Noise and Vibration Analysis of Elevator Traction Machine

Ryo KAWASAKI\textsuperscript{A}; Yasuo HIRONAKA\textsuperscript{B}; Masaharu NISHIMURA\textsuperscript{C}

\textsuperscript{A}Mitsubishi Electric Corporation; \textsuperscript{B}Tottori University

Kawasaki.Ryo@ap.MitsubishiElectric.co.jp; Hironaka.Yasuo@eb.MitsubishiElectric.co.jp; mnishimura@mech.tottori-u.ac.jp

Abstract

Elevator car noise of 1000 Hz or less radiated from elevator traction machines is a particular problem because it is not easily insulated by the elevator car’s walls. A noise evaluation method for elevator traction machines has been developed. First, an excitation test is conducted with an excitation machine to develop a numerical model. Then, the structure-borne sound that radiated from the elevator traction machine excited by electromagnetic force is calculated. This force is obtained from magnetic field analysis using measured current and computed Fourier transforms of the time and space domains. The calculated results matched well with the measured results. The space orders of excitation force that had a large influence on the noise radiated from elevator traction machine are clarified.

Keywords: Electromagnetic Induced Vibration, Noise, Elevator, Method of Vibration Analysis

1 Introduction

Elevator traction machines are bigger and faster due to an increase in high-rise buildings. In addition, elevator traction machines have conventionally been installed in machine rooms in the roofs of buildings, but installing them into the elevator shaft is becoming mainstream. Therefore, the demand for low-noise elevator traction machines is increasing[1][2]. Elevator shafts are becoming narrower to increase the available floor space in buildings. Therefore, elevator traction machines are being made thinner to increase the space inside the elevator shafts. It is necessary to grasp the change of noise level of 1000 Hz or less that penetrates the elevator car walls easily according to a structural change.

In general, the harmonics frequency of electromagnetic force that works between the stator and the rotator corresponds to the natural frequency of the stator, the vibration generated, and the noise radiated from the housing that covers the outer layer of the stator and the rotator in the electric motor[3][4][5]. Therefore, the eigenmode of the stator and the noise...
radiation from the motor cover have been analyzed in conventional studies. However, an elevator traction machine pulls the rope to drive the elevator car with a sheave formed by one end of the rotor directly, so the rotor and sheave are located at the outer surface. The mechanism of noise radiation from the elevator traction machine is more complicated than in a conventional motor, so it is necessary to consider not only the housing but also the rotor and sheave in the vibration mode and noise radiation analysis.

For low-noise design technology and noise evaluation methods for thin elevator traction machines, this paper shows the shapes of electromagnetic force with a high influence in the noise radiated from elevator traction machines and introduces a technique to grasp the noise level radiated from elevator traction machine.

2 The structure of the elevator traction machine

The elevator traction machine used in this study has a permanent magnet motor. Figure 1 shows the front, and cross-section view of the elevator traction machine. The part with a large influence on the noise of this elevator traction machine is the housing and a stator located on the inside of the housing and a rotor and a sheave formed by one end of the rotor. The stator core is made of laminating steels. The teeth are inside the housing. The housing and the rotor are cast iron. Copper coils are wound around the teeth and attached with varnish. The rotor has permanent magnets on the outside. It is rotated by the electromagnetic force generated between a magnet on the rotor and teeth when current is applied to the coil. The rope that drives the elevator car is wound up to the sheave that is united with the rotor, so the elevator car moves when the rotor rotates.

The elevator traction machine is a thin motor. The diameter is bigger than the axial direction of the rotor because it is installed in the elevator shaft. In most motors, a housing covers up the most of rotors, but, in thin type elevator traction machines, the housing and not only the stator but also rotor and sheave united with a rotor are not covered. In other words, the noise radiation characteristics of the elevator traction machine are different from conventional motors in that noise radiates from the rotor and sheave as well as from the housing.

Therefore, it is necessary to analyze which characteristics of vibration and noise radiation characteristic of the housing, rotor, and sheave to grasp the car’s interior noise of remodeled elevator traction machine. There are several causes of interior elevator car noise. For example, the mechanical noise due to rope friction and braking, the noise due to torque change, and the noise due to radial electromagnetic forces. Especially, radial

Figure 1 - Elevator traction machine.
electromagnetic force contributes greatly to the noise equal to or less than 1000 Hz, so it is calculated by using radial electromagnetic forces as excitation forces.

3 Vibration Mode

3.1 Numerical Analysis

Figure 2 shows the analysis flow. The numerical model is based on drawings and eigenvalue analysis carried out with a limited element method. The teeth composed of laminated steel sheet and the copper wire hardened with varnish are modeled as solid elements. In this paper, border element method software VirtualLab is used for sound field analysis, electromagnetic field analysis software EMSolution is used for electromagnetic field analysis, and limited element method software ANSYS is used for eigenvalue analysis and frequency response analysis. Figure 3 shows modal shapes and natural frequencies of 1000 Hz or less, which is the frequency range at numerical analysis indicates that noise becomes a problem. A natural frequency mode of 536.5 Hz is the mode at which the rotor and sheave transformed into an oval, and the eigenmode of the rotor and sheave in the frequency band up to 1000 Hz is only this mode. The others are the eigenmodes at which the housing transforms into circular shapes, such as an oval, or a triangular shape, as in a conventional motor, in both the axial and radial directions.

Figure 2 - Analysis flow.

Figure 3 - Radial modal shape obtained from calculation.
3.2 Experimental Modal Analysis

Experimental modal analysis is carried out to study the validity of the models for numerical analysis and the modal damping ratio. Figure 4 shows a schematic of the test device. The test is carried out with the upper part of the elevator traction machine hung from two points by a chain block and the lower part elastically supported with isobutylene-isoprene rubber and no rope. An electrodynamic shaker (made by Wilcoxon, F3 type) is used for excitation. The vibration mode at which the stator core transforms into a circular shape and the housing transforms in the axial direction are thought to greatly influence the noise, so the shaker is installed on the housing to excite the modes in the axial direction. Three axis acceleration sensors (made by Endevco, 66A11) are used to measure acceleration. Figure 5 shows the installation position of the acceleration sensors. An experimental modal analysis device (made by LMS, Test.Lab) is used to analyze the excitation force and acceleration.

Figure 6 shows the damping ratio and eigenmode obtained from experimental modal analysis at less than 1000 Hz. Although there is an error of 6% with mode 5 in comparison with the natural frequency obtained by numerical analysis, the validity of the numerical model is confirmed because there is an error from 1% to around 3% in the other modes. The damping ratio of the eigenmode of the housing is around 1%. The damping ratio of the eigenmode of the rotor and the sheave is around 0.1%, so the radiated noise of the mode is thought to have a large contribution when electromagnetic force frequency and space shape order of excitation force accorded with the eigenmode of the rotor and the sheave. An analysis of the vibration and noise in operation is carried out using the damping ratio obtained from experimental modal analysis and a numerical analysis model.

![Experimental setup](image)

![Position of accelerometers](image)

![Modal shapes](image)
4 Electromagnetic vibration noise analysis

The vibration analysis is carried out by using electromagnetic force and the numerical model, and acoustic analysis is carried out by using the results of the vibration analysis as a boundary condition. In addition, the rotary speed of the rotor is sufficiently small in comparison with the propagation speed of a bending wave. Therefore, the influence of the rotation of the rotor is not considered in the vibration and acoustic analyses.

The equation of motion in the eigenvalue analysis of multiple degree of freedom system is shown in the next equation by using a reply vector \( \{x\} \) of the system[6].

\[
[M]\{x\} + [C]\{\dot{x}\} + [K]\{x\} = \{F\}
\]

where \([M]\) is the mass matrix, \([C]\) is the damping matrix, \([K]\) is the hardness matrix, and \([F]\) is the force vector. The natural frequencies of multiple degree of freedom systems are given by Equation (1), and the eigenmode matrix composed by the eigenvector is obtained.

The response of the physics coordinate system is shown in the next equation as the stack alignment of M modes by using orthogonality characteristics of the eigenvector.

\[
\{x\} = \sum_{r=1}^{M} \frac{\{\phi_r\}^T \{F\}\{\phi_r\}}{-\omega_r^2 M_r + j \omega C_r + K_r}
\]

where \(M_r\) is the modal mass of the \(r\) next, \(C_r\) is the modal damping, \(K_r\) is the modal stiffness, and \(\{\phi_r\}\) is a mode vector of order \(r\).

The force vector is calculated from the flux density and magnetic permeability between the stator and the rotor[7]. It is clarified that the force vector can be divided into time order \(h\) and spatial order \(s\) by performing a Fourier transform about a rotation direction position, and the time, and the force vectors are shown as follows[1].

Figure 7 shows the radial excitation force \(f_h\) of time order \(h\) at node \(n\) on the surface of the rotor. The excitation force \(f_h\) is equal to the sum of the force of the regular time order (traveling wave) and the force of negative time order force (retreat wave). The excitation force \(f_h\), the sum of all space orders of the traveling wave \(f_{1h}\), and the sum of all space orders of the retreat wave \(f_{2h}\) are expressed as

![Figure 7 - Exciting force.](image)
\[ f_h = f_{1h} + f_{2h} \] (3)

\[ f_{1h} = \sum_s \left( \frac{B_{h,s}^2}{2\mu_0} S_r \right) \left[ \frac{1}{N} \cos(\omega_h t + \theta_{h,s}) + j \sin(\omega_h t + \theta_{h,s}) \right] \] (4)

\[ f_{2h} = \sum_s \left( \frac{B_{h,s}^2}{2\mu_0} S_r \right) \left[ \frac{1}{N} \cos(\omega_h t + \theta_{2h,s}) + j \sin(\omega_h t + \theta_{2h,s}) \right] \] (5)

\[ \theta_{1h} = \theta_{h,s} + s\theta \] (6)

\[ \theta_{2h} = \theta_{h,s} - s\theta \] (7)

where \( B_{h,s} \) is the flux density of time order \( h \) and space order \( s \) obtained from magnetic field analysis using the measured current of a constant velocity drive test, \( S_r \) is the surface area of the magnet side of the rotor, \( \mu_0 \) is magnetic permeability, \( N \) is the number of nodes of the rotor surface, \( \theta_{h,s} \) is the initial phase of time order \( h \) and space order \( s \) at the standard position, \( \theta \) is the rotation direction coordinates of the node from the joint position, and \( \omega_h \) is a circular frequency of time order \( h \).

The acoustic analysis is carried out by using the vibration response obtained from Equation (2) as a boundary condition. The border element method is expressed for acoustic field analysis. The equation for the relation between a vibration response and the pressure caused by a vibration of the acoustical medium on the boundary surface is expressed by using the normal direction element \( \hat{n}_x \) of the response obtained from expression (2).

\[ \frac{\partial \hat{p}}{\partial \hat{n}} = -j \rho \omega \hat{x}_n \] (8)

where \( \rho \) is the density of the acoustical medium, \( \partial \hat{p}/\partial \hat{n} \) is the differential operator of the differentiation operator in the normal direction, and \( \omega \) is the circular frequency of the excited acoustical medium. In addition, the pressure \( p_l \) of element \( l \) when the boundary surface \( S \) is divided into the element of \( N \) units is expressed as Equation (9) by using Equation (8).

\[ p_l = -\frac{1}{2\pi} \sum_{n=1}^{N} \left\{ \frac{\partial \hat{p}}{\partial \hat{n}} \int_{S_n} \phi_{lm} dS - \rho \omega \int_{S_n} \frac{\partial \phi_{lm}}{\partial n} dS \right\} \] (9)

where \( \phi_{lm} \) is a basic function to satisfy a differential equation of Helmholtz about elements \( l \) and \( m \).

The sound pressure distribution in boundary surface \( S \) is calculated by using Equation (9), and the sound pressure level in an arbitrary point of an acoustic analysis domain is obtained.
5 Drive test

5.1 Contents
A constant velocity drive at the no-load condition with the elevator car speed at 100 m/min and an acceleration driving test at the no-load condition with the elevator car speed changing from 0 to 200 m/min are carried out. The acceleration of the housing and the noise level in the front 1 m position of the elevator traction machine are measured. Figure 8 shows the experimental setup.

The test is carried out with the upper part of the elevator traction machine supported from two points by a chain block and the lower part elastically supported with isobutylene-isoprene rubber. The noise is measured by a 1/2-inch microphone (B&K and used 4190-C). In the driving acceleration test, the elevator car accelerated from 0 to 200 m/min, and the maximum pressure level is measured by peak hold. The other measurement devices are the same as those used in the excitation test.

5.2 Constant Speed
Figure 9 shows the axial vibration of the results of the test and calculation at constant speed in the center of the housing sidepiece. Figure 10 shows the pressure level of the test and calculated pressure level at constant speed in the front 1 m position of the elevator traction machine. Each calculation result is a response analyzed with the time order ingredient of the electromagnetic force for the frequency of the driving current. Several similar tendencies such as the acceleration level of time order 2 maximum and pressure levels of time order 6 and 12 dominant are shown in Figures 9 and 10. This shows that this analysis technique is appropriate. It is thought that the natural frequency error of the analysis and measurement influence the error in the analysis and measurement.

5.3 Acceleration Drive
For elevator car speed acceleration from 0 to 200 m/min, for example, the frequency of time order 6 of the electromagnetic force changed from 0 to 880 Hz, and eigenmodes in the frequency band are excited.
Therefore it is thought that the eigenmodes shown in figure 3 influence the vibration and noise at the time of the acceleration. This is evaluated by the noise, because it is difficult to measure acceleration of moving rotor and sheave. Figure 11 shows the sound pressure level of the test and calculation results at constant speed in the front 1 m position of the elevator traction machine. Figure 11 shows that the analysis is sufficient to simulate the test. The eigenmode at which the noise level radiated from the elevator traction machine maximum in the frequency band of 1000 Hz or less is the housing mode of 791 Hz. In addition, the eigenmode of the rotor, which has a damping ratio at 536 Hz of 0.1%, had a large contribution to the radiated noise.

Reducing radiated noise by these eigenmodes makes it possible to build a low-noise elevator traction machine. Therefore, calculations are carried out to study the correlation between these eigenmodes and space order $s$ of excitation force by using the force of a single space order as the excitation force. Figure 12 shows the relation between the axial acceleration level of the housing sidepiece at a housing mode of 791 Hz and a space order of excitation force $s$, and Figure 13 shows the relation between the axial acceleration level of the rotor sidepiece at the rotor and sheave mode of 536 Hz.

Figures 12 and 13 show that the acceleration level of each eigenmode changed in every space order of excitation force. In particular, the excitation force of space order 2 had a large contribution to the acceleration level, so considering the relation between eigenmodes and space order $s$ of the excitation force makes it possible to grasp the change of noise level of 1000 Hz or less when the structure of the elevator traction machine changes.
6 Conclusions

A technique that divides the electromagnetic force provided by magnetic field analysis in spatial distribution of excitation force and calculates the vibration and noise is developed to study the change of noise level of 1000 Hz or less radiated from a remodeled elevator traction machine. The following conclusions are obtained by driving tests of the machine and calculations.

(1) This technique is able to analyze noise radiated from the outer surface of a motor in which a rotor is located with high accuracy.

(2) The eigenmodes of the thin elevator traction machine are obtained from a test, and it is clarified that housing transforms not only in the radial direction but also in the axial direction in circular shapes, such as an oval, or triangular shapes, as in conventional motors.

(3) The eigenmode not only of the housing but also of the rotor and the sheave greatly contributed to the noise radiated from the elevator traction machine. In addition, the change of the noise radiated from the remodeled elevator traction machine of 1000 Hz or less was obtained by considering the relation between the eigenmodes and space order of the excitation force.

References


