotes the equivalent static load of the bearing, C_0 denotes the basic static load rating of the bearing, F_r denotes the radial load, and F_a denotes the bearing impact force.

During auxiliary-bearing operation, spin slip occurs at the contacts between the rolling elements and the inner- and outer-ring raceways. At high rotational speeds, spin slip plays

an important role in the auxiliary bearing [50,51]. To simplify the calculation, the friction coefficient in the contact region is assumed to be constant. If the relative motion of the rolling element with respect to the inner and outer rings is represented solely by spin about the normal passing through the center of the contact surface, the spin friction torque can be calculated by Equation (24):

$$M_s = \frac{3\mu Q a \Sigma}{8} \quad (24)$$

In the above equation, μ denotes the coefficient of friction, Q denotes the contact load, Σ denotes the complete elliptic integral of the second kind, and a denotes the semi-major axis of the contact ellipse. Accordingly, the spin friction torques between the rolling element and the inner- and outer-ring raceways can be obtained as follows:

$$M_{si} = \frac{3\mu Q_i a_i \Sigma_i}{8} \quad (25)$$

$$M_{so} = \frac{3\mu Q_o a_o \Sigma_o}{8} \quad (26)$$

If the friction torque generated in the contact regions between the rolling elements and the inner and outer rings of the auxiliary bearing is assumed to be evenly distributed among all the rolling elements, the friction torque of each rolling element can be calculated using Equations (27) and (28):

$$M_{ij} = D_w / (z \cdot d_e) \cdot M \quad (27)$$

$$M_{oj} = D_w / (z \cdot d_i) \cdot M \quad (28)$$

In the above equations, the subscript j denotes the rolling element at a specific angular position, D_w denotes the rolling element diameter, z denotes the number of rolling elements, d_e denotes the outer-raceway diameter, and d_i denotes the inner-raceway diameter. Accordingly, the frictional heat generation rates between the rolling elements and the outer and inner rings can be expressed by Equations (29) and (30), respectively:

$$H_o = \sum_j H_{oj} = \sum_j (w_b M_{oj} + w_{so} M_{so}) \quad (29)$$

$$H_i = \sum_j H_{ij} = \sum_j (w_b M_{ij} + w_{si} M_{si}) \quad (30)$$

Here, w_b denotes the rolling angular velocity, and w_{so} and w_{si} denote the spin angular velocities of the rolling element in the outer-ring and inner-ring contact regions, respectively.

The formula for calculating the frictional heat generated by the axial impact between the rotor and the auxiliary bearing is given in Equation (31):

$$H_a = T_a \cdot w_{rel} \quad (31)$$

The formula for calculating the frictional heat generated by the radial impact between the rotor and the auxiliary bearing is given in Equation (32):

$$H_r = \vec{F}_f \cdot \vec{v}_r \quad (32)$$

In the above equation, F_f denotes the friction force vector at the radial contact interface, and v_r denotes the relative velocity between the rotor and the auxiliary bearing at the radial contact interface.

Based on the above analysis, the formula for calculating the total frictional heat generated in the auxiliary bearing during the rotor drop process is given in Equation (33):

$$H_{total} = \sum_{j=1}^2 (H_i^k + H_o^k + H_r^k + H_a^k) \quad (33)$$

Here, the subscript $k = 1, 2$ denotes the upper and lower auxiliary bearings, respectively.

2.3. Finite Element Model and Simulation Settings

In this study, the finite element simulation model was established using LS-DYNA to analyze the dynamic response and transient temperature-field evolution during the rotor-drop process after magnetic suspension failure. The three-dimensional model was first constructed and assembled using three-dimensional modeling software. To improve computational efficiency and reduce unnecessary resource consumption, chamfers and fillets that have limited influence on the overall dynamic and thermal responses were neglected. It should be noted that this simplification may introduce artificial sharp edges and local numerical stress concentrations near the bearing-ring edges. Therefore, the local stress maxima at these simplified edges were not used as quantitative indicators of bearing failure or structural strength in this study. The assembled model was then imported into LS-DYNA for material assignment, mesh generation, contact definition, and boundary-condition setup.

The inner and outer rings of the auxiliary bearing were made of GCr15, the rolling elements were made of Si₃N₄, and the rotor together with the other components was made of Q235. The mesh was generated primarily using tetrahedral elements, supplemented by hexahedral elements, with different meshing strategies adopted for different components [51]. Tetrahedral elements were used for the rotor, the inner and outer rings of the auxiliary bearing, and the rolling elements, whereas hexahedral elements were used for the touchdown pad, shaft sleeve, and similar components. In addition, local mesh refinement was applied to the contact surfaces of the inner- and outer-ring raceways. The contact boundary conditions were defined as follows. The rotor–inner-ring, rolling element–inner-ring, rolling element–outer-ring, and rolling element–rolling element interfaces were defined as frictional contact regions. These contact definitions were used to capture axial impact, radial rubbing, combined rolling–sliding motion, rolling-element interactions, and the corresponding frictional heat generation during rotor drop.

The dynamic boundary conditions were defined as follows. Gravity was applied along the negative Y direction of the global coordinate system, with a gravitational acceleration of 9806.6 mm/s². The outer ring of the auxiliary bearing was defined as a fixed support, with all six degrees of freedom constrained to represent the support effect of the bearing housing. No additional motion constraints were imposed on the rotor, inner rings, rolling elements, shaft sleeve, or other components; their motion was determined by gravity, the initial rotational speed, and interactions at the contact interfaces. The initial rotor speed was set to 20,000 r/min. The explicit dynamic-contact simulation was conducted for 0.15 s, with 15,000 time steps, and the results were recorded every 0.00001 s. For comparison with the experimental temperature measurement, the subsequent transient thermal-conduction calculation of the outer ring was further extended to 180 s using the frictional heat generated during the rotor-drop stage as the thermal input.

The thermal boundary conditions were defined as follows. The initial temperature field was set as a uniform ambient temperature of 22 °C. The thermal input during rotor drop mainly originated from axial contact friction, radial rubbing, combined rolling–sliding friction between the rolling elements and the inner/outer raceways, and frictional interactions between adjacent rolling elements. The frictional heat generated at the contact

interfaces was applied as a transient thermal input, and the spatial temperature-field distribution was obtained through heat conduction among the auxiliary-bearing components. The main parameters of the auxiliary bearing and the material properties of its components are listed in Tables 1 and 2.

Table 1. Main Parameters of the Auxiliary Bearing.

No.	Parameter	Value
1	Bearing Outer Diameter, D (mm)	90
2	Bearing Inner Diameter, d (mm)	65
3	Bearing Width, B (mm)	13
4	Rolling Element Diameter, D_w (mm)	7.94
5	Number of Rolling Elements/ Z	30

Table 2. Material Properties of the Components.

No.	Property	Bearing Ring	Rolling Element	Rotor and Other Components
1	Density (kg/m^3)	7810	3200	7850
2	Young's Modulus (GPa)	206	320	200
3	Poisson's Ratio	0.3	0.27	0.3
4	Thermal Conductivity ($\text{W}\cdot\text{m}^{-1}\cdot\text{K}^{-1}$)	46	20	60.5
5	Specific Heat Capacity ($\text{J}\cdot\text{kg}^{-1}\cdot\text{K}^{-1}$)	460	700	434
6	Coefficient of Thermal Expansion ($10^{-6}/^\circ\text{C}$)	11.5	3.2	12

3. Results and Discussion

3.1. Thermal Response of the Auxiliary Bearing

During the rotor drop process, the large rotational kinetic energy is rapidly converted into thermal energy through mechanical friction, resulting in a markedly non-uniform temperature field within the auxiliary bearing. Figure 4a–d show the transient temperature field distribution induced by severe contact friction after the high-speed rotor drops onto the auxiliary bearing.

As can be clearly observed from the overall sectional view in Figure 4a, the thermal response of the system exhibits a pronounced localized concentration feature. High-temperature regions are mainly distributed in the auxiliary bearing and its adjacent contact zones, whereas the rotor body and the structures far from the contact interface remain at relatively low temperatures. The maximum temperature of the system is approximately 169.59°C , while the minimum temperature is about 22.012°C . The large temperature gradient indicates that, during the rotor drop process, kinetic energy is converted into thermal energy through contact friction and is concentrated at the contact interface.

Figure 4b presents the temperature contour of the outer ring of the auxiliary bearing. It can be seen that the overall temperature of the outer ring is significantly lower than that of the inner ring and the rolling elements, with a maximum temperature of 94.785°C . The high-temperature regions are mainly located at the contact positions between the outer-ring raceway and the rolling elements. Along the circumferential direction, they exhibit a certain periodic distribution pattern, forming a continuous annular thermal band. By contrast, as shown in Figure 4c, the inner ring exhibits the highest peak temperature, which is identical to the maximum temperature of the entire system, reaching 169.59°C . This is because the high-speed rotor comes into contact with the inner ring of the auxiliary bearing, causing the inner ring to undergo rapid acceleration and severe sliding friction with the rotor, thereby generating a large amount of frictional heat at the contact surface. The generated heat is distributed circumferentially along the raceway, but the distribution is non-uniform. This non-uniformity is manifested in the shifting impact locations

and stress concentration caused by the rotor precession during the drop process. As shown in Figure 4d, the temperature of the rolling elements is also non-uniformly distributed, remaining higher than that of the outer ring and close to that of the inner ring overall. The maximum temperature is concentrated in the elliptical contact region between the ball and the inner-ring raceway, reaching 154.66 °C. Under high-speed drop conditions, the rolling elements are subjected not only to large contact loads but also to severe rolling–sliding combined motion and spin-friction-induced heat generation.

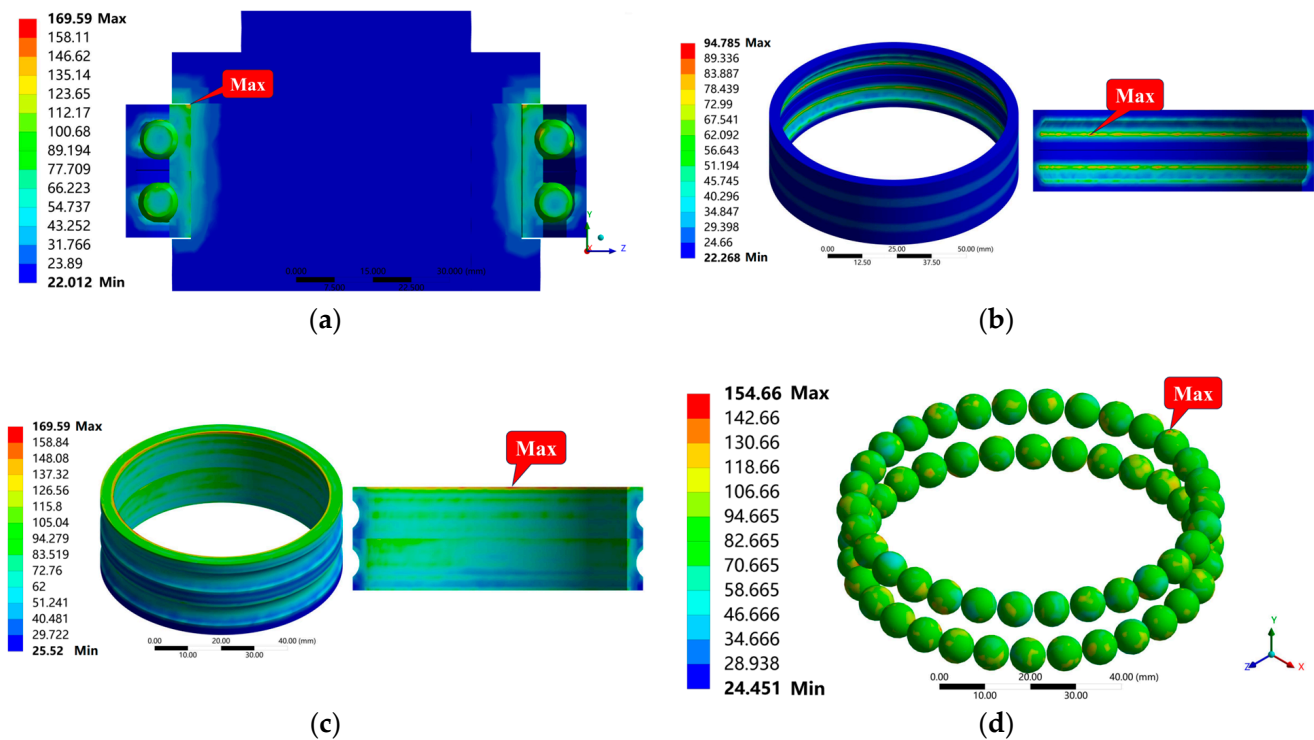


Figure 4. Transient temperature field contours of the auxiliary bearing system during the rotor drop process: (a) Overall temperature contour of the auxiliary bearing–rotor system; (b) Temperature contour of the outer ring of the auxiliary bearing; (c) Temperature contour of the inner ring of the auxiliary bearing; (d) Temperature contour of the rolling elements.

Figure 5 further compares the maximum, average, and minimum temperatures of the rotor–auxiliary bearing system and its individual components. Overall, the maximum temperature of each component is significantly higher than its average temperature, whereas the minimum temperature remains close to the initial temperature. This indicates that heat generated during the drop process is mainly concentrated in local contact and friction regions, and that the temperature rise exhibits pronounced spatial non-uniformity. A comparison between the upper and lower auxiliary bearings shows that the overall temperature level of the upper auxiliary bearing is significantly higher than that of the lower auxiliary bearing. This is consistent with the fact that, under the vertical configuration, the upper auxiliary bearing comes into contact earlier and is subjected to greater impact loads and frictional heat input. This asymmetric thermal response can be attributed to the load-transfer characteristics induced by the vertical rotor-drop configuration. After magnetic suspension failure, gravity drives the rotor to drop in the axial direction, and the initial axial kinetic energy is preferentially transferred to the upper auxiliary bearing at the first contact. Meanwhile, the rotor drop process is accompanied by initial eccentricity and rotor inclination, which causes the rotor mass center to deviate from the bearing geometric center. As a result, the contact load is not uniformly distributed between the upper and lower auxiliary bearings. The upper auxiliary bearing therefore receives a larger portion of

the initial impact energy and frictional work, leading to faster accumulation of frictional heat. Among the bearing components, the inner ring exhibits the highest temperature, followed by the rolling elements, while the outer ring remains the coolest. This indicates that frictional heat mainly accumulates at the rotor–inner ring contact interface and in the rolling-element contact regions, and gradually attenuates toward the outer ring. These results further demonstrate that the inner ring is the primary site of heat accumulation in the auxiliary bearing under rotor drop conditions and the key component most susceptible to thermal damage.

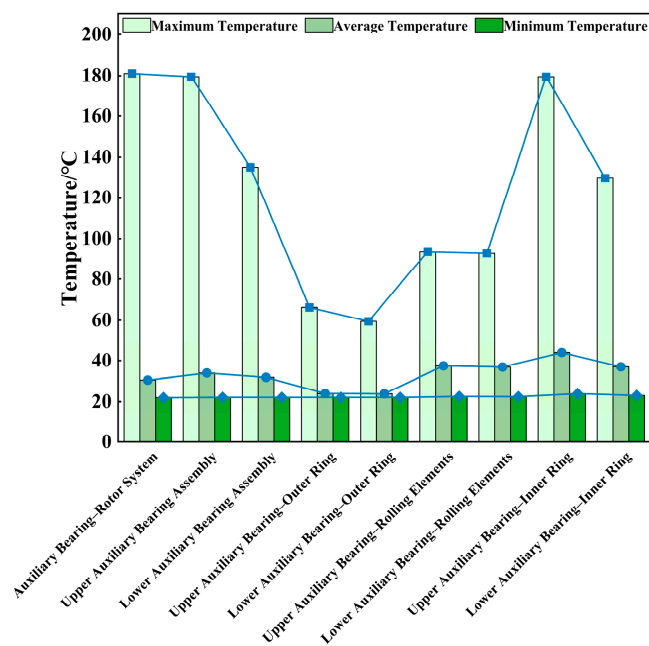


Figure 5. Comparison of the maximum, average, and minimum temperatures of the auxiliary bearing components.

Figure 6 illustrates the short-duration transient thermal response of the auxiliary bearing during the explicit rotor-drop and contact stage from 0 to 0.15 s. Figure 6a–c present the component-level temperature-rise trends of the inner rings, outer rings, and rolling elements during the initial impact and friction stage. It should be noted that Figure 6 is intended to compare the short-time thermal responses of different auxiliary-bearing components under the same dynamic-contact calculation. The direct model validation against the experimental outer-ring temperature measurement over the 0–180 s time scale is discussed in the following experimental-validation section.

As shown in Figure 6a, the temperatures of the inner rings of both the upper and lower auxiliary bearings rise sharply at approximately 0.03 s. The temperature-rise rate and final temperature of the inner ring of the upper auxiliary bearing are significantly higher than those of the lower auxiliary bearing. During the vertical rotor drop process, owing to the combined effects of gyroscopic motion, eccentricity, and rotor inclination during drop, the upper auxiliary bearing experiences greater impact and friction.

As shown in Figure 6b, the temperature-rise curves of the outer rings of the upper and lower auxiliary bearings exhibit generally similar overall trends, both characterized by an initial slow increase followed by sustained growth. During the initial stage of the drop process, the outer-ring temperature remains close to the initial temperature, showing only a slight increase. As impact and friction continue during the drop process, frictional heat accumulates continuously and is transferred to the outer ring, causing the outer-ring temperature to rise markedly after approximately 0.05 s. The temperature difference between the outer rings of the upper and lower auxiliary bearings is relatively

small, indicating that the thermal response of the outer ring is governed more by internal heat conduction than by local contact-state variations, to which it is comparatively less sensitive.

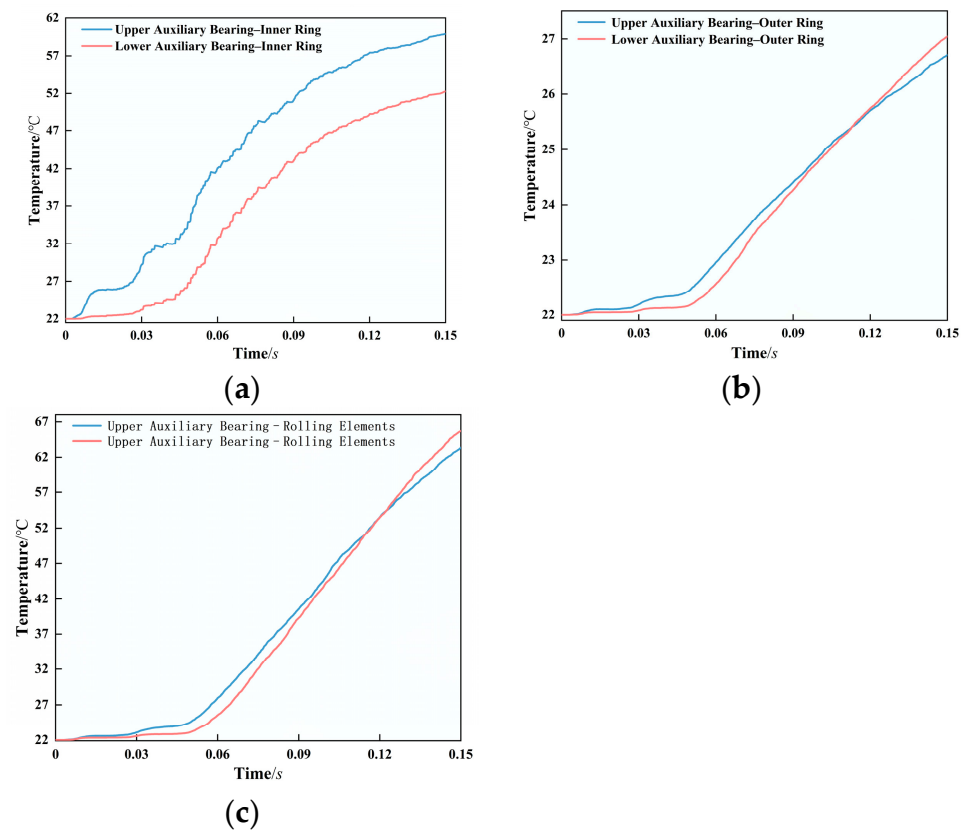


Figure 6. Short-duration temperature-rise curves of the auxiliary bearing during the explicit rotor-drop and contact stage from 0 to 0.15 s: (a) Temperature-rise curves of the inner rings of the upper and lower auxiliary bearings; (b) Temperature-rise curves of the outer rings of the upper and lower auxiliary bearings; (c) Temperature-rise curves of the rolling elements of the upper and lower auxiliary bearings.

As shown in Figure 6c, the temperature-rise trend of the rolling elements is similar to that of the inner ring, whereas the overall temperature level is significantly higher than that of the outer ring. During the initial stage of the drop process, the temperature variation of the rolling elements is relatively small. As continuous impact and friction occur between the rotor and the auxiliary bearing, the temperature of the rolling elements rises rapidly, and the average temperature exceeds 60 °C within 0.15 s. This indicates that the rolling elements are subjected not only to large alternating contact loads but also to significant rolling–sliding combined motion and frictional heat generation. A comparison of the temperature-rise curves of the rolling elements in the upper and lower auxiliary bearings shows that, during the initial stage, the temperature of the rolling elements in the upper auxiliary bearing is slightly higher than that in the lower auxiliary bearing. This is consistent with the fact that the upper auxiliary bearing comes into contact earlier and is subjected to stronger impact and friction. After approximately 0.12 s, however, the temperature of the rolling elements in the lower auxiliary bearing becomes slightly higher than that in the upper auxiliary bearing. This suggests that, in the later stage, the rolling elements in the lower auxiliary bearing may be more strongly affected by spin slip and local sliding friction, leading to further accumulation of frictional heat in local regions.

3.2. Dynamic Response of the Auxiliary Bearing

The contact process between the rotor and the auxiliary bearing involves a complex dynamic behavior characterized by nonlinear impact, friction, and multibody motion. Figure 7 illustrates the impact-force response during the process in which the rotor drops and comes into contact with the auxiliary bearing.

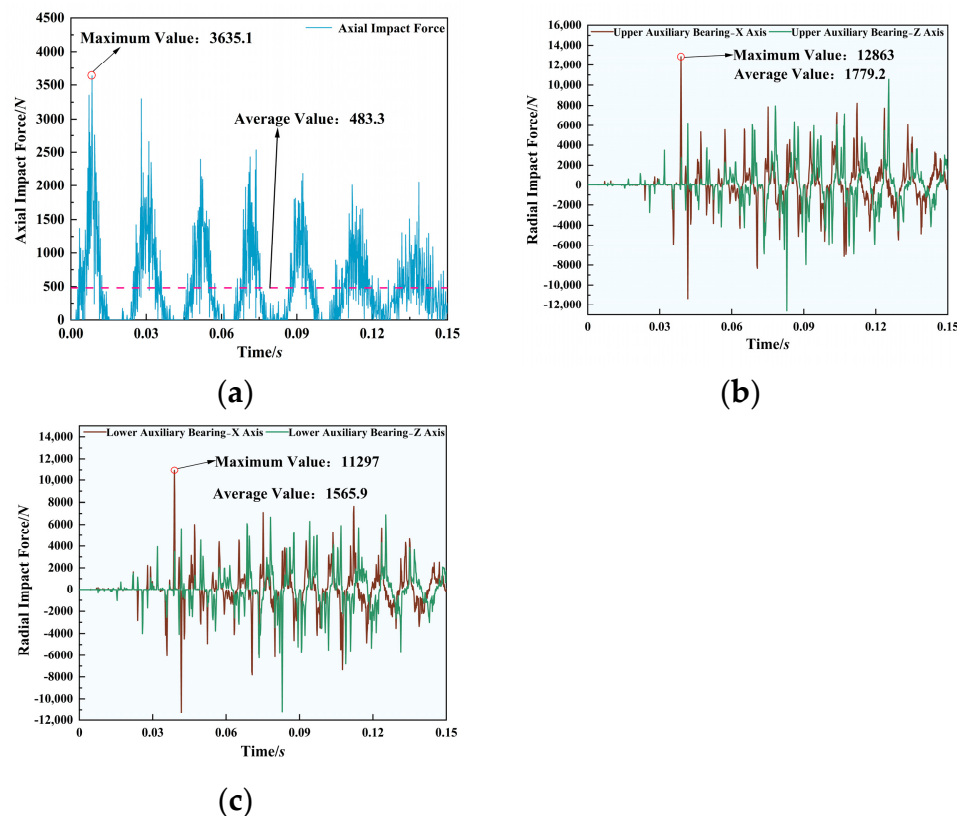


Figure 7. Impact force response of the auxiliary bearing during the rotor drop process: (a) Axial impact force of the auxiliary bearing; (b) Radial impact force of the upper auxiliary bearing; (c) Radial impact force of the lower auxiliary bearing.

Figure 7a shows the axial impact force between the rotor and the auxiliary bearing. At the instant of system instability, the rotor falls freely under its own weight and undergoes axial contact impact with the auxiliary bearing. The axial impact force exhibits pronounced periodic impulsive behavior, with the peak occurring at the first contact between the rotor and the auxiliary bearing during the drop process. The peak value reaches 3635.1 N, which is approximately eight times the rotor weight (450 N), whereas the overall average level is about 483.3 N, close to the rotor weight. As the rotor undergoes multiple intermittent axial impacts and rebounds with the inner ring of the upper auxiliary bearing, the impact amplitude gradually decreases under the effects of structural damping, contact friction, and other dissipative mechanisms, exhibiting a typical nonlinear attenuation process.

Figures 7b and 7c correspond to the radial impact forces of the upper and lower auxiliary bearings, respectively. In contrast to the axial impact force, the radial impact force is characterized by pronounced transient oscillations. At the initial stage of rotor drop, the rotor has not yet undergone large deflection within the protective clearance, and the radial contact force therefore remains relatively low overall. As the complex rotor motions, including deflection and whirl, develop and superimpose, the radial load acting on the auxiliary bearing increases sharply after approximately 0.04 s. For the upper auxiliary bearing, the maximum radial impact force reaches approximately 12,863 N, with an average

value of 1779.2 N. This peak radial impact load is far greater than the rotor weight and is 3.6 times the maximum axial impact force. By contrast, the maximum radial impact force of the lower auxiliary bearing is 11,297 N. The difference in radial impact load further confirms the asymmetric contact mechanism. Thus, the peak radial load of the upper auxiliary bearing is approximately 13.9% higher than that of the lower auxiliary bearing. This difference is mainly caused by the combined effects of gravity-induced axial impact, rotor eccentricity, rotor inclination during drop, and gyroscopic motion. Under high-speed rotation, the rotor retains considerable angular momentum. Once the rotor tilts during the drop process, gyroscopic torque promotes precession and whirl motion, causing the rotor to rub more severely against the upper auxiliary bearing. Consequently, the upper auxiliary bearing is subjected to both axial impact and intensified radial rubbing, resulting in a larger transient contact load.

Figure 8 illustrates the dynamic response during the rotor drop process, including the rotational speed responses of the rotor and the inner ring of the auxiliary bearing (Figure 8a), the axial displacement of the rotor (Figure 8b), and the displacement trajectory of the rotor center of mass during the drop process (Figure 8c).

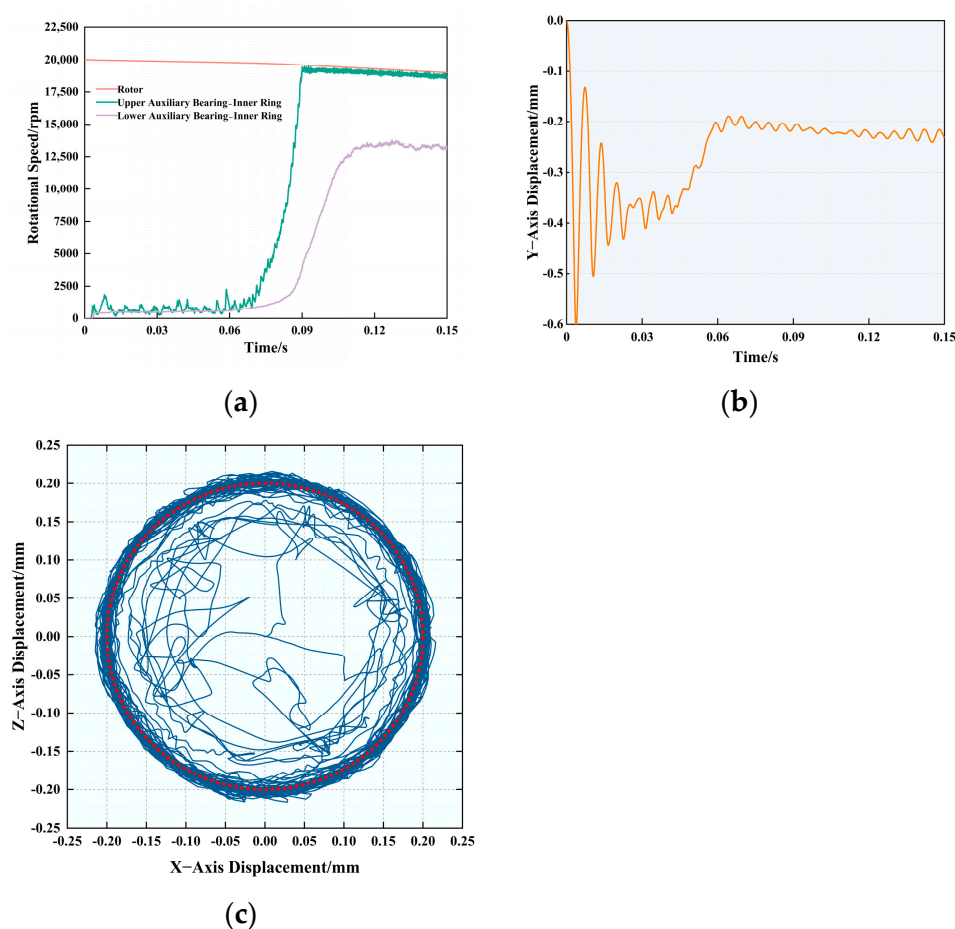


Figure 8. Transient response of the rotor during touchdown on the auxiliary bearing: (a) Rotational speeds of the rotor and inner ring; (b) Axial displacement trajectory of the rotor center of mass; (c) Radial displacement trajectory of the rotor center of mass.

As can be seen from the speed response curve in Figure 8a, owing to the large rotational inertia of the rotor, its rotational speed decreases only gradually during the drop process, with an overall variation of limited magnitude. However, the acceleration responses of the inner rings of the upper and lower auxiliary bearings show significant differences. The inner ring of the upper auxiliary bearing remains at a relatively low speed during the initial

stage of rotor drop. However, at approximately 0.05 s, the contact state changes abruptly, and under the combined action of intense impact and friction, the inner ring accelerates rapidly and approaches the rotor speed within a short time, reaching about 19,000 r/min. By contrast, the inner ring of the lower auxiliary bearing exhibits a certain delay in the onset of rotation, and a certain degree of slip persists throughout the acceleration process. Subsequently, the speed difference between the rotor and the inner rings of the upper and lower auxiliary bearings gradually decreases, and the three bodies ultimately tend toward synchronous rotation. This asymmetric speed response indicates that, during the rotor drop process, changes in rotor attitude and redistribution of contact loads cause the upper auxiliary bearing to come into contact with the rotor earlier and to sustain higher transient loads and frictional power input, making it more susceptible to impact loading and thermal shock.

The faster temperature rise of the upper auxiliary bearing is closely related to its stronger frictional power input. This rapid spin-up indicates that a large amount of tangential work is transferred from the rotor to the upper auxiliary bearing through the contact interface. Since the frictional heat-generation rate is governed by the normal contact load and the relative sliding velocity, the earlier contact, larger impact load, and stronger tangential driving effect of the upper auxiliary bearing jointly lead to higher frictional heat generation. In addition, the inner ring is the component directly subjected to rotor contact, while heat transfer from the inner ring to the outer ring and bearing housing is limited by thermal resistance. Therefore, heat accumulates more rapidly near the inner ring and rolling-element contact regions, explaining why the upper auxiliary bearing exhibits a higher temperature-rise rate and a greater thermal-failure risk.

Figures 8b and 8c show the axial and radial displacements of the rotor during the drop process, respectively. Driven by gravity, the rotor falls rapidly and first impacts the inner ring of the upper auxiliary bearing at approximately 0.005 s, with the maximum transient axial displacement reaching -0.6 mm. Thereafter, the rotor enters a high-frequency impact-rebound oscillation stage. Under the damping dissipation effect of the auxiliary bearing, the oscillation amplitude gradually decays and finally stabilizes at approximately -0.2 mm around 0.06 s. At the initial stage of rotor drop, the radial displacement trajectory of the rotor center of mass exhibits irregular and chaotic characteristics. As the contact state intensifies, the trajectory gradually evolves into high-frequency periodic whirl along the protective clearance, indicating that the rotor has entered a full-annular rubbing state. This is also the reason why the inner ring of the upper auxiliary bearing accelerates rapidly at this stage.

The rotor drop process is not an ideal vertical descent, but it is accompanied by tilted motion and lateral swinging. During the drop process, the rotor first impacts the upper auxiliary bearing, and the impact kinetic energy at the moment of contact is directly transferred to it. As a result, the upper auxiliary bearing is subjected to greater impact and friction forces than the lower auxiliary bearing, and severe frictional heating is instantaneously generated at the upper surface of the inner ring of the upper auxiliary bearing. Subsequently, owing to factors such as rotor eccentricity and unbalance, the rotor further collides with the auxiliary bearing, leading to increased contact forces between the rolling elements and the inner and outer rings. At the same time, the sliding friction between the rotor and the auxiliary bearing increases, thereby driving the auxiliary bearing into rotation. The total load acting on the upper auxiliary bearing, including both axial and radial loads, is much greater than that on the lower auxiliary bearing. Consequently, the total heat generation of the upper auxiliary bearing is higher, and its average temperature exceeds that of the lower auxiliary bearing.

The heat sources of the auxiliary bearing mainly originate from rolling-friction-induced heat generation and sliding-friction-induced heat generation. During rotor–auxiliary bearing contact, the large kinetic energy of the rotor causes the auxiliary bearing to accelerate rapidly from rest to a high rotational speed. The inner ring is located at the high-speed contact end and is subjected to greater impact loading. In the contact region between the inner ring and the rotor, the relative sliding velocity is higher and the friction is more severe. In addition, outward heat conduction from the inner ring is restricted, causing heat to accumulate near the raceway and produce the highest temperature peak. The rolling elements are located in the primary load-bearing contact region and have a relatively small heat capacity, making them more susceptible to rapid heating. However, because heat can be dissipated to both the inner and outer rings, their peak temperature remains lower than that of the inner ring. By contrast, the outer ring receives relatively less direct heat input, and its interference fit with the bearing housing forms an efficient heat-conduction boundary. As a result, the generated heat can be rapidly conducted to the bearing housing, and the outer ring therefore exhibits the lowest temperature.

The above results demonstrate that the asymmetric response of the auxiliary bearings is a typical dynamic–thermal coupling phenomenon rather than a purely mechanical or thermal effect. In the early stage of rotor drop, the load distribution is mainly governed by gravity-driven axial impact and the initial contact sequence. As the rotor motion evolves, the coupled effects of eccentricity, rotor inclination, gyroscopic motion, and radial rubbing change the contact state and redistribute the frictional work between the upper and lower auxiliary bearings. Therefore, the thermal response is essentially determined by the time-varying contact load, relative sliding velocity, and heat-transfer path.

4. Experimental Validation

To investigate the evolution of thermal characteristics and the dynamic response of the auxiliary bearing in a vertical magnetic suspension system under instability-induced drop conditions, and to experimentally validate the established simulation model, an auxiliary bearing test platform for the vertical magnetic suspension system was constructed, as shown in Figure 9.

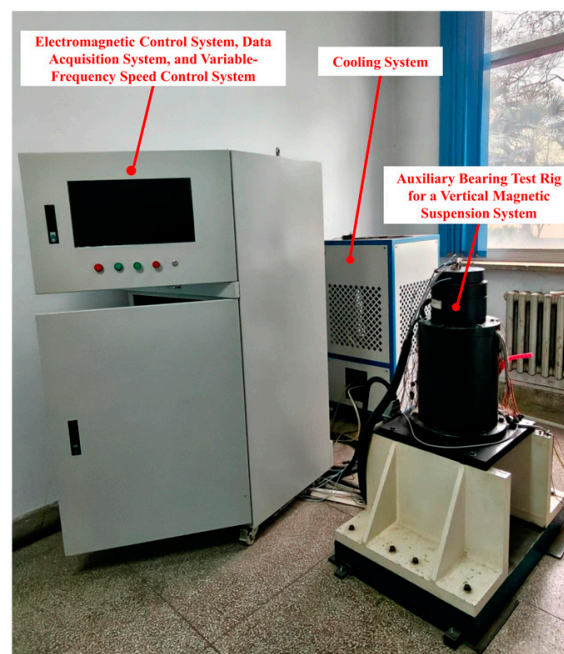


Figure 9. Auxiliary bearing test platform for the vertical magnetic suspension system.

The experimental platform mainly consists of the test rig body, an electromagnetic control system, a variable-frequency speed regulation system, an experimental monitoring system, a cooling system, and auxiliary equipment. Using sensors such as vibration sensors, accelerometers, speed sensors, miniature pressure sensors, displacement signal sensors, eddy-current sensors, and temperature sensors, data such as rotor vibration, rotational speed, maximum axial impact force, maximum radial impact force, displacement trajectory, and the outer-ring temperature of the upper auxiliary bearing can be collected in real time.

The auxiliary bearing used in the experiment was a 71913 full-complement angular-contact ball bearing manufactured by Luoyang Bearing Research Institute Co., Ltd. (Luoyang, China), with P4 accuracy. The bearing rings were made of GCr15 bearing steel, and the rolling elements were made of silicon nitride ceramic, Si_3N_4 . The bearings were mounted in a face-to-face configuration, as shown in Figure 10. The main system parameters are listed in Table 3.



Figure 10. Auxiliary bearing used in the experiment.

Table 3. System Parameters.

No.	Parameter	Value
1	Rotor Mass (kg)	45
2	Rotor Length (mm)	765
3	Mean Rotor Diameter (mm)	45
4	Axial Protective Clearance Between the Rotor and Auxiliary Bearing (mm)	0.2
5	Radial Protective Clearance Between the Rotor and Auxiliary Bearing (mm)	0.2
6	Initial Rotor Speed at Drop (rpm)	20,000

Figure 11 shows the main experimental and validation results obtained during the rotor-drop test, including the comparison between the experimental and simulated temperatures, the axial impact force, the radial impact force, and the displacement trajectory of the rotor mass center. As shown in Figure 11a, the experimental and simulated temperatures are compared over the same time span of 0–180 s. The experimental temperature increases gradually after rotor drop and reaches a maximum value of 41.361 °C, followed by a slow cooling stage. The simulated temperature rises more rapidly and reaches a maximum value of 50.107 °C, after which it also decreases gradually.

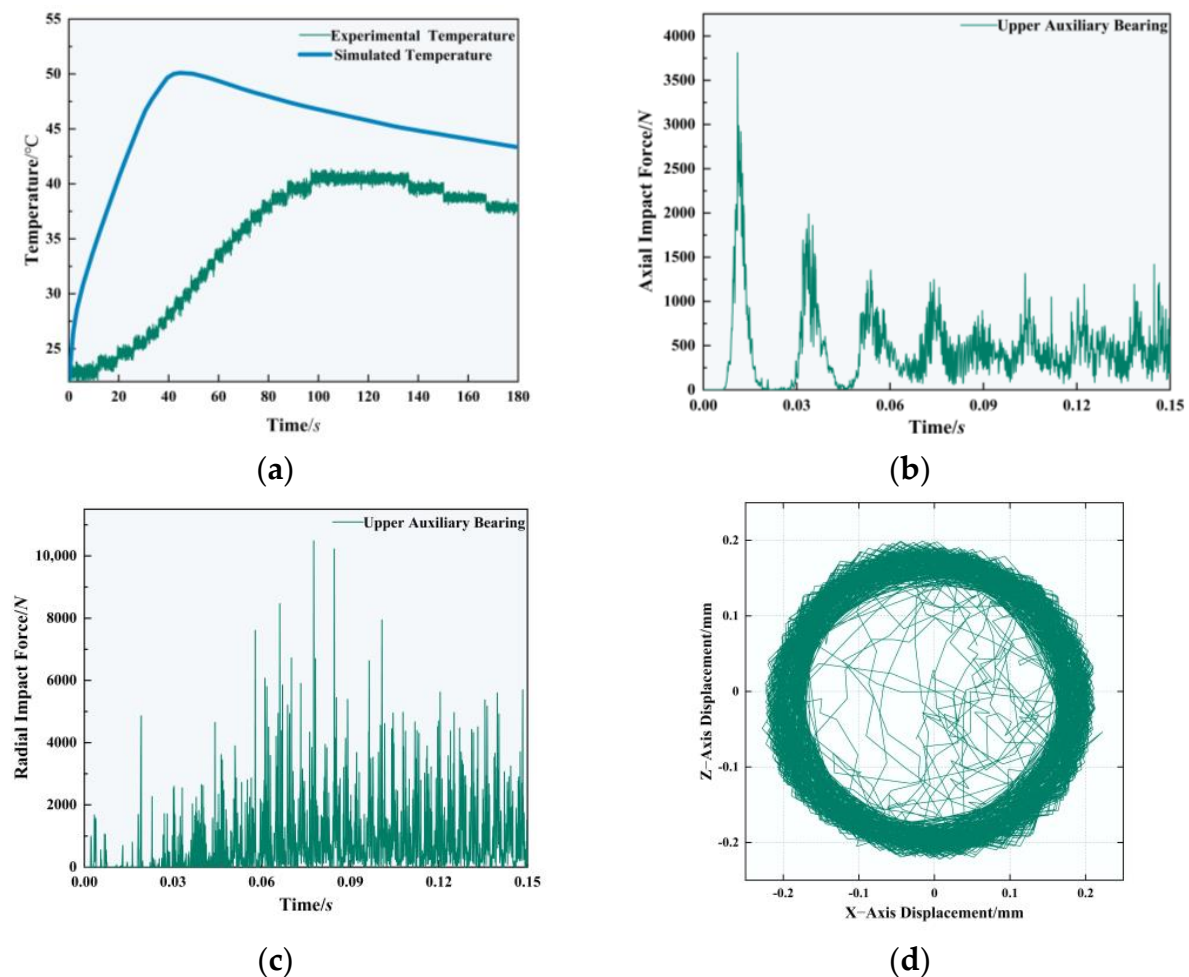


Figure 11. Experimental responses of the upper auxiliary bearing and rotor during the rotor-drop test: (a) Comparison between the experimental and simulated outer-ring temperatures of the upper auxiliary bearing; (b) Axial impact force acting on the upper auxiliary bearing; (c) Radial impact force acting on the upper auxiliary bearing; (d) Displacement trajectory of the rotor mass center in the X–Z plane.

Although the simulated peak temperature is higher than the experimental value, the two curves exhibit a consistent overall trend, namely a temperature-rise stage followed by a slow cooling stage. The difference between the experimental and simulated curves can be attributed to the temperature measurement location, the heat-transfer path from the bearing components to the sensor position, environmental heat dissipation, sensor response delay, and the idealization of the thermal boundary conditions in the simulation model. Therefore, the temperature comparison indicates that the proposed model can reasonably predict the overall evolution trend of the outer-ring temperature during the rotor-drop process.

Figure 11b, c show the measured axial and radial impact-force responses of the upper auxiliary bearing. The impact-force signals exhibit pronounced transient and intermittent characteristics during the rotor-drop process. The measured maximum axial impact force is 3811.7 N, and the measured maximum radial impact force acting on the upper auxiliary bearing is 11,146 N. Figure 11d presents the displacement trajectory of the rotor mass center in the X–Z plane. The trajectory shows that the rotor motion evolves into a bounded annular rubbing state after touchdown, which is consistent with the simulated radial motion characteristics.

To quantitatively validate the proposed model, the measured impact-force results were compared with the corresponding simulation predictions, as summarized in Table 4. The simulated maximum axial impact force is 3635.1 N, compared with the measured value of 3811.7 N, giving a relative error of 4.63%. The simulated maximum radial impact force of the upper auxiliary bearing is 12,863 N, compared with the measured value of 11,146 N, corresponding to a relative error of 15.4%. These comparisons indicate that the model can reasonably predict the main axial and radial impact-load responses during the rotor-drop process. In addition, the temperature comparison in Figure 11a shows that the simulated and experimental outer-ring temperatures exhibit a consistent overall trend over the same time span of 0–180 s. Together with the measured displacement trajectory and post-test damage observations, the experimental results provide direct support for the validity of the dynamic–thermal model.

Table 4. Comparison between simulation and experiment.

Quantity	Simulation	Experiment	Relative Error
Maximum axial impact force	3635.1 N	3811.7 N	4.63%
Maximum radial impact force	12,863 N	11,146 N	15.4%

Figure 12 presents the surface morphologies of the key components of the auxiliary bearing after the drop test. Figure 12a shows the post-test surface morphology of the outer-ring raceway of the auxiliary bearing. Distinct wear marks, local material adhesion, and continuous thermal discoloration bands can be observed along the rolling contact region of the outer-ring raceway, indicating that severe impact friction occurred between the rolling elements and the outer-ring raceway after rotor drop. This observation is generally consistent with the simulated distribution of wear and thermal discoloration on the outer-ring raceway shown in Figure 4b.

Figure 12b–d show the post-test surface morphologies of the inner ring of the auxiliary bearing. As shown in Figure 12b, extensive thermal discoloration appears on the upper surface of the inner ring, accompanied by obvious surface scratches and friction marks, indicating that this surface experienced severe friction and a significant temperature rise during the rotor drop process. As shown in Figure 12c, the inner-ring raceway exhibits significant thermal damage, with continuous rainbow-colored oxidation bands, obvious scuffing, and local material spalling and adhesion observed in some areas. As shown in Figure 12d, distinct banded wear marks and local darkened areas are present on the inner surface of the inner ring after the test. This indicates that, in addition to the raceway contact zone, the inner ring was also subjected to additional friction caused by rotor tilting, eccentricity, and transient microslip during the drop process. These observations agree well with the location of the high-temperature bands and the temperature distribution pattern of the inner ring shown in the simulation results in Figure 4c, indicating that the inner ring is the primary site of heat accumulation in the auxiliary bearing under rotor drop conditions. The rainbow-coloured oxidation bands on the inner-ring raceway provide qualitative evidence of local high-temperature oxidation, supporting the simulated heat accumulation near the rotor–inner-ring contact region. However, they were not used for exact temperature estimation because oxidation colour is affected by surface condition, oxidation duration, atmosphere, lubricant residues, and cooling history.

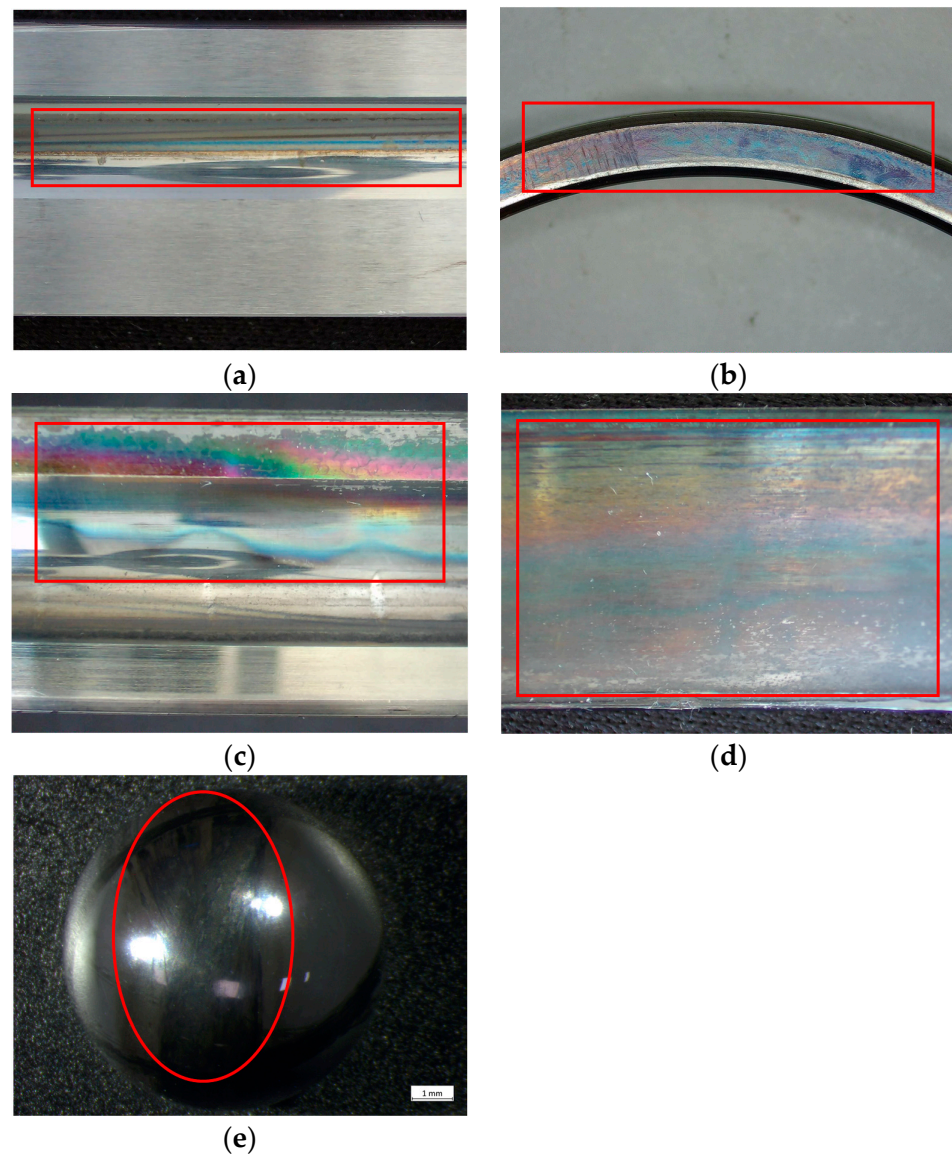


Figure 12. Surface morphologies of the key components of the auxiliary bearing after the experiment: (a) Outer-ring raceway of the auxiliary bearing; (b) Upper surface of the inner ring of the auxiliary bearing; (c) Inner-ring raceway of the auxiliary bearing; (d) Inner surface of the inner ring of the auxiliary bearing; (e) Surface of the rolling element of the auxiliary bearing.

As shown in Figure 12e, banded scuff marks and slight thermal discoloration appear on the surface of the rolling elements, indicating that the rolling elements were subjected to alternating contact loads between the inner and outer rings, together with sliding friction and spin friction. Combined with the simulation results in Figure 4d, it can be inferred that the overall temperature level of the rolling elements is higher than that of the outer ring but lower than that of the inner ring, and that their local high-temperature regions are mainly distributed at positions that frequently contact the raceways. This pattern is in good agreement with the distribution of local wear and thermal discoloration observed on the rolling-element surface in Figure 12e.

In summary, the experimental results provide both quantitative and qualitative support for the proposed model. The quantitative comparison of the axial and radial impact-force peaks shows that the model can reasonably predict the main transient impact-load response during rotor drop. In addition, the temperature comparison in Figure 11a demonstrates that the simulated and experimental outer-ring temperatures exhibit a consistent

overall evolution trend over the same time span of 0–180 s. The measured rotor mass-center trajectory further supports the predicted radial rubbing behavior. Moreover, the post-test thermal discoloration regions, wear locations, and damage severity are generally consistent with the simulated high-temperature regions. These results demonstrate that the proposed dynamic–thermal model can reasonably describe the main dynamic response and thermal damage distribution of the auxiliary bearing during rotor drop.

5. Conclusions

This study focuses on rotor drop events induced by instability in a vertical magnetic suspension system. Dynamic and thermodynamic models of the rotor–auxiliary bearing system were established. Finite element simulations and rotor drop experiments were then conducted to investigate the impact load, rotational speed response, and transient temperature field evolution of the auxiliary bearing during the drop process. The main conclusions are as follows:

1. During the rotor drop process, the rotor's kinetic energy is rapidly converted into thermal energy through contact impact and friction. As a result, the temperature field of the auxiliary bearing exhibits a non-uniform distribution, with an overall temperature gradient that decreases from the inner region to the outer region. Among the bearing components, the inner ring is the primary region of heat accumulation, with a maximum temperature of 169.59 °C. The rolling elements rank second, reaching a maximum temperature of 154.66 °C, whereas the outer ring shows the lowest temperature, with a maximum of 94.785 °C. These results indicate that the inner ring and the rolling elements are the regions most susceptible to thermal damage in the auxiliary bearing.
2. Under the vertical structural configuration, the rotor drop process exhibits pronounced asymmetric contact characteristics due to the combined effects of gravity, rotor eccentricity, and gyroscopic effects. Compared with the lower auxiliary bearing, the upper auxiliary bearing comes into contact earlier and is subjected to higher impact loads and greater frictional heat input. Consequently, it exhibits a faster temperature rise and a higher overall temperature level, indicating that the upper auxiliary bearing is the critical component most prone to thermal failure under rotor drop conditions in a vertical magnetic suspension system.
3. The impact response of the auxiliary bearing during the rotor drop process exhibits pronounced transient nonlinear characteristics. The axial impact force appears as periodic impulsive pulses, with the first contact peak reaching 3635.1 N, approximately eight times the rotor weight, and then gradually decaying because of energy dissipation induced by damping and friction. By contrast, the radial impact force exhibits much more intense transient oscillations, with a peak significantly higher than that of the axial impact force. For the upper auxiliary bearing, the maximum radial impact force reaches 12,863 N, approximately 3.6 times the maximum axial impact force, indicating that radial impact is the key dynamic factor affecting the structural safety and service reliability of the auxiliary bearing.
4. The rotor drop is accompanied by coupled tilting, swinging, and whirling motions rather than an ideal vertical descent, which governs the asynchronous speed-up and asymmetric loading of the upper and lower auxiliary bearings. After contact, the inner ring of the upper auxiliary bearing is rapidly accelerated by the rotor and reaches approximately 19,000 r/min, close to the rotor speed, at about 0.05 s, gradually entering a full-circumference rubbing state. In contrast, the inner ring of the lower auxiliary bearing exhibits a pronounced start-up delay and slip.

5. The experimental measurements provide quantitative support for the proposed model. The measured maximum axial impact force is 3811.7 N, compared with the simulated value of 3635.1 N, giving a relative error of 4.63%. The measured maximum radial impact force acting on the upper auxiliary bearing is 11,146 N, compared with the simulated value of 12,863 N, giving a relative error of 15.4%. In addition, the measured displacement trajectory of the rotor mass center shows a bounded annular rubbing motion, which is consistent with the simulated radial motion characteristics. The post-test thermal discoloration regions, wear locations, and damage severity are also generally consistent with the simulated high-temperature regions and thermal damage distribution. These results indicate that the proposed model can reasonably capture the main impact-load response, radial rubbing behavior, frictional heat-generation mechanism, and thermal damage distribution of the auxiliary bearing during rotor drop.
6. From an engineering perspective, the upper auxiliary bearing should be regarded as the priority component for thermal-shock-resistant design in vertical magnetic suspension systems. Because it is subjected to higher impact loads and greater frictional heat input, materials or coatings with improved high-temperature resistance, scuffing resistance, and low-friction characteristics should be preferentially used for the inner ring, rolling elements, and rotor–inner-ring contact region. In addition, the heat-dissipation path around the upper auxiliary bearing should be strengthened by improving the thermal conductivity of the bearing housing, optimizing the heat-transfer boundary, or introducing local cooling structures. Considering the asymmetric response of the upper and lower auxiliary bearings, differentiated protection strategies are also recommended, including stricter monitoring thresholds for the upper bearing and separate optimization of the axial and radial protective clearances, support stiffness, and damping parameters.

Author Contributions: Conceptualization, X.P. and C.J.; methodology, Z.S.; software, D.Z.; validation, X.P., C.J. and D.Z.; resources, Z.S.; data curation, A.W.; writing—original draft preparation, C.J.; writing—review and editing, X.P. and C.J.; visualization, A.W. and K.W.; supervision, X.P.; funding acquisition, K.W. All authors have read and agreed to the published version of the manuscript.

Funding: This research was funded by the Zhejiang Provincial Leading Innovation and Entrepreneurship Team under the Advanced Manufacturing Team for Robot Bearings, grant number 2024R01014. The APC was funded by Xiaoxu Pang.

Data Availability Statement: All data included in this study are available upon request by contact with the corresponding author.

Acknowledgments: The authors have reviewed and edited the output and take full responsibility for the content of this publication.

Conflicts of Interest: Author Zhixin Shen was employed by Luoyang Bearing Research Institute Co., Ltd. Author Kaili Wang was employed by Zhejiang Pinnuo Machinery Co., Ltd. The remaining authors declare that the research was conducted in the absence of any commercial or financial relationships that could be construed as a potential conflict of interest.

Nomenclature

Symbol	Description	Symbol	Description
A	Axial contact point between the touchdown pad and the inner ring	O_r	Geometric center of the rotor
B	Bearing width	O_c	Center of mass of the rotor
C_a	Axial contact damping	P_a	Contact stress at the axial contact interface
C_r	Radial contact damping	P_0	Equivalent static load of the bearing

Symbol	Description	Symbol	Description
C_0	Basic static load rating of the bearing	P_1	Equivalent dynamic load of the bearing
D	Bearing outer diameter	Q	Contact load
D_w	Rolling element diameter	Q_i, Q_o	Contact loads at the inner- and outer-ring raceway contacts
d	Bearing inner diameter	R	Contact radius or equivalent radius used in the rotational equations
d_m	Bearing pitch diameter	R_{b1}	Radius of the inner surface of the inner ring
d_i	Inner-raceway diameter	R_{b2}	Radius of the inner-ring shoulder
d_e	Outer-raceway diameter	r	Radial coordinate in the polar coordinate system
e	Contact-type-dependent exponent in the Hertzian contact model	T_a	Axial friction torque
F_a	Axial impact force	T_b	Internal bearing friction torque
F_r	Radial contact force	T_{b1}, T_{b2}	Internal friction torques of the upper and lower auxiliary bearings
F_t	Tangential contact force,	T_0	No-load internal bearing torque
F_{t1}, F_{t2}	Tangential friction forces of the upper and lower auxiliary bearings	T_1	Load-dependent internal bearing torque
F_f	Friction force vector at the radial contact interface	v	Kinematic viscosity of the lubricant
f_0	Coefficient related to bearing type and lubrication method	v_r	Relative velocity at the radial contact interface
f_1	Coefficient related to bearing type and applied load	w_b	Rolling angular velocity of the rolling element
H	Frictional heat-generation rate	w_{si}, w_{so}	Spin angular velocities at the inner- and outer-ring contacts
H_i, H_o	Frictional heat-generation rates at the inner- and outer-ring contacts	z	Number of rolling elements
H_a, H_r	Heat generated by axial and radial impact friction	z_1	Empirical coefficient used in the calculation of f_1
H_{total}	Total frictional heat-generation rate of the auxiliary bearing	α	Rotor whirl angle
I_p	Polar moment of inertia of the rotor	$\dot{\alpha}$	Rotor whirl angular velocity
I_b	Moment of inertia of the auxiliary bearing inner ring	$\dot{\theta}_b$	Angular velocity of the auxiliary-bearing inner ring
K_a	Axial contact stiffness		Relative distance between O_b and O_r , representing the rotor motion radius in the horizontal plane
K_r	Radial contact stiffness	ρ	Spin angle of the rotor
M	Total friction torque of the bearing	$\dot{\gamma}$	Spin angular velocity of the rotor
M_0	No-load friction torque caused by lubricant resistance	δ_a	Axial impact depth
M_1	Load-dependent friction torque	δ_r	Radial impact depth
M_s	Spin friction torque	μ	Friction coefficient
M_{si}, M_{so}	Spin friction torques at the inner- and outer-ring raceway contacts	μ_s	Static friction coefficient
M_{ij}, M_{oj}	Friction torque assigned to the (j)-th rolling element at the inner- and outer-ring contacts	μ_d	Kinetic friction coefficient
n	Bearing rotational speed	Σ	Complete elliptic integral of the second kind
O_b	Geometric center of the auxiliary bearing		

References

1. Maslen, E.H.; Schweitzer, G. (Eds.) *Magnetic Bearings: Theory, Design, and Application to Rotating Machinery*; Springer: Berlin/Heidelberg, Germany, 2009.
2. Supreeth, D.K.; Bekinal, S.I.; Chandranna, S.R.; Doddamani, M. A Review of Superconducting Magnetic Bearings and Their Application. *IEEE Trans. Appl. Supercond.* **2022**, *32*, 1–15. [CrossRef]
3. Breńkacz, Ł.; Witanowski, Ł.; Drosińska-Komor, M.; Szewczuk-Krypa, N. Research and Applications of Active Bearings: A State-of-the-Art Review. *Mech. Syst. Signal Process.* **2021**, *151*, 107423. [CrossRef]
4. DellaCorte, C. Oil-Free Shaft Support System Rotordynamics: Past, Present and Future Challenges and Opportunities. *Mech. Syst. Signal Process.* **2012**, *29*, 67–76. [CrossRef]
5. Agrawal, G.L. *Foil Air/Gas Bearing Technology—An Overview*; ASME: Orlando, FL, USA, 1997.

6. Martowicz, A.; Roemer, J.; Kantor, S.; Zdziebko, P.; Żywica, G.; Bagiński, P. Gas Foil Bearing Technology Enhanced with Smart Materials. *Appl. Sci.* **2021**, *11*, 2757. [\[CrossRef\]](#)
7. Li, X.; Palazzolo, A. A Review of Flywheel Energy Storage Systems: State of the Art and Opportunities. *J. Energy Storage* **2022**, *46*, 103576. [\[CrossRef\]](#)
8. Hutterer, M.; Schrödl, M. Control of Active Magnetic Bearings in Turbomolecular Pumps for Rotors with Low Resonance Frequencies of the Blade Wheel. *Lubricants* **2017**, *5*, 26. [\[CrossRef\]](#)
9. Jiang, S.; Wang, H.; Wen, S. Flywheel Energy Storage System with a Permanent Magnet Bearing and a Pair of Hybrid Ceramic Ball Bearings. *J. Mech. Sci. Technol.* **2014**, *28*, 5043–5053. [\[CrossRef\]](#)
10. Werfel, F.N.; Floegel-Delor, U.; Rothfeld, R.; Riedel, T.; Goebel, B.; Wippich, D.; Schirrmeister, P. Superconductor Bearings, Flywheels and Transportation. *Supercond. Sci. Technol.* **2012**, *25*, 14007. [\[CrossRef\]](#)
11. Zhao, Y.; Yang, G.; Liu, X.; Shi, Z.; Zhao, L.; Zhang, Z. Dynamics and Thermal Analysis of the Auxiliary Bearing for the Helium Circulator of HTR-PM. *Ann. Nucl. Energy* **2018**, *112*, 474–485. [\[CrossRef\]](#)
12. Keogh, P.S.; Yong, W.Y. Thermal Assessment of Dynamic Rotor/Auxiliary Bearing Contact Events. *J. Tribol.* **2007**, *129*, 143–152. [\[CrossRef\]](#)
13. Salehi, M.; Heshmat, H. On the Dynamic and Thermal Performance of a Zero Clearance Auxiliary Bearing(Zcab) for a Magnetic Bearing System. *Tribol. Trans.* **2000**, *43*, 435–440. [\[CrossRef\]](#)
14. Kong, Y.N.; Xue, Y.J.; Ye, J.; Yu, Y.J. Research on Thermal Analysis of Auxiliary Bearings Based on Thermal Network Method and Finite Element Method. *Mach. Des. Manuf.* **2023**, *10*, 31–35. [\[CrossRef\]](#)
15. Liu, X.N.; Chen, G.H.; Wang, M.Q.; Shi, Z.G.; Mo, N. Analysis for Friction Heating Power of Non-lubricated Auxiliary Bearing of Electromagnetic Bearing during Rotor Drop. *At. Energy Sci. Technol.* **2024**, *58*, 131–140. Available online: <https://yznkxjs.xml-journal.net/article/doi/10.7538/yzk.2024.youxian.0136?viewType=HTML>.
16. Zhao, Y.; Yang, G.; Keogh, P.; Zhao, L. Dynamic Analysis for the Rotor Drop Process and Its Application to a Vertically Levitated Rotor/Active Magnetic Bearing System. *J. Tribol.* **2017**, *139*, 41701. [\[CrossRef\]](#)
17. Zhao, Y.; Liu, X.; Yang, G.; Shi, Z.; Zhao, L. Dynamic Analysis for the Rotor Drop Process and Its Application to a Horizontal Rotor-Active Magnetic Bearing System in Helium Gas. *J. Tribol.* **2018**, *140*, 41704. [\[CrossRef\]](#)
18. Zhao, Y.; Yang, G.; Shi, Z.; Zhao, L. Thermal Analysis and Simulation of Auxiliary Bearings and Its Application in the High Temperature Reactor-10. *J. Tribol.* **2016**, *138*, 11102. [\[CrossRef\]](#)
19. Neisi, N.; Heikkinen, J.E.; Sopanen, J. Influence of Surface Waviness in the Heat Generation and Thermal Expansion of the Touchdown Bearing. *Eur. J. Mech. A Solids* **2019**, *74*, 34–47. [\[CrossRef\]](#)
20. Li, Z.; Yao, J.; Ma, L.; Yang, G.; Lyu, M.; Wang, Y.; Wang, Z. Analysis of the Dry Whirl and Dry Whip Response in a Vertical Active Magnetic Bearing Drop Test Rig. *J. Sound Vib.* **2023**, *549*, 117579. [\[CrossRef\]](#)
21. Li, Z.; Lyu, M.; Yang, G.; Zhao, J.; Wang, Y.; Wang, Z. Dynamic and Thermal Investigations of the Forward Dry-Friction Whirl/Whip of a Vertical Rotor-AMB System during Touchdowns. *Actuators* **2022**, *11*, 291. [\[CrossRef\]](#)
22. Li, Z.; Wang, C.; Wang, Z.; Wang, Y. Dynamics of Vertical Magnetic Suspension Rotor and Touchdown Bearing: Touchdown Mechanism. pp. 1–12. Available online: https://www.magneticbearings.org/app/uploads/publications/ismb17/ismb17_020.pdf (accessed on 2 March 2020).
23. Nie, A.N. Research on Collision Characteristics and Friction Reduction Design of Auxiliary Bearings in an Active Magnetic Bearing System. Master's Thesis, Henan University of Science and Technology, Luoyang, China, 2025.
24. Li, Y.C.; Nie, A.N.; Yang, M.X.; Zhu, D.K.; Qiu, M.; Yang, G.S. Research on Thermal Characteristics of Auxiliary Bearing in AMBs and Friction Reduction Design. *China Mech. Eng.* **2024**, *35*, 646–655. [\[CrossRef\]](#)
25. Ebrahimi, R.; Ghayour, M.; Mohammad Khanlo, H. Chaotic Vibration Analysis of a Coaxial Rotor System in Active Magnetic Bearings and Contact with Auxiliary Bearings. *J. Comput. Nonlinear Dyn.* **2017**, *12*, 31012. [\[CrossRef\]](#)
26. Halminen, O.; Aceituno, J.F.; Escalona, J.L.; Sopanen, J.; Mikkola, A. Models for Dynamic Analysis of Backup Ball Bearings of an AMB-System. *Mech. Syst. Signal Process.* **2017**, *95*, 324–344. [\[CrossRef\]](#)
27. Zhu, D.; Pang, X.; Wang, D.; Qiu, M.; Liu, D.; Dong, Y.; Xu, B. Dynamic Analysis of Vertical Magnetic Suspension System Rotor Dropping to Protective Bearing with Different Structural Parameters. *Int. J. Precis. Eng. Manuf.* **2025**, *26*, 133–155. [\[CrossRef\]](#)
28. Zhu, D.K.; Pang, X.X.; Huang, K.; Qiu, M.; Dong, Y.F. Analysis of Factors Influencing the Impact Resistance of Auxiliary Bearings in a Vertical Magnetic Bearing System Based on Fuzzy Theory. *Bearing* **2024**, *10*, 55–64. [\[CrossRef\]](#)
29. Jarroux, C.; Dufour, R.; Mahfoud, J.; Defoy, B.; Alban, T.; Delgado, A. Touchdown Bearing Models for Rotor-AMB Systems. *J. Sound Vib.* **2019**, *440*, 51–69. [\[CrossRef\]](#)
30. Luo, Y.; Tu, W.; Fan, C.; Zhang, L.; Zhang, Y.; Yu, W. A Study on the Modeling Method of Cage Slip and Its Effects on the Vibration Response of Rolling-Element Bearing. *Energies* **2022**, *15*, 2396. [\[CrossRef\]](#)
31. Li, Z.L.; Shi, Z.G.; Tie, X.Y.; Yang, G.J.; Ren, W.L.; Yao, J.K.; Wang, Y.M.; Wang, Z.X. Failure Mechanism on Touchdown Bearings of Heavy Maglev Rotor Drop. *China Mech. Eng.* **2023**, *34*, 1009–1018. [\[CrossRef\]](#)
32. Zhao, Y.; Zhou, G.; Wang, Q. Discrete Dynamics of Balls in Cageless Ball Bearings. *Symmetry* **2022**, *14*, 2242. [\[CrossRef\]](#)

33. Zhang, J.; Zhang, Y.; Zhou, E.; Wang, Y.; Liu, Z.; Zhao, Y. Research on Multi-Body Dynamic Model of Automatic Discrete System for Protection Bearing without Cage. *Nonlinear Dyn.* **2023**, *111*, 20869–20897. [\[CrossRef\]](#)
34. Jarroux, C.; Mahfoud, J.; Dufour, R.; Legrand, F.; Defoy, B.; Alban, T. Investigations on the Dynamic Behaviour of an On-Board Rotor-AMB System with Touchdown Bearing Contacts: Modelling and Experimentation. *Mech. Syst. Signal Process.* **2021**, *159*, 107787. [\[CrossRef\]](#)
35. Sun, G.; Palazzolo, A.B.; Provenza, A.; Montague, G. Detailed Ball Bearing Model for Magnetic Suspension Auxiliary Service. *J. Sound Vib.* **2004**, *269*, 933–963. [\[CrossRef\]](#)
36. Ma, Z.K.; Zhao, D.X.; Ni, Y.G. Dynamic Response Analysis of the Dropped Rotor–Auxiliary Bearing in a Magnetic Bearing System. *Bearing* **2024**, *3*, 43–51. [\[CrossRef\]](#)
37. Kong, Y.; Xue, Y.; Ye, J.; Yu, Y.; Tie, X.; Yin, Y.; Zheng, H. Analysis on the Impact Factors of Auxiliary Bearing in Vertical Magnetic Bearing. In *Proceedings of the 2019 2nd World Conference on Mechanical Engineering and Intelligent Manufacturing (WCMEIM)*; IEEE: Shanghai, China, 2019; pp. 64–68.
38. Xiao, Z.; Yang, G.J.; Li, Y.; Shi, Z.G.; Yu, S.Y. Load Characteristics Analysis of Auxiliary Bearings for the HTR-10 AMB Helium Circulator. *Nucl. Power Eng.* **2014**, *35*, 82–86.
39. Fonseca, C.A.; Santos, I.F.; Weber, H.I. Influence of Unbalance Levels on Nonlinear Dynamics of a Rotor-Backup Rolling Bearing System. *J. Sound Vib.* **2017**, *394*, 482–496. [\[CrossRef\]](#)
40. Pang, X.; Zhu, D.; Qiu, M.; Wang, D.; Wang, X. Eccentric Rotor Drop Dynamics Study of Vertical Maglev Bearing System. *Lubricants* **2023**, *11*, 246. [\[CrossRef\]](#)
41. Pang, X.; Zhu, D.; Wang, D.; Guan, D.; Liu, D.; Qiu, M.; Dong, Y.; Xu, B. Dynamic Study of Vertical Suspension Rotor Falling on Active Clearance Elimination Protective Bearing. *J. Vib. Eng. Technol.* **2024**, *12*, 773–792. [\[CrossRef\]](#)
42. Zhao, Y.; Yang, G.; Liu, X.; Shi, Z.; Zhao, L. Research on Dynamics and Experiments about Auxiliary Bearings for the Helium Circulator of the 10MW High Temperature Gas-Cooled Reactor. *Ann. Nucl. Energy* **2016**, *95*, 176–187. [\[CrossRef\]](#)
43. Sun, G. Rotor Drop and Following Thermal Growth Simulations Using Detailed Auxiliary Bearing and Damper Models. *J. Sound Vib.* **2006**, *289*, 334–359. [\[CrossRef\]](#)
44. Lai, H.; Pengyang, J.; Zixi, W.; Yuming, W. Study on Type of Magnetic Suspension Rotor Groove and Wear of Drop Touchdown Bearing. *Eng. Appl. Comput. Fluid Mech.* **2024**, *18*, 2422065. [\[CrossRef\]](#)
45. Lee, J.G.; Palazzolo, A. Catcher Bearing Life Prediction Using a Rainflow Counting Approach. *J. Tribol.* **2012**, *134*, 31101. [\[CrossRef\]](#)
46. Jia, P.Y.; Cui, B.L.; Li, A.Q.; Hu, L.; Wang, Z.X. Drop Test of a Horizontal Magnetically Suspended Rotor and Failure Analysis of Auxiliary Bearings. *Bearing* **2025**, *8*, 102–111. [\[CrossRef\]](#)
47. Zhang, K.; Xu, Y. Preliminary Study on Recontrol Evaluation of a Magnetic Bearing Rotor Falling on Touch-down Bearings. *Energies* **2024**, *17*, 3457. [\[CrossRef\]](#)
48. Du, Y.; Zhang, G.; Hua, W. Review on Research and Development of Magnetic Bearings. *Energies* **2025**, *18*, 3222. [\[CrossRef\]](#)
49. Palmgren, A. *Ball and Roller Bearing Engineering*, 3rd ed.; SKF Industries Inc.: Philadelphia, PA, USA, 1959.
50. Zhao, Y.; Yang, G.; Shi, Z.; Zhao, L. Thermal Growth Simulations of the Rotor and Auxiliary Bearing Contact Events. In *Proceedings of the Volume 1: Plant Operations, Maintenance, Engineering, Modifications, Life Cycle and Balance of Plant; Nuclear Fuel and Materials; Plant Systems, Structures and Components; Codes, Standards, Licensing and Regulatory Issues*; American Society of Mechanical Engineers: Prague, Czech Republic, 2014; p. V001T03A001.
51. Yang, G.; Sun, Z.; Liu, X.; Shi, Z. Research on Auxiliary Bearing Structure with Buffer Shim Based on LS-DYNA for Helium Circulator of HTR-10. In *Proceedings of the Volume 2: Plant Systems, Structures, Components, and Materials; Risk Assessments and Management*; American Society of Mechanical Engineers: London, UK, 2018; p. V002T03A013.

Disclaimer/Publisher’s Note: The statements, opinions and data contained in all publications are solely those of the individual author(s) and contributor(s) and not of MDPI and/or the editor(s). MDPI and/or the editor(s) disclaim responsibility for any injury to people or property resulting from any ideas, methods, instructions or products referred to in the content.