Bakka Pushpa, 2024, 12:3
ISSN (Online): 2348-4098
ISSN (Print): 2395-4752

International Journal of Science, Engineering and Technology

An Open Access Journal

Dynamic Analysis on High-Speed Electrical Machines

Assistant Professor Bakka Pushpa
Department of Mechanical Engineering,
Holymary Institute of Technology and Science, Bogaram, Hyderabad, Telangana, India

Abstract- This paper centers around rotor dynamic investigation of the attractive levitation based fast electrical machine First and foremost, the powerful model of the attractive levitation rotor is laid out and the help solidness of the attractive heading is determined. Moreover, the rotor framework's dynamic investigation with ANSYS programming is carried out with the supposition that attractive bearing is considered as a specific firmness of spring model. Finally, the critical speeds and vibration modes of the rotor system are uncovered. In addition to providing theoretical support for the safe operation of the rotor system, the analytical result also contains the global stiffness and mass matrix coefficients [M] and [K], the damping matrix coefficient [C], and the external force vector [F(t)]. In the event that there is no outer power, it is free vibration condition and {F(t)}=0. The damping effect can be ignored when calculating natural frequencies and modes for free vibration. As a result, the free vibration equation will be simplified in the following manner: by utilizing lower-order critical speed control, but it will also serve as an important reference for the rotor's structure design and optimization.

Keywords- high-speed; vibration mode; stiffness; critical speed

I. INTRODUCTION

Current mathematically controlled (NC) machine instruments are creating toward high velocity and superior execution.

The dynamic performance of the electric spindle, which is an essential component of the high-speed grinding machine tool, has a direct impact on the grinding machine tool's maximum speed and machining quality. The electric spindle's dynamic performance is heavily influenced by its critical speed.

The working speed of the spindle is usually set so that it is as far away from its primary critical speed as possible, but not close enough to the critical speeds of the second, third, and other higher orders, in order to improve the spindle's self-centering performance and operation stability.

Scholars 1 through 3 have examined the factors that influence dynamic spindle performance: the spindle system's mass, the bearing's force on the spindle, and the types and parameters of the The HSK tool system and the electric spindle's grinding wheel for grinding, among other things The structural aspects of the electric spindle's influence on its critical speeds and change patterns are the primary focus of this investigation.

The critical speed of the rotor-bearing system of the electric spindle and related patterns of changes have been studied using the transfer-matrix method and taking into account the gyroscopic couple, shear, the variable cross-section, and other influential factors, such as the effects of the axial pre-tightening force of the bearing, the span of the fulcrum bearing, and the overhang variation at the front and rear ends.

II. DYNAMIC MODELING OF THE ELECTRIC SPINDLE

1. Model Simplification

The transfer-matrix method's lumped mass modeling principle indicates that the subsection points are situated at fulcrum bearings and variable cross sections. For longer spindle sections, additional subsection points are added to the rotor. The spindle is divided into a finite number of unit sections, with each section's mass, polar moment of inertia, and diameter moment of inertia distributed on an equivalent disk at both ends. Additionally, the following measures have been implemented to simplify the model:

The bearing can be represented as a spring with radial and angular rigidities of Kr, where Kr is the equivalent of the radial rigidities of two bearings connected in series and l is the distance between their centers. The site for the support bearing is set to be at the focal point of the line associating the focuses of the two sequentially associated orientation. b) The bearing rigidity is treated as a constant, and the effects of speed and load are not taken into account.

Because damping would result in a significant amount of error for the rolling bearing, it is ignored. Figure depicts the simplified calculation model. 1.

2. Transfer Matrix for the Typical Unit

The state Z of each unit node is expressed by four parameters: the shearing force Q, the bending moment M, the deflection Y, and the corner θ, or $[Z] = [Q\ M\ \theta\ Y]^T$. The transfer relationship between the state vectors of the left and right nodes of In the expression,4,5 l is the length of one unit section; m is the total mass at the node (the dial mass needed to be taken into account in addition to the section mass); Ω is the velocity of the rotor's precession angle; ω is the spin velocity of the rotor; Jd is the diameter moment of inertia of the rotor; JP is the polar moment of inertia of the rotor; EI is the flexural rigidity at the cross-section of a spindle section; $\nu = 6EI/k_tGAl^2$, where G is the coefficient of elasticity in shear; A is the cross-section area of a spindle section; kt is the shape coefficient for the cross-section (kt is 2/3 for hollow circular sections and 0.9 for solid circular sections); Kr is the equivalent radial rigidity of two serially connected bearings; Kθ is the equivalent angular rigidity of two serially connected bearings.

3. Solution of Critical Speeds and Mode of Vibration

The main limitations of the UHSEM design were that the current density shouldn't be higher than 15.5 A/mm2, that the temperature of the permanent magnets shouldn't be higher than 80 °C, and that the UHSEM efficiency should be higher than 80% at a rotational speed of more than 500 000 rpm. It will be additionally shown that super durable magnets Sm_2Co_{17} are utilized, which have a high working temperature. To achieve the maximum energy density in the UHSEM, a temperature limit of 80 °C is required because, at higher temperatures, both the energy characteristics of permanent magnets and the energy density in the UHSEM will significantly decrease. Without taking into account losses in bearings and sleeves, the UHSEM's efficiency should be greater than 75%. In view of the rotor elements conditions and the warm materials developments, the air hole ought to be at the very least 1 mm. The choice of the stator conductor In the designed UHSEM, copper losses and eddy-current losses in the winding are shown at a voltage frequency of 20 kHz. Thusly, high-recurrence Litz wires are regularly utilized in the UHSEM. For the UHSEM plan, the determination of the Litz strand width and the equal channel number are significant, wherein the copper and vortex current misfortunes are insignificant. The technology for winding wires is very simple. To choose the strand measurement and to ascertain the swirl current misfortunes, the Maxwell's condition framework depicting the conveyor attractive field was utilized. The value of the magnetic field at this particular point is what determines the current density at some strand point. The strand diameter ought to be significantly less than the magnetic field penetration depth in light of this condition. As a result, the magnetic field penetration depth is typically used to select the UHSEM strand diameter, and then the strand diameter is assumed to be two to four times smaller than the magnetic field penetration depth. The strand diameter is then adjusted if necessary to account for the eddy current losses in the selected diameter. The equations presented in [14] determine the magnetic field penetration depth. The vortex current misfortunes in the not entirely set in stone as

4. Mechanical Bearings

High-speed mechanical bearings with a rotational speed limit of 1.1 Mrpm and a short lifespan are the simplest option. The destruction of a ball bearing at a rotational speed of 1.1 Mrpm was demonstrated in particular in [7].

Ball bearings' primary advantages include their short operating times, high stiffness, and dependability. High friction losses are one of mechanical bearing's drawbacks; these losses can have a significant impact on UHSEM efficiency and long-term capability.

III. ANALYSIS OF STRUCTURAL PARAMETERS THAT AFFECT CRITICAL SPEEDS

1. Axial pre-Tightening Force

Bearing rigidity directly relates to the inherent frequencies of the first several orders of the entire electric spindle and their dynamic response and is determined by the bearing's pre-tightening degree. The progressions in the pre-fixing force (50-550 N) and basic rates of the initial three sets of the electric axle were determined, as displayed in overhangs was fixed.

It suggests that critical speeds of varying orders may be affected by the span of fulcrum bearings, but only within a certain range. Inside 135-185 mm, the essential basic speed expanded from 123,300 r/min to 125,520 r/min, an increment of 1.8%; a cuspidal point showed up at 185 mm. Therefore, the electric spindle design with a span of 185 mm was the best option for increasing the primary critical speed within the range.

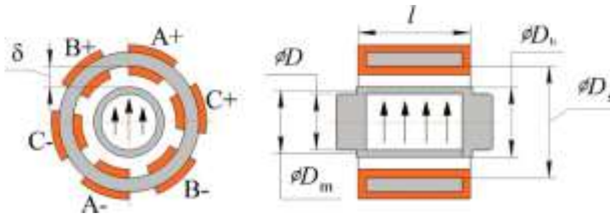

Fig 1 Axial pre-tightening force

2. Jib Length

Figure 2 when the axial pre-tightening force and the span of the fulcrum bearings were fixed and the spindle diameter at the overhang section remained the same, the curves of the influences of the front overhang on critical speeds are depicted in Figure 2. The front overhang reduced critical speeds of various orders, but only within a certain range, as shown.

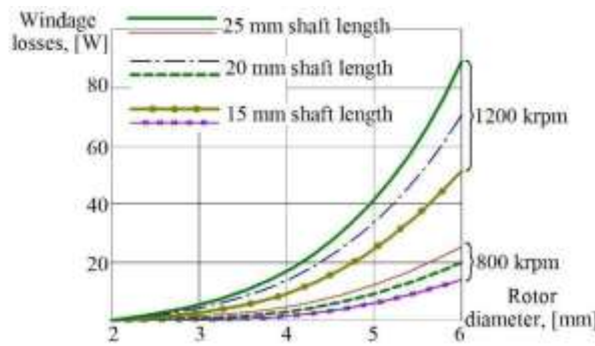

Fig 2 Jib length

shows the curves of how the rear overhang affected critical speeds when the spindle diameter at the overhang section was the same and the axial pre-tightening force and fulcrum bearing span were fixed. The rear overhang reduced the critical speeds of various orders, but only within a certain range, as shown in Figure 6. The rotor dynamics, thermal and mechanical extensions of the rotor bandage, and the minimum windage losses at the maximum magnetic flux density in the air gap are the criteria used in UHSEM to choose the air gap. The most effective air gap value for a UHSEM operating within the investigated power and speed range is demonstrated to be 1 mm in [18]. Experimental testing reveals that the friction coefficient is 0.006 [18]. (2) was used to calculate the windage losses. Figure 3 depicts the windage loss dependences on the UHSEM rotor diameter at various rotational speeds. These results indicate that a UHSEM design with a power output of less than 40 W and a rotation speed of 1.2 m/s is not recommended because it will not be possible to achieve an efficiency of nearly 50% at any rotor diameter.

IV. CONCLUSION

On the basis of the calculation results, the following conclusions are drawn:

Increasing the bearings' pre-tightening force has the potential to raise the critical speeds of various spindle orders, but it also has a high degree of sensitivity to the influence on higher orders. When the variation rates of the pre-tightening forces are compared to the critical speeds, it becomes clear that increasing the pre-tightening force has little effect on improving the critical speeds. (2) For critical speeds of various orders, increasing the span of fulcrum bearings has distinct influencing ranges. The critical speed decreases significantly within the range, while it fluctuates steadily outside of the range. (3) The fact that increasing the span of fulcrum bearings equates to decreasing the front and rear overhangs explains why increasing the front and rear overhangs has similar effects on critical speeds of different orders. All in all, the previously mentioned variables ought to be thought about in the underlying model of the shaft framework to get the sensibly arranged primary boundaries that fulfill the powerful exhibition of the electric shaft framework and thusly a sensible foundational layout for the shaft.

Acknowledgments
This research was financially supported by the Holy mary Institute of Technology and Science by Encouraging and Providing Laboratory facilities to complete my Research work successfully.

REFERENCES

1. D. S. Lee and D. H. Choi, Reduced weight design of a flexible rotor with ball bearing stiffness characteristics varying with rotational speed and load, Journal of Vibration and Acoustics-Transactions of the ASME 122(3) (2000) 203–208.
2. C. Nataraj and H. Ashrafiuon, Optimal design of centered squeeze film dampers, Jour- nal of Vibration and Acoustics-Transactions of the ASME 115(2) (1993) 210–214.
3. J.-S. Chen and K.-W. Chen, Bearing load analysis and control of a motorized high speed spindle, International Journal of Machine Tools & Manufacture 45(12–13) (2005) 1487–1493.
4. V. Gviniashvili, J. Webster and B. Rowe, Fluid flow and pressure in the grinding wheel-workpiece interface, Transactions of the ASME 127(2) (2005) 198–205.
5. M. N. Morgan, A. R. Jackson, H. Wu et al., Optimisation of fluid application in grind- ing, CIRP Annals-Manufacturing Technology 57(1) (2008) 363–366.
6. C. H. Li, C. Q. Cai, S. C. Xiu et al., Hydrodynamic pressure modeling and verifica- tion of contact zone on abrasive jet finishing with grinding wheel as restraint, Acta Armamentarii 28(2) (2007) 202–205.
7. F. Klocke, A. Baus and T. Beck, Coolant induced forces in CBN high speed grinding with shoe nozzle, Annals of the CIRP 49(1) (2000) 241–244.