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Analysis of Multi-physics Field Coupling Characteristics of Liquid-Immersed Shielded Motors

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ABSTRACT

This paper focuses on the special operating conditions of shielded motors in liquid environments, using a 7kW aviation pump shielded motor as a case study. A systematic analysis of its multi-physics coupling characteristics is conducted to reveal the effects of the interaction between different physical fields on the motor. First, by combining practical operating conditions, the inherent frequencies and vibration modes of the rotor system of the immersion liquid shielded motor are compared under different conditions. Then, based on the fluid field, a fluid-structure coupling analysis is performed to solve for the prestressed modal frequencies. The results show that multi-physics coupling analysis of the rotor system provides a more accurate representation of the motor's operational characteristics compared to conventional analyses and should not be overlooked. This provides a reference for the design and optimization of such motors.

KEYWORDS

Shielded Motor; Modal Analysis; Rotor Dynamics; Multi-physics Coupling.

1. INTRODUCTION:

As a type of special-purpose motor, the shielded motor has the characteristics of leak-proof, safe, and stable operation in unique working environments, which has led to its widespread application in industries such as aerospace and aviation [1-3]. A shielding sleeve is placed between the stator and rotor to prevent leakage, while the internal circulating medium can cool the motor. However, due to the interaction between the rotating fluid field inside the shielded motor and the rotor structure, the rotor experiences pressure pulsations exerted by the fluid, which can easily induce motor vibrations and affect its safe and stable operation [4-6]. Therefore, to improve the operational stability of shielded motors used in aviation pumps, it is essential to conduct in-depth research into their stress, dynamics, and other characteristics.

Since the 1950s, with the development of aerospace, industry, and other fields, many scholars both domestically and internationally have studied the characteristics of rotor systems [7-9]. However, there has been limited research on the effects of internal fluid flow on the rotor of shielded motors, as well as the consideration of multi-physics characteristics of shielded motors.

In the area of fluid-structure interaction (FSI), Yang Yongfei [10] conducted a modal analysis of a multistage centrifugal pump using ANSYS, and the study revealed that under wet modal conditions, the rotor system's natural frequency decreases due to the additional mass of the fluid. Wu Jie [11] performed a unidirectional fluid-structure coupling study of multiphase flow in a multistage centrifugal pump, with results indicating that the flow inside the impeller is more stable than the flow inside the guide vanes. M. Chouksey et al. from India [12] investigated the rotor-shaft system modes under the combined effects of rotor shaft material damping and fluid. Reference [13] focused on a radial flow pump, conducting an FSI dynamic study on the deformation and stress distribution of impeller blades, concluding that the impeller stress fluctuates with changes in flow rate, and operation under low flow conditions should be avoided. Reference [14] analyzed the effects of motor speed, temperature rise, and sleeve parameters on the structural strength of the motor rotor, comparing the stress of the motor rotor and sleeve at different operating temperatures. Reference [15] took a nuclear primary pump shielded motor as an example, using fluid mechanics principles to establish a coupled physical model of the three-dimensional fluid field and temperature field, and conducted finite element calculations to determine the distribution pattern of internal temperature rise, providing a reference for the structural design of such motors.

In summary, the structural characteristics of rotor systems have long been a focus of both domestic and international scholars. Current research primarily concentrates on areas such as impellers and pump turbines, while studies on the rotor system of shielded motors under special operating conditions are relatively scarce. To address this gap, this paper takes the example of a shielded motor used in an aerospace pump. First, a comparative analysis of the modal characteristics of the motor rotor system under different conditions is conducted. In consideration of the unique operating conditions of the shielded motor in a liquid environment, a simplified fluid field model for the rotor components is established. Simulations of the full flow field pressure and convergence, influenced by external excitations, are performed. The results of the fluid field calculations are coupled with the stress field and modal characteristics, leading to the fluid-structure interaction (FSI) modal that accounts for prestress. This study contributes valuable insights to the development of multi-physics coupling analysis methods in the field of electric motors.

2. LIQUID-SHIELDED MOTOR STRUCTURE

Taking a 7kW shielded motor used in an aerospace pump as an example. During operation, the motor is designed with a cooling structure that allows external fluid to enter the motor through the rear end cover. It then flows through the bearings and shielded sleeve cavity before exiting from the front end cover, creating a cooling system based on an internal and external fluid circulation. This internal-external circulation design not only significantly simplifies the motor's cooling system structure but also ensures that the shielded cavity remains filled with fluid during operation. This design further enhances both the cooling and heat dissipation effects while simultaneously lubricating the bearings, ensuring the safe and stable operation of the motor under prolonged and high-load conditions. The main parameters of the shielded motor analyzed in this paper are presented in Table 1.

Table 1. Main Parameters of the motor

Parameters	Numerical value	Parameters	Numerical value
Polar logarithm	4	Parallel branch circuits	1
Stator outer diameter (mm)	112	Rotor outer diameter (mm)	76.4
Stator inner diameter (mm)	77.2	Effective air gap length (mm)	0.2
Axial length (mm)	160	Shielding sleeve thickness (mm)	0.2
Stator slots	48	Shielding sleeve material	PEEK
Rotor slots	44	Rated speed (rpm)	5700

3. ROTOR STRUCTURE MODAL ANALYSIS

Modal analysis refers to the inherent vibration characteristics of an object's structure, generally classified into dry modes and wet modes. Dry modes refer to the inherent modal characteristics of the rotor structure in air, while wet modes refer to the inherent modal characteristics of the rotor structure in water. The key difference lies in the fact that wet modes account for the influence of surrounding fluid on the structural modes. During analysis, a fluid-structure coupling interface must be established, and the added mass effect of the liquid in the fluid field must also be considered [16]. In practical scenarios, when the shielded motor rotor structure is fully submerged in water, its inherent frequencies will change due to environmental effects, making the rotor structure more prone to vibration. Therefore, this section primarily focuses on a comparative analysis of the dry modes of the shielded motor used in aerospace pumps, extracting rotor mode shapes and inherent frequencies, and analyzing the deformation of various modes under different conditions.

3.1. Principle of Modal Analysis

Modal analysis is essentially the process of solving the undamped vibration equation of the rotor structure, i.e., the characteristic equation. Assuming the rotor structure operates in a vacuum, with no consideration of the fluid environment or external excitations, the structural dynamics equation is given by Equation 1.

$$[M]\{a\}+[C]\{v\}+[K]\{x\}=\{F(t)\}$$

In the equation, $[M]$, $[C]$, $[K]$ represent the mass matrix, damping matrix, and stiffness matrix, respectively [17-18]. $\{a\}$, $\{v\}$, $\{x\}$are the acceleration vector, velocity vector, and displacement vector, respectively. $\{F(t)\}$ represents the external excitation applied to the system. In dry modal analysis, the influence of external loads and structural damping on the system's modal characteristics is ignored, simplifying the equation as follows:

$$[M]\{a\}+[K]\{x\}=0$$

To solve this characteristic equation, let its solution be:

$$\{X(t)\}=\{z\}\sin(\omega\bullet t+\varphi)$$

By combining the above equation, the structural vibration equation can be obtained as:

$$([K]-w^2[M])\{z\}=0$$

From Equation 4, the structural vibration equation indicates that the dry modal natural frequency and mode shape of the rotor are primarily related to the system's mass matrix and stiffness matrix. In dry modal analysis, the mass matrix is determined by the material properties of the structure itself, while the stiffness matrix is derived by simplifying the bearings as spring supports that consider stiffness.

Unlike dry modal analysis, the calculation of wet modes requires considering the effects of fluid-structure interaction, which involves the coupling of the structural equation with the fluid motion equation. When the shielded motor rotor is immersed in fluid, the inherent frequency of the rotor structure will change, requiring consideration of the fluid's influence on the structure. Since the shielded pump, under normal operating conditions, does not exhibit heat transfer even when internal fluid flow becomes unstable, and the fluid medium is water (which can be treated as an incompressible fluid), the coupled fluid-structure vibration equation is:

$$[M]\{a\}+[C]\{v\}+[K]\{x\}=[B]^T\{p\}-\{f_0\}$$

$$[E]\{p^{''}\}+[A]\{p^{'}\}+[H]\{p\}=-\rho[B]\{a\}-\{q_0\}$$

In the equation, [B] is the coefficient matrix, $\{p\}$is the pressure vector coupling the fluid field and the structural field, $\{f_0\}$represents other external excitations, and $\{q_0\}$is the input excitation.

Since the medium is an incompressible fluid, simplifying the above equation gives:

$$[M]\{a\}+[K]\{x\}=[B]^T\{p\}$$

$$[H]\{p\}=-\rho[B]\{a\}$$

By combining, we obtain:

$$([M]+[M_a])\{a\}+[K]\{x\}=0$$

In the equation, $[M_a]$ represents the added mass matrix, and it is given by $[M_a]=\rho[B]^T[H]^{-1}[B]$

From the equation, it can be seen that the key difference in considering fluid-structure interaction compared to dry modal analysis lies in the added mass matrix. The influence of the fluid on the structural field is also reflected through the added mass.

3.2. Dry Mode Setting

Considering the effects of the mass and stiffness of each part of the rotor system, the rotor model of the shielded motor is appropriately simplified. Figure 2 shows the dry modal model of the shielded motor rotor system.

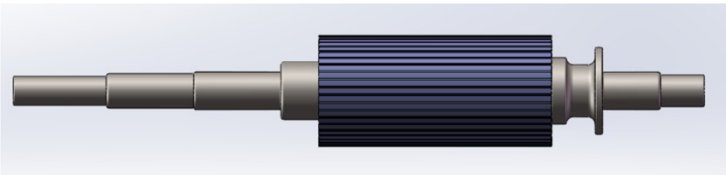
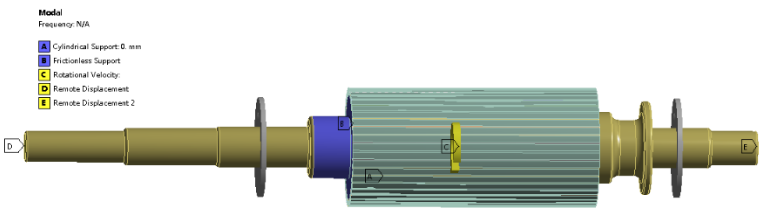
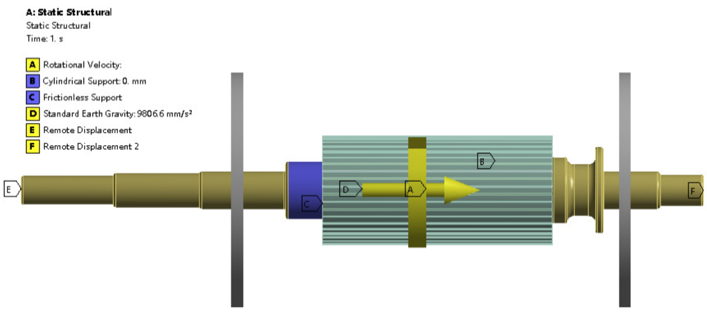


Figure 1. Dry mode model of the rotor system

During the dry modal analysis, the rotor system is subjected to corresponding constraints and divided into two operating conditions for comparative analysis: the no-prestress condition and the condition with applied gravitational and centrifugal forces. The applied loads and constraints for the different conditions are shown in Figure 1.



a) Non-prestressed working condition



b) Gravity and centrifugal force conditions

Figure 2. Applied Loads and Constraints

As shown in Figure 2, compared to the no-prestress condition, in the second operating condition, the gravity and centrifugal forces on the rotor system are primarily implemented by setting the rotational speed and gravitational acceleration. In addition to the bearing constraints, other parts of the rotor system are subjected to remote displacement constraints to ensure that the boundary conditions align with the actual operating conditions of the rotor system.

3.3. Modal Comparison of the Rotor System under Different Working Conditions

The first four modes are selected for comparison, and the modal frequencies of the rotor system under different operating conditions are shown in Table 2.

Table 2. Modal Frequencies of the rotor system under different working conditions

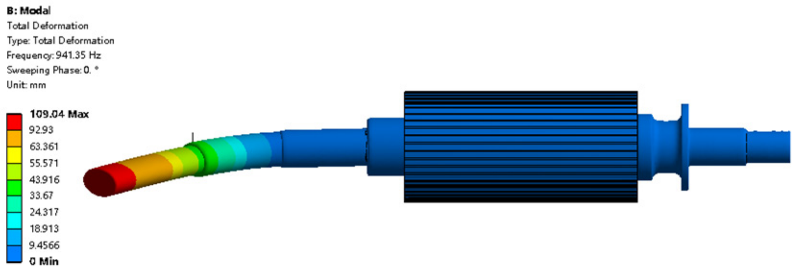
Working condition	First stage /Hz	Second stage /Hz	Third stage /Hz	Fourth stage /Hz
No prestress	1062	1067.7	3967.6	3976.5
Consider gravity and centrifugal force	941.35	943.03	3780.7	3788.7

As shown in Table 2, with the increase in mode order, the natural modal frequencies also increase. Among them, the modal frequencies considering gravity and centrifugal force are lower compared to the no-prestress condition, and the reduction trend gradually diminishes. This indicates that gravity and centrifugal forces affect the natural frequencies of the rotor system. In the simulation process, the rotor system was appropriately simplified, so the actual rotor modal frequencies will be slightly lower than the simulation results. Additionally, in the first four modes, there is a phenomenon where the modal frequencies of adjacent orders are close. Taking the no-prestress condition as an example, the first mode is 1062 Hz and the second mode is 1067.7 Hz, while the third mode is 3967.6 Hz and the fourth mode is 3976.5 Hz. The main reason for the close frequencies is that, on the one hand, the periodic structure symmetry of the rotor system leads to similar frequencies, and on the other hand, the rotor system exhibits both forward and backward whirl, with backward whirl modes appearing in the first four modes. These will be analyzed in detail in the subsequent sections of this paper.

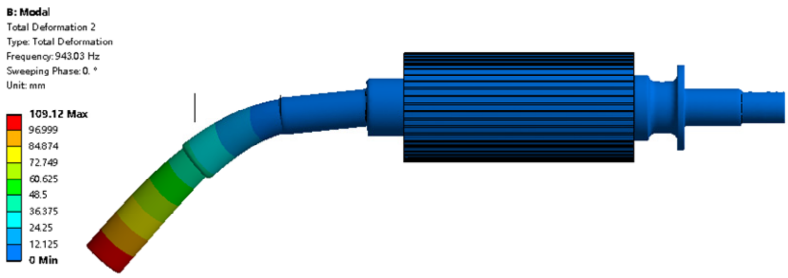
Figure 3 shows the modal deformation shapes of the rotor system under the condition with applied gravity and centrifugal forces. The deformations in the figure are not the actual deformations of the rotor system; the results obtained from the simulation primarily reflect the deformation trends of the rotor system, and the specific values are not of practical reference. Since the modal deformation shapes of the rotor system under the no-prestress condition are almost identical to those shown here, further explanation is omitted. From the figure, it can be seen that the first mode shows bending vibration in the Y direction, with the bending occurring near the bearing at the rotor's shaft end, and the maximum displacement is the largest among the four modes. The second mode exhibits torsional vibration in the X and Y directions, with the bending occurring near the bearing at the shaft end. The third mode shows bending vibration in the Y direction, with the bending occurring near the rotor core at the shaft's far end. The fourth mode shows bending vibration in the Y direction, with the bending occurring near the bearing at the rotor's shaft end, accompanied by significant expansion deformation, and the maximum displacement is the smallest among the four modes.

4. FLUID PRESTRESSED MODE

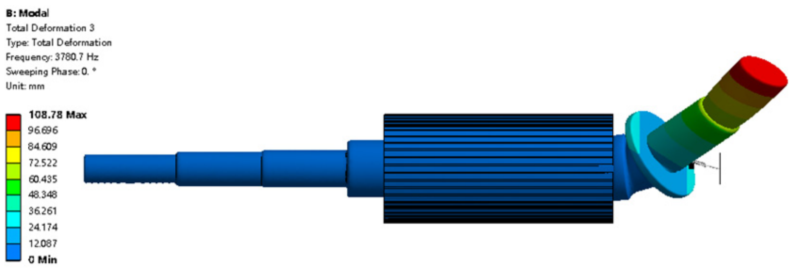
Pre-stressed modes, as a type of special mode, refer to modes where the structure itself contains a certain amount of pre-stress during modal analysis. Since the rotor system of the shielded motor used in the aviation pump rotates in a fluid, considering the fluid pre-stress provides modal results that more accurately reflect the actual operating conditions of the motor rotor during normal operation.



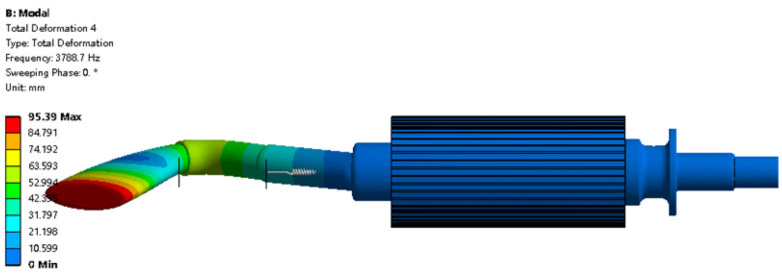
a) The first vibration mode of the rotor system



b) The second-order vibration mode of the rotor system



c) The third-order vibration mode of the rotor system



d) The fourth-order vibration mode of the rotor system

Figure 3. Modal vibration patterns under gravity and centrifugal force conditions

4.1. Consider the Fluid-Structure Coupling Prestressed Modal Analysis

The fluid-structure interaction (FSI) problem in the shielded motor refers to the interaction between the fluid field and the structural field. When the fluid inside the motor flows through the rotor components, there is an interaction between the fluid and the rotor components. The rotor components deform due to the pressure load generated by the fluid flow, which in turn affects the inherent characteristics of the shielded pump.

To investigate the impact of fluid-structure coupling forces on the natural frequencies of the rotor system, the results of the fluid field pre-stress will be compared with the dry modes discussed earlier. The natural frequencies of the rotor system under different operating conditions are shown in Table 3.

Table 3. Natural Frequencies under different working conditions

Working condition	First stage /Hz	Second stage /Hz	Third stage /Hz	Fourth stage /Hz
No prestress	1062	1067.7	3967.6	3976.5
Gravity and centrifugal force	941.35	943.03	3780.7	3788.7
Fluid-structure coupling	926	927.66	3766.7	3771.5

As shown in Table 3, compared to the dry modes, the natural frequencies of the rotor system at all orders decrease when considering fluid-structure coupling. Comparing operating conditions 1 and 3, it can be observed that the presence of pre-stress causes a decrease in the natural frequencies of the motor rotor system at varying degrees. The second-order natural frequency is most significantly affected, decreasing by 12%, while the third-order natural frequency is least affected, decreasing by 5%. Comparing operating conditions 2 and 3, it can be observed that after applying the fluid's effect on the structure, the natural frequencies of the rotor system decrease to varying extents. The primary reason for this is that the fluid adds additional mass to the rotor system and alters its damping effect. From these results, it is evident that fluid-structure coupling pre-stress influences the structural properties of the rotor system, and it is crucial to consider the impact of fluid-structure coupling pre-stress during practical analysis.

5. CONCLUSION

This paper takes a multi-physics approach, focusing on the special operating conditions of a liquid-immersed shielded motor. It performs coupled analysis of the motor's fluid field and stress field to investigate the interactions of different physical fields on the motor and reaches the following conclusions:

1.Pre-stress Effect: The pre-stress causes a reduction in the inherent frequencies of the rotor system across all modes to varying degrees. This reduction is non-negligible. Moreover, compared to the gravitational and centrifugal force conditions, considering the fluid field provides a more realistic reflection of the rotor system's dynamic characteristics.

2.Mode Shapes: In the case of fluid-structure interaction (FSI), the mode shapes of the rotor system are similar to those under other conditions, suggesting that the effect of pre-stress on the modal shapes of the rotor system is minimal. The primary impact of pre-stress is on the structure’s inherent frequencies.

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