Research on Bearing Radiation Noise and Optimization Design Based on Coupled Vibro-Acoustic Method

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***Abstract* - In bearings research, noise is an important evaluation index for mechanical property, in particular in muting machinery technology. Environmental pollution caused by bearing noise has always been the focus in bearing industry. In this paper we consider: i) sliding incentive of the rolling element bearing, and ii) its own variable stiffness incentive, to: i) accomplish the vibration coupling between the bearing and bearing seat, and ii) the coupling between bearing vibration and noise. This is achieved by means of combination of: i) kinetic model, ii) FEM method and iii) BEM method. The aim is to: i) build a perfect coupled vibro-acoustic model of the bearing, and ii) compare the test results and numerical simulation results of the bearing vibration velocity in order to verify the feasibility of the simulation model. Then the noise is calculated by the simulation model. Based on this, the improved design was carried out for low-noise rolling bearings, and the feasibility of the improvement program was verified by means of simulation.**

Keywords - Rolling bearing; Vibration velocity test; Coupled vibro-acoustic; Optimization design.

I. INTRODUCTION

For rolling bearings installed on bearing seats, the rolling bearing noise is mainly transmitted by two ways: firstly, the bearing noise directly spreads to the air; secondly, the bearing vibration excites the vibrations of the bearing seats, so the bearing noise is transmitted to the air through the bearing seats. Since the bearings are often installed in shells, with small radiation surface, so the noise transmission way of directly spreading into the air can be ignored; therefore, the bearing noise is mainly transmitted by the second way. To acquire the bearing noise, researchers should couple the bearings and the bearing seats firstly, then couple and calculate bearing seat vibration and noise; the results obtained in such a way are true and reliable.

Sunnersjo [1] considered inertia force and damping to research the variable stiffness effects theoretically, and carried out tests. Fukata [2] firstly researched nonlinear vibration caused by variable stiffness effects. He simplified single disk rigid rotor system under symmetrical simply support to single rolling bearing under rotating load, and carried out numerical analysis on the nonlinear vibration behaviour of the rotator system, which revealed the ultra-harmonics, sub-harmonics and chaotic behaviour of the rotator system. Mevel [3] also built a kinetic model of rigid rotator system of ball bearing support, and the obtained results were similar to Fukata's.

The working conditions and structure parameters of the rolling bearings have an important influence on the vibration characteristics of the entire system, in particularly the nonlinearity caused by radial clearance attracts wide attention. Childs [4] researched the influence of bearings' asymmetric clearance on rotator movement by means of nonlinear perturbation method. Saito [5] solved the nonlinear unbalanced response of horizontal Jeffcott rotator of ball bearing support with radial clearance, calculated the nonlinear vibration of the rotators by means of incremental harmonic balance method, and gave the approximate expression of nonlinear force. Lioulios [6] considered the radial clearance of the bearings, researched the influence of rotator speed fluctuation on nonlinear vibration. Harsha et al. [7] considered Hertz contact force, variable stiffness, radial clearance and surface waviness to research the influence of rotating speed on the nonlinear vibration characteristics of the ball bearings. Kappaganthu et al. [8] built a nonlinear kinetic model of rotator-bearing system and proposed bearing clearance model, researched the furcation and chaotic behaviour of the dynamic response caused by the bearing clearance.

Currently, bearing noise is mainly researched by means of expensive experiment test methods. There are few researches on algorithms for coupled vibro-acoustic between the bearing and bearing seat as well as the noise characteristics of bearings under the status when the rolling elements are sliding. In this paper, sliding incentive of the rolling element bearing and its own variable stiffness incentive were considered to accomplish the vibration coupling between the bearing and bearing seat as well as the coupling between bearing vibration and noise by means of combination of kinetic model, FEM method and BEM method, so as to build a perfect coupled vibro-acoustic model of the bearing and verify the effectiveness of the simulation model of the bearing by means of vibration velocity test. Based on this, improved design was carried out for low-noise rolling bearings, and the feasibility of improvement program was verified by means of simulation.

II. COUPLED VIBRO-ACOUSTIC MODEL OF BEARING-BEARING SEAT

A. Vibration model of bearing-bearing seat system

Vibration of bearing is the main source that causes bearing seat noise. Vibration characteristics can be acquired by building a vibration model of bearing-bearing seat system. A vibration model of bearing-bearing seat system was built by comprehensively considering the factors such as variable stiffness, clearance and friction due to sliding, etc. See Fig.1 for the bearing-bearing seat system; the rolling bearing was installed on the bearing seat, the bearing borne radial load W, its inner ring rotated at the angular speed of . See Fig.2 for the vibration model of the bearing-bearing seat system. In the model, the degrees of freedom of inner ring and outer ring on X and Y directions were considered. The rolling element was simplified to be a nonlinear spring which could only be compressed; the relationship between contact force and deformation was described by Hertz contact theory. A sliding pair was built between the rolling element and the inner ring; sliding friction was calculated according to Coulomb's Friction Law. The interaction between the bearing outer ring and the bearing seat was simulated by linear spring [9-10].

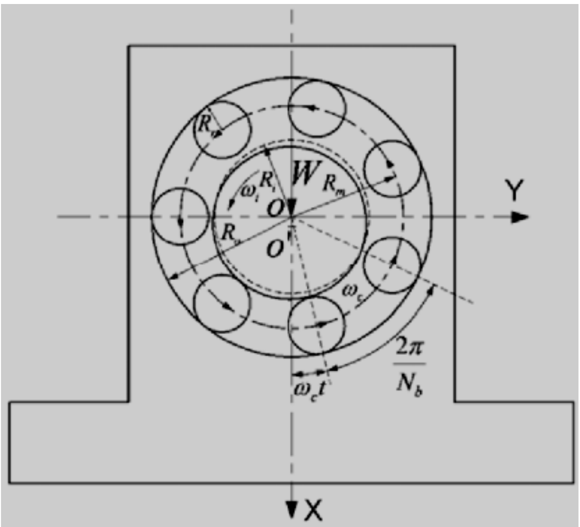

Fig.(1) Diagram of Bearing-bearing Seat System

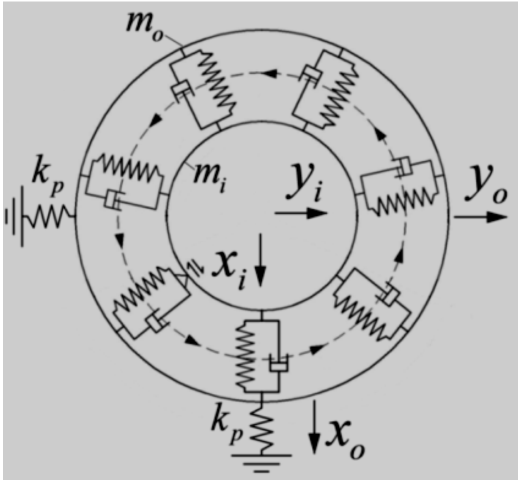

Fig.(2) Vibration Model of Bearing-bearing Seat System

The translation of the mass of inner and outer rings is considered in the model; the differential equation of the motion of bearing-bearing seat system is:

$$\begin{cases} m_i\ddot{x}_i + c\dot{x}_i + N_x + F_x = W \\ m_i\ddot{y}_i + c\dot{y}_i + N_y + F_y = 0 \\ m_o\ddot{x}_o + k_p x_o - N_x - F_x = 0 \\ m_o\ddot{y}_o + k_p y_o - N_y - F_y = 0 \end{cases} \tag{1}$$

Where x_i , y_i : Displacement of the barycenter of the bearing inner ring.

x_o , y_o : Displacement of the barycenter of the bearing outer ring.

m_i : Mass of bearing inner ring.

m_o : Mass of bearing outer ring.

c : Equivalent damping coefficient.

N_x : Total contact force on the bearing inner ring applied by all rolling elements at X direction.

N_y : Total contact force on the bearing inner ring applied by all rolling elements at Y direction.

F_x : Component force of the friction force due to sliding of the rolling elements at X direction.

F_y : Component force of the friction force due to sliding of the rolling elements at Y direction.

k_p : support stiffness of the bearing seat.

Vibration model of bearing-bearing seat system was second order nonlinear differential equations with four degrees of freedom; they were solved by means of fourth-

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order fixed step-length Runge-kutta numerical method [11-12].

B. Parameter Determination

B1. Contact force

The position angle (included angle with X axle) of the jth rolling element is the function of time t, which can be expressed as:

$$\theta_j = \omega_c t + \frac{2\pi}{N_b}(j-1) \quad j = 1, 2, ..., N_b \tag{2}$$

The contact deformation of the jth rolling element is the function of displacement of inner and outer rings, radial clearance and position angle, which can be expressed as:

$$\delta_j = \left[(x_i - x_o)\cos\theta_j + (y_i - y_o)\sin\theta_j - e\right]_+ \tag{3}$$

In the equation, e indicates the radial clearance of the rolling bearing, x_i and y_i indicate the displacements of inner ring at X direction and Y direction respectively, x_o and y_o indicate the displacements of outer ring at X direction and Y direction respectively. Subscript "+" indicates that the deformation is made to be zero when the value in the brackets is less than zero (rolling element is located in non-bearing zone).

The contact force of the jth rolling element can be indicated as:

$$N_j = K_n \delta_j^{1.5} \tag{4}$$

Where K_n is the total contact stiffness coefficient between the rolling elements and the inner and outer rings.

The inner ring and outer ring bear the contact force and friction force from all the rolling elements respectively. The resultant forces of contact force of all rolling elements at X direction and Y direction are respectively indicated as:

$$N_x = K_n \sum_{j=1}^{N_b} \left[(x_i - x_o)\cos\theta_j + (y_i - y_o)\sin\theta_j - e\right]_+^{1.5} \cos\theta_j \tag{5}$$

$$N_y = K_n \sum_{j=1}^{N_b} \left[(x_i - x_o)\cos\theta_j + (y_i - y_o)\sin\theta_j - e\right]_+^{1.5} \sin\theta_j \tag{6}$$

B2. Friction force

The friction force occurred due to sliding of the rolling elements when entering into the bearing zone can be indicated as:

$$F_j = \mu K_n \delta_j^{1.5} \tag{7}$$

In the equation, μ is coefficient of sliding friction.

The component forces of the friction force (occurred due to sliding of the rolling elements when entering into the bearing zone) at X direction and Y direction are respectively indicated as:

$$F_x = -\mu K_n \left[(x_i - x_o)\cos\theta_j + (y_i - y_o)\sin\theta_j - e\right]_+^{1.5} \sin\theta_j \tag{8}$$

$$F_x = -\mu K_n \left[(x_i - x_o)\cos\theta_j + (y_i - y_o)\sin\theta_j - e\right]_+^{1.5} \cos\theta_j \tag{9}$$

Where μ is coefficient of sliding friction, θ_j is the position angle of the rolling elements in sliding area. The rolling elements are provided with sliding friction forced when they are located in the sliding area; otherwise, the sliding friction force of the rolling elements is zero.

C. Finite Element Model

The finite element model of the bearing seat was built. Large grid division software ANSA was adopted for dividing grids of the bearing. To make it favourable for improving computational accuracy while controlling the scale of the finite element model, the grids of the model were divided in mapping mode by eight-node hexahedral elements, with the element type of SOLID185. See Fig.3 for the finite element model of the bearing seat and outer ring. The model included 20354 elements and 21568 nodes; the coordination and connection between the bearing outer ring and bearing seat were simulated by way of conode. The material of the bearing seat as well as inner and outer rings was linear elastic material whose parameters were: density ρ=7830 kg/m^3, elasticity modulus $E = 270GPa$, Poisson's ratio γ=0.3. The constraint plane and installation holes were fixed and constrained to restrain the degrees of freedom in XYZ three directions. The outer ring vibration response acquired by vibration model of bearing-bearing system was applied to the inner surface of the bearing seat by means of displacement mapping; transient dynamics simulation was carried out for the bearing seat by means of complete method in ANSYS.

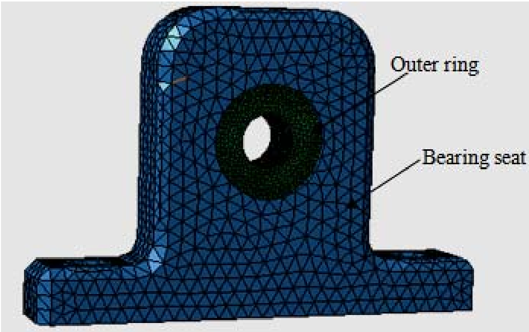

Fig.(3) Finite Element Model of the Bearing Seat

D. Boundary Element model

The radiation noise of the bearing seat was simulated by noise analysis software Virtual lab. The pre-processing for the boundary element model of bearing seat was completed in software ANSA; the external surface of bearing seat was extracted on the basis of finite element model of bearing seat, the grids were divided by shell elements to divide the external surface of the bearing seat into shell elements whose shape was consistent with the finite element model and whose element number and node number were one-to-one correspondent with the finite element model. The shell elements were imported to Virtual lab. The boundary element model of bearing seat was shown in Fig.4. The outer sound field of the bearing seat radiation noise was solved by direct boundary element method. In Virtual lab software, analysis type and air material parameters were set; the normal direction of the grids was checked; the normal of the boundary element model must direct the side where the fluid existed, otherwise, the normal direction of the model should be adjusted. The normal of the model grids outside the direct boundary elements directed outward. The vibration response of all nodes on the external surface was acquired by extracting transient finite element analysis on the bearing seat based on APDL language development program of ANSYS; it was applied to acoustic boundary element model of bearing seat as boundary conditions.

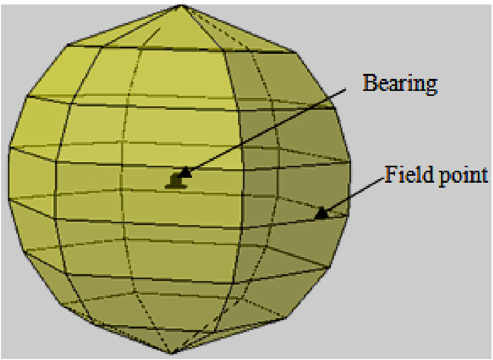

Fig.(4) Boundary Element Model of the Bearing Seat

III. SIMULATION MODEL VERIFICATION AND OPTIMIZATION DESIGN

A. Model Verification

To verify the effectiveness of the finite element model of the bearing seat proposed in this paper, vibration experiment was carried out on the bearing-bearing seat system. It was completed on bearing fatigue test bed which could realize application of bearing rotating speed and radial loading, as shown in Fig.5.

The rotating speed of the tested bearing was set as 900r/min; the calculated results of the finite element model and test results were compared and analysed. Fig.6 is comparison and analysis between the curves of vibration velocity of simulation and test. Fig.6 indicates that the simulation result agrees with the test result, which verifies the effectiveness of the finite element model proposed in this paper. Then, sound pressure is calculated by simulation as shown in Fig.7. It can be shown that there is a obvious peak value. We should take some measures to reduce the peak value.

Fig.(5) Vibration Experiment Facility

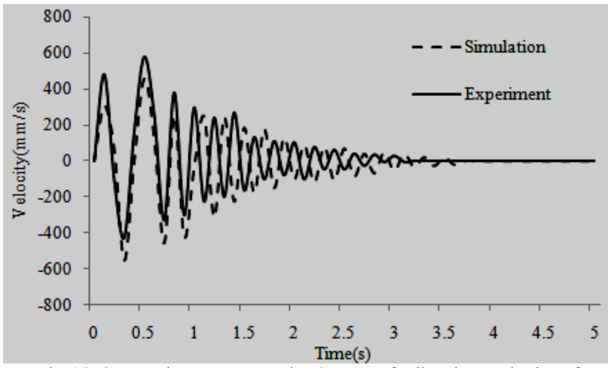

Fig.(6) Comparison Between the Curves of Vibration Velocity of Simulation and Test

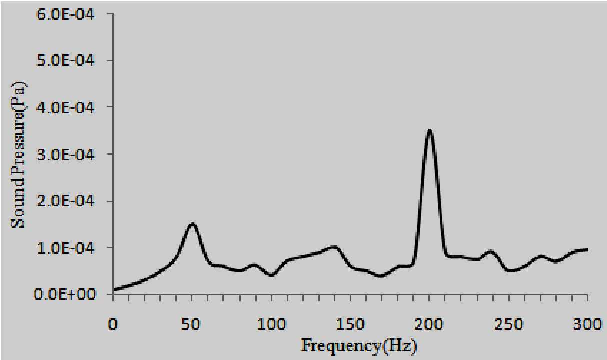

Fig.(7) Radiation Sound Pressure of Bearing

B. Optimization Design

To reduce bearing noise, increasing the preload is carried out, and the noise is calculated by the revised model. Sound pressure comparison between the origin and optimization model is shown in Fig.8. it indicates that the peak value has been reduced sharply.

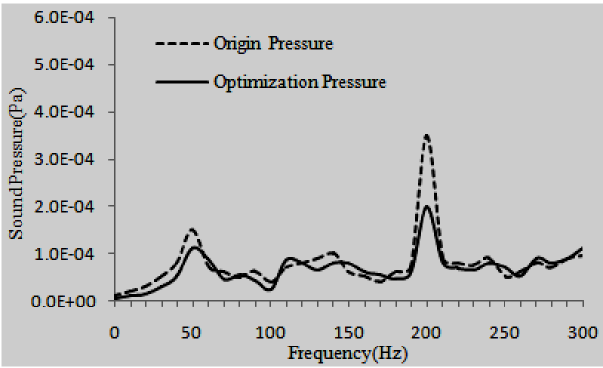

Fig.(8) Sound Pressure Comparison Between the Origin and the Optimization

IV. CONCLUSIONS

1) In this paper, sliding incentive of the rolling element bearing and its own variable stiffness incentive are considered to accomplish the vibration coupling between the bearing and bearing seat as well as the coupling between bearing vibration and noise by means of combination of kinetic model, FEM method and BEM method, so as to build a perfect coupled vibro-acoustic model of the bearing.

2) The bearing vibration experiment is completed by bearing fatigue test bed and LMS logger, which verifies the effectiveness of the simulation of the bearing in the paper, thus the noise is calculated by the simulation model, which provide a new thought and method for analysing and calculating bearing noise, and providing theoretical basis for developing low-noise rolling bearings.

3) Based on the conclusion, an improved rolling bearing program is proposed. Its superiority is verified by means of simulation.

CONFLICT OF INTEREST

The authors confirm that this article content has no conflicts of interest.

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