Model Analysis of Axial PM Bearingless Flywheel Machine

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Abstract

The special rotor structure of the axial PM bearingless flywheel machine (APM-BFM) makes its critical speed more easily affected at high speed, but there are few studies on its rotor dynamics, especially under load conditions. In the paper, the model analysis study of the APM-BFM is carried out. First, the mechanism model of APM-BFM is used to explain its structure and its system is analyzed in conjunction with the experimental prototype; The rotor centroid deviates from the center of rotation due to rotor eccentricity, which is an important reason for the serious complex vibration; and then, the rotor model principle is explained from the mechanism model. To be more specific, the finite element method and the analytical technique are used to compare and check the accuracy of the finite element method model analysis; after that, the rotor model is simulated and studied. Finally, the influence of bearing stiffness, gyro effect and flywheel load on critical speed is analyzed, which provides theoretical basis for high-speed and stable motor control.

Index Terms

Permanent magnet bearingless, critical speed, model, finite element

I. INTRODUCTION

Flywheel Energy Storage (FES) is a mechanical energy storage technology whose advantages include high power and energy density, fast response speed and long life. FES has received a lot of attention because of its unique advantages in the fields of power grid frequency regulation, new energy grid connection, rail transportation, uninterruptible power supply, aerospace satellites, and so on [1]. Due to the high-speed operation of the flywheel, mechanical bearings will produce a lot of heat, and the cost of the cooling equipment to ensure steady operation is high. The magnetic bearing (MB) allows the shaft to rotate without touching anything, which reduces heat and saves cost. As a result, combining a MB with a switched reluctance machine (SRM) to create a bearingless switched reluctance machine (BSRM) with compact structure, high critical speed, and minimal friction is a compelling option. This can be incorporated into Flywheel Energy Storage Systems (FESSs) to form a bearingless flywheel machine (BFM), which can improve system integration, critical speed, and reliability, and be known as the ideal choice of flywheel energy storage motor [2].

An axial PM bearingless flywheel machine (APM-BFM) separates the stator and rotor cores into two phases (phase A and B) along the axial direction. The PM is attached between the two-phase stator cores, giving an axial path for the PM flux [3], allowing for four-degree-of-freedom suspension through bias magnetic flux generated by the PM. APM-BFM merges two sets of windings into one motor, providing for four degrees of freedom suspension and integrated charging or discharging operation, which greatly improves system integration and reduces system power consumption. In recent years, it has received considerable attention. Reference [4] proposed that the use of a magnetic guide sleeve improves the magnetic permeability and reduce the amount of rare earth permanent magnet materials consumed, and the design of pole shoe teeth effectively improves the output torque and suspension force. To construct an APM-BFM nonlinear dynamic equivalent magnetic network model, reference [5] divided the stator-rotor air gap magnetic circuit into multiple regions and introduced local saturation coefficients to consider the local magnetic saturation of the rotor yoke, which improved the model accuracy. The APM-BFM
equivalent thermal network model was developed in Reference [6] to solve the difficulties of high thermal load and small heat dissipation area of the centralized winding, and the validity of model was confirmed using magneto-thermal coupling simulation. There has been a lot of studies on the mechanism model, electromagnetic model, and magneto-thermal coupling model of APM-BFM, but there is little study on the rotor motion model features under high-speed operation.

High-speed and ultra-high-speed flywheel rotor stability problems are particularly prevalent in FESSs, which makes the model characteristics of rotor motion become a hot topic, and it is also the key to exploit the high-speed superiority of APM-BFM. Dynamic analysis of the APM-BFM model is needed to prevent the resonance-induced instability problem due to the influence of system bearing stiffness, gyroscopic effect, rotor structure, and other variables that cause the APM-BFM speed to be closer to the critical speed. When evaluating complex rotor systems, dynamic analysis methods such as finite element analysis (FEA) and transfer matrix method (TMM) are used [7]. When studying complicated rotor systems, the matrix order in TMM does not grow with the number of system degrees of freedom, which makes the computation fast and compact, but prone to numerical instability. FEA is characterised by discretizing the continuous object, dividing it into finite units, interpolating each unit, and establishing the interpolation function. FEA is more complicated than TMM, but it has better computation accuracy and can effectively avoid numerical instability, which makes it frequently utilized and has become a generally acknowledged and successful algorithm.

Based on the FEA of electromagnetic and structural coupling, the finite element model of SRM rotor dynamics was established to simulate the dynamic characteristics of rotor eccentricity [8]-[9]. FEA was used to analyze the rotor mode and inherent frequency of high-speed PM motor, and the critical speed of motor in the free state. Bearing position, and actual operating condition were obtained, and compared with the experimental method to verify the effectiveness of the model method [10]-[11]. FEA was used to calculate the inherent frequency of high-speed motor, and the influence of support stiffness and rotor gyroscopic effect on the inherent frequency of the system were obtained [12]-[13].

APM-BFM is a new structure of magnetically levitated motor, lots of previous papers on the mechanism model and magneto-thermal coupling of APM-BFM have ignored the operating mode characteristics of APM-BFM under high-speed operation, and its outer rotor structure will have obvious gyroscopic effects under high-speed drive. However, little attention has been paid to the performance of the APM-BFM system in actual operation. This paper proposed the influencing rule of bearing stiffness, gyroscopic effect and flywheel load on APM-BFM from the perspective of model dynamics. Based on the fundamental structure and the experimental prototype of APM-BFM system, the FEA method is introduced into APM-BFM for model analysis and simulation, so as to lay the theoretical basis for the high speed and high stability operation of APM-BFM in the flywheel field.

II. STRUCTURE OF APM-BFM SYSTEM

A. STRUCTURE OF APM-BFM

The structure of APM-BFM is shown in Fig.1, including rotor, torque winding, suspension winding, isolated magnetic ring, torque pole, suspension pole, shaft, permanent magnets, phase A and phase B. The outer rotor of APM-BFM is installed inside the flywheel and integrated with the flywheel. There are 12 rotor poles of equal width evenly distributed on the inner side of phase A and phase B rotors respectively, and the axes of the two-phase rotors differ by 15°. An axially permanent magnet is installed in the middle of the two-phase stators. Each phase adopts inner stator and outer rotor, which are made up of a 12/12 pole arrangement. Four suspension poles (wide tooth) and eight torque poles (narrow tooth) constitute the 12 poles of the inner stator. The APM-BFM achieves decoupling of the suspension and torque by using a magnetic isolation ring between the torque pole and the suspension pole. It utilizes torque windings and suspension windings to realize rotor rotation and radial four-degree freedom suspension, and then the flywheel operates to realize the conversion between mechanical energy and electrical energy.

![FIGURE 1. The structure of APM-BFM (a) Front view. (b) Side view.](image-url)
B. FLYWHEEL ROTOR SYSTEM

Fig. 2 shows the flywheel rotor system of APM-BFM, where the three-dimensional assembly structure section and the experimental prototype are shown in Fig. 2(a) and (b). APM-BFM system structure mainly includes induction ring for sensors, motor shell, stator shaft, eddy-current sensors and winding outlet. Some structural materials are listed in Table II. The flywheel system’s essential component is the APM-BFM with rated power of 3 KW and steady speed of 14000r/min.

III. ROTOR MODE ANALYSIS IN PRINCIPLE

In the high-speed functioning of the APM-BFM, rotor eccentricity is a natural occurrence. The eccentricity of the rotor causes the centre of mass of the rotor to wander from the rotation centre, resulting in noticeable and complicated vibration phenomena. This is a significant factor impacting high-speed operation stability of the APM-BFM, and the rotor model principle must be investigated.

According to the finite element theory of elasticity, the kinematic equations of the rotor system can be expressed as:

\[ M \ddot{x} + C \dot{x} + K x = F(t) \]  

(1)

Here: \( M \) represents the mass matrix, \( C \) is the damping matrix, \( K \) is the stiffness matrix, \( \dot{x} \) is the displacement vector, \( \dot{x} \) is the velocity vector, \( \ddot{x} \) is the acceleration vector, and \( F(t) \) is the system load.

As long as there is no deflection of the rotor rotation, the differential equation of rotor dynamics can be stated as:

\[
\begin{aligned}
& m \frac{d^2 x(t)}{dt^2} + c_n \frac{dx(t)}{dt} + c_r \left[ \frac{dx(t)}{dt} + \Omega y(t) \right] + kx(t) = F_x \\
& m \frac{d^2 y(t)}{dt^2} + c_n \frac{dy(t)}{dt} + c_r \left[ \frac{dy(t)}{dt} + \Omega x(t) \right] + ky(t) = F_y
\end{aligned}
\]  

(2)

Here: \( m \) stands for the rotor mass, \( c_n \) is the rotor external damping matrix, \( c_r \) is the rotor system internal damping, \( \Omega \) is the rotor mechanical angular velocity, \( k \) is the rotor bearing stiffness, \( x(t) \) and \( y(t) \) are the displacements in the \( x \) and \( y \) directions, \( F_x \) and \( F_y \) are the excitation forces in the \( x \) and \( y \) directions. When the rotor is eccentric, the rotor differential equation may be stated as:
\[ \begin{bmatrix} F_x \\ F_y \\ F_z \end{bmatrix} = \begin{bmatrix} c_{xx} & c_{xy} & c_{xz} \\ c_{yx} & c_{yy} & c_{yz} \\ c_{zx} & c_{zy} & c_{zz} \end{bmatrix} \begin{bmatrix} x' \\ y' \\ z' \end{bmatrix} + \begin{bmatrix} k_{xx} & k_{xy} & k_{xz} \\ k_{yx} & k_{yy} & k_{yz} \\ k_{zx} & k_{zy} & k_{zz} \end{bmatrix} \begin{bmatrix} x \\ y \\ z \end{bmatrix} \] (3)

Here: \( c_{xx}, c_{xy}, c_{xz}, c_{yx}, c_{yy}, c_{yz}, c_{zx}, c_{zy}, c_{zz} \) are the rotor’s damping coefficients in x, y, and z directions respectively; \( k_{xx}, k_{xy}, k_{xz}, k_{yx}, k_{yy}, k_{yz}, k_{zx}, k_{zy}, k_{zz} \) are the rotor’s bearing stiffness coefficients in x, y, and z directions, respectively. The displacement of the flywheel rotor system may be stated as:

\[ r(t) = x(t) + jy(t) = r_0 e^{\lambda t} \] (4)

Here: \( r_0 \) is a constant.

When no external force is applied, both \( F_x \) and \( F_y \) in formula (2) are zero. Substituting formula (4) into formula (2), formula (2) can be expressed as:

\[ mS^2 + (c_r + c_n)S + k - j\Omega c_r = 0 \] (5)

By solving formula (5), it has:

\[ S = \sigma + j\omega = \frac{-c_r + c_n}{2m} \pm \sqrt{\frac{(c_r + c_n)^2 - 4m(k - j\Omega c_r)}{4m^2}} \] (6)

The imaginary component \( \omega \) of formula (6) represents the intrinsic frequency of the rotor in free rotation, and \( S \) has two roots, positive and negative. Idealizing the rotor’s surroundings with ignoring external damping, the rotor’s undamped modal frequency in free rotation under shaft-supported circumstances is:

\[ \omega_{n,2} = \sqrt{\frac{k}{m}} \] (7)

Here: \( \omega_{n,2} \) is the motor rotor’s inherent frequency in forward and against precession, respectively.

In the absence of a rotor, the motor’s uniform mass rotor is removed and regarded as a free rotor, at which point the rotor’s intrinsic frequency can be expressed as:

\[ \omega_n = a_n \sqrt{\frac{EJ}{\rho D^4}} \] (8)

Here: \( E \) is the outer rotor material’s modulus of elasticity; the outer rotor material is silicon steel sheet 50W470. \( J \) is the rotational inertia of outer rotor, \( \rho \) is the density, \( D \) is the rotor diameter. The values of constant are listed as follows: \( E = 4.2 \times 10^8 \text{ Pa} \), \( J = 0.238 \text{kg} \cdot \text{m}^2 \), \( \rho = 7.65 \text{g} / \text{cm}^3 \), \( D = 130 \text{mm} \) and \( a_n \) is the model coefficient that can be expressed as:

\[ a_n = S_n \cdot Y_n \] (9)

Here: \( S_n \) is the value of frequency response, \( Y_n \) is vibration participation factor. In bending mode, \( a_1 = 0.16 \) in 1st order and \( a_2 = 0.4 \) in 2nd order.

Analytical method and FEA were used to analyze the frequencies of the APM-BFM rotor in free state, and the results of the comparative analysis are shown in Table III.

| Bending Modes | Rotor frequency \( \omega_n / \text{Hz} \) | Error |
|---------------|--------------------------------------|-------|
| 1st order mode | 1082.44 | 950 | 0.139 |
| 2nd order mode | 2691.04 | 2505 | 0.069 |

IV. ANALYSIS OF ROTOR MODE SIMULATION

The previous sections analyzed the APM-BFM system structure, whose special structure makes the process of FEA difficult, and therefore rare literature studies critical speed of APM-BFM, analyzing the influence of bearingless stiffness, gyroscopic effect, flywheel load on critical speed which are beneficial to study the modal characteristics of APM-BFM under high speed operation.

A. CRITICAL SPEED ANALYSIS OF APM-BFM

1) PRE-PROCESSING OF FEA ANALYSIS

3D model is imported into the workbench and simplified, and then the bearing is converted to a spring unit with stiffness based on the APM-BFM structural characteristics, so that the stiffness of the spring unit can be changed to see how the bearing stiffness affects the critical speed. A simple finite element module is shown in Fig.3.

**FIGURE 3. The system simplified model of APM-BFM**

The mesh division level is set to 2 and free mesh division is utilized. The geometry of the APM-BFM is discretized automatically into basic building blocks, that is tetrahedra in 3D. The assembly of all tetrahedra is referred to as the finite element mesh of the model or simply the mesh, and the mesh is automatically refined to achieve the required level of accuracy in field computation. Fig.4(a) illustrates the mesh section. Fig.4(b) demonstrates how Workbench is used to build up the pre-stress model analysis and how the computation is done by inputting the boundary pressure values.
FIGURE 4. Preprocessing diagram with Workbench (a) Section of the system's finite element mesh, (b) Module for Pre-stress Model Analysis.

2) CALCULATION OF CRITICAL ROTOR SPEED

The model of the APM-BFM system is analyzed using FEA, and the model analysis is utilized to understand the rotor's vibration characteristics. The vibration pattern of the system from the 1st order to the 6th order is shown in Fig.5.

FIGURE 5. The diagram of vibration for APM-BFM (a) 1st order. (b) 2nd order. (c) 3rd order (d) 4th order (e) 5th order (f) 6th order

Fig.5 shows the energy transfer trend of APM-BFM. The 1st and 2nd order model deformation locations are mainly located at the motor shell and the connection between the motor shell and the induction ring for sensors, which indicates that the force here is greater. Under the 3rd, 4th, and
5th order model conditions, the stress deformation near the connection between the motor shell and the induction ring for sensors becomes increasingly apparent as the rotating speed increases. This means that the higher the APM-BFM speed, the stronger the motor shell material should be, and the stronger the material at the motor shell-to-sensor inductor ring connection should be.

The intrinsic frequencies associated with each rank are listed in Table IV. It shows that the 1st order modal frequency corresponding to the stiffness of 10^12 N/M is 49.421 Hz, and the corresponding speed is 2965.3 r/min. At this time, the APM-BFM system vibrates axially. The 2nd and 3rd order inherent frequencies are similar, representing the 1st critical speed of bending. The 4th and 5th order inherent frequencies are similar, representing the 2nd critical speed of bending. The critical speed value is high, and when the rated speed is close to the critical speed, the motor will vibrate violently. Therefore, the safe working range of the 1st order bending mode and the 2nd order bending mode is between the two critical speeds.

**TABLE IV**

| Mode   | Natural frequency/Hz | Critical speed/rpm |
|--------|----------------------|-------------------|
| 1st order | 49.421               | 2965.3            |
| 2nd order | 950.87               | 63241.2           |
| 3rd order | 950.89               | 57208.9           |
| 4th order | 2505.4               | 68407.3           |
| 5th order | 2505.5               | NONE              |
| 6th order | 3542.5               | NONE              |

**B. THE INFLUENCE OF BEARINGLESS STIFFNESS ON THE CRITICAL SPEED**

The influence of bearing stiffness on APM-BFM critical speed is further investigated. Fig.6 depicts the relationship between the bearing stiffness and the 1st order to 4th order critical speed of the motor.

**FIGURE 6. The influence of bearingless stiffness on the critical speed**

The bearing stiffness has different impacts on the critical speed in different modes, as shown in Fig.6. In the 1st order mode, bearing stiffness has little effect on the rotor critical speed, however in the 2nd, 3rd, and 4th order modes, bearing stiffness has a greater impact on the rotor critical speed. Furthermore, as shown in Fig.6, the critical rotor speed changes more slowly in the 2nd, 3rd and 4th order modes when the bearing stiffness is less than 10^8 N/M; in the 2nd order mode, the critical rotor speed remains relatively stable when the bearing stiffness is between 10^9 N/M and 10^10 N/M, whereas the critical rotor speed increases faster in the 3rd and 4th order modes when the bearing stiffness is greater than 10^9 N/M. When the rotor 2nd order model bearing stiffness is more than 10^10 N/M, this phenomenon happens. It can be seen that when the bearing stiffness is between 10^7 N/M and 10^8 N/M, the influence on the critical speed is minimal; however, when the bearing stiffness exceeds 10^9 N/M, the effect on the critical speed is significant, and as the stiffness rises, the critical speed increases more rapidly.

**C. THE GYROSCOPIC EFFECT ON CRITICAL SPEED**

The rotational inertia of the APM-BFM flywheel rotor will not change drastically due to the addition of any form of load, but the gyroscopic effect will have an influence on the critical speed. The gyroscopic effect of the rotor should be addressed in the dynamics simulation of APM-BFM, the impact of gyroscopic effect on the motor should be verified using FEA, and the Campbell diagram is created as shown in Fig.7.

**FIGURE 7. Campbell graft**

The red line segment with Ratio=1 in Fig.7 denotes the line segment where the flywheel speed and critical speed are equal, from which the system's critical speed value may be calculated, as indicated in the red square. Under the impact of the gyroscopic effect, the APM-BFM rotor mode may be divided into forward and reverse precession as the speed rises, and the forward precession frequency increases as the speed rises, while the reverse precession frequency decreases. However, as shown in the Fig.7, the structural design of this study reduces the influence of the gyroscopic effect on the critical speed by making the change of both forward and reverse precession frequencies relatively modest.

**D. EFFECT OF FLYWHEEL LOAD ON CRITICAL SPEED**

The rotor structure will alter in real-world operation as a result of varied flywheel loads, and the impact of the flywheel structure on the critical speed cannot be overlooked. Fig.8 depicts the APM-BFM model with the flywheel module. The green and blue sections of the illustration depict the six-flywheel load rings installed on the housing, each flywheel load ring weights 1 kg, a total of 6 kg. The effect of
carrying the flywheel load ring on the critical speed is studied using FEA, and Table IV shows the inherent frequency of the APM-BFM system carrying the flywheel load ring.

FIGURE 8. APM-BFM model with flywheel load ring

TABLE V

| Mode   | Natural frequency/Hz | Critical speed/rpm |
|--------|-----------------------|---------------------|
| 1<sup>st</sup> order | 108.24               | 6494.2              |
| 2<sup>nd</sup> order | 662.39               | NONE                |
| 3<sup>rd</sup> order | 663.96               | NONE                |
| 4<sup>th</sup> order | 1709                 | NONE                |
| 5<sup>th</sup> order | 1739.7               | NONE                |
| 6<sup>th</sup> order | 2749.6               | NONE                |

VI. APM-BFM STATIC OPERATION ANALYSIS

The vibration stabilization is achieved using conventional PID with P=2, I=60, D=0.006. The blue trajectory curve represents the x-direction amplitude curve of APM-BFM static levitation, whereas the red trajectory curve represents the y-direction amplitude curve, as shown in Fig.9. At first APM-BFM achieves 0.15mm where is the center of mass displacement in the x-direction, then achieves stable levitation in the x-direction within a certain range after 50ms of vibration, and the amplitude change in the y-direction occurs at 0.65s due to the small value given by the limiting link, after which APM-BFM achieves stable levitation in both the x and y directions simultaneously.

FIGURE 9. Static amplitude curve of APM-BFM

VI. CONCLUSIONS

This paper focuses on the operation stability of the APM-BFM, emphasizing its high-speed superiority, develops a simulation model, and uses FEA to undertake a joint simulation analysis of the APM-BFM system model. The following are the primary conclusions:

1. By comparing and evaluating the FEA and the analytical method, the correctness of the FEA was validated, and its error range was estimated.
2. If the APM-BFM speed is raised, the material strength of motor shell and the connection between the motor shell and the induction ring of sensors must be improved.
3. Both the 1<sup>st</sup> and 2<sup>nd</sup> critical speeds of bending are high, and the APM-BFM can run in a safe range without resonance until it achieves a steady speed of 14000r/min.
4. When the bearing design stiffness is between 10<sup>7</sup>N/M and 10<sup>8</sup>N/M, the critical speed effect is essentially nonexistent, and when the bearing stiffness surpasses 10<sup>9</sup>N/M, the critical speed change accelerates.
5. The critical rotor speed is effectively raised by adding a flywheel load ring, demonstrating that the APM-BFM can ensure that the rated speed is far from the critical speed under load to avoid resonance affecting operation.

In general, improving the edge strength of the motor can reduce the deformation, the rotor structure is designed as a rigid rotor that can effectively avoid the resonance phenomenon, operation range of the APM-BFM increases with the increase of bearing stiffness and flywheel load, and using the traditional PID method, the stable suspension of APM-BFM can be realized in a short time. The simulation and analytical results presented in this study give the theoretical foundation for optimum design and high-speed stable control of APM-BFM.

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