Activities and Results of the Rope Vibration Analysis Working Group in the Japan Society of Mechanical Engineers

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Abstract. Resonance of wire ropes or cables in a lift due to earthquakes and strong winds is a critical event for the safe operation of the lift, because if the ropes catch equipment in a shaft by the resonance, the passengers will be trapped and it will take a long time to rescue them. To prevent damage caused by rope vibration or resonance, it is important to estimate the vibration of ropes by simulation analysis in advance. However, the simulation method of rope vibration is complicated because it consists of various eigenmodes, the tension is not constant due to its own weight, and it is influenced by various factors, resulting in nonlinear vibration. Therefore, the Panel of Elevator Safety and Reassurance of the Japan Society of Mechanical Engineers (JSME) established the Rope Vibration Simulation Working Group. The working group studied simulation methods for rope vibration. This paper reports on the activities and results of the working group.

1 INTRODUCTION

Lifts are an important facility for high-rise buildings. However, lift ropes such as hoist ropes for suspending cages and counterweights and governor ropes for detecting overspeed have become longer, causing vibration-related problems. For example, in the Great East Japan Earthquake, rope snagging is one of the main causes of earthquake damage, and 24% of the damage to lifts was caused by rope snagging [1]. In addition, lift ropes in skyscrapers in Osaka Prefecture, 700 km from the epicentre, resonated, causing damage such as trapping of passengers [2].

When buildings resonate due to earthquakes or strong winds, the rope ends are forced to vibrate, and if the natural period of the building is close to that of the rope, there is a risk of resonance, which can cause snagging of ropes on rail brackets or other equipment in the hoistway. In addition, the early earthquake warning for long-period seismic motion has been put into operation in Japan since February 2023 [3], and lift control is one of the examples of its use, so resonance of the long ropes in lifts by earthquake is one of the important phenomena at present. The first step in preventing such damage is to properly estimate the response of the rope. Given the size and variety of buildings and lifts, simulation analysis is suitable for estimation of the rope vibration.

When performing simulation analysis, lift ropes can be represented by a string with time-varying length and location-varying tension, so partial differential equations have to be solved numerically, which requires a high level of expertise. As a result, vibration response analysis of ropes in lifts is carried out using each company's own methods or special calculation tools, and no generic, simplified method has been established that can be used throughout the industry.

From this background, the Panel of Elevator Safety and Reassurance of the Japan Society of Mechanical Engineers (JSME) established the Rope Vibration Analysis Working Group to investigate analytical methods for rope transverse vibration, to publish for various manufacturers and researchers, and to share and develop knowledge regarding rope vibration analysis. Firstly, the working group investigated vibration analysis methods and techniques related to wire rope vibration that have already been proposed. Then damping ratio of the wire rope, which is an important factor in determining the vibration amplitude, was investigated from the literature. Then seismic response analysis was carried out. This paper describes an overview of the activities and results of the Rope Vibration Analysis Working Group in the Japan Society of Mechanical Engineers.

2 ABOUT THE ROPE VIBRATION ANALYSIS WORKING GROUP

The Rope Vibration Simulation Working Group is a part of the Panel of Elevator Safety and Reassurance in the Transportation and Logistics Division, in the Japan Society of Mechanical Engineers (JSME).

The JSME, founded in 1897, is a Japanese society for mechanical engineering with more than 30,000 members, including researchers, engineers, teachers, students and so on. The JSME aims to improve and exchange the academic knowledge of the members and to return technological outcomes to society by publishing journals, bulletins, and textbooks, or by organising conferences, workshops, lectures, and international cooperation. The JSME has 22 technical divisions such as fluids engineering, thermal engineering, dynamics and so on, where more specialised discussions and activities are conducted.

The Transportation and Logistics Division is one of the technical divisions in the JSME and deals with technologies and research related to railways, automobiles, aerospace, ships, lifts, escalators and other vehicles. The Transportation and Logistics Division has 5 panels for further specialised and detailed discussions.

The Panel of Elevator Safety and Reassurance was established in 2011 to investigate and analyse issues related to lift and escalator technology. The Rope Vibration Analysis Working Group was established as a part of the Panel of Elevator Safety and Reassurance to investigate analytical methods and techniques related to wire rope vibration from 2015 to 2021. About 10 members, who are engineers or researchers from lift manufacturers and universities, participated.

3 INVESTIGATION OF ROPE VIBRATION ANALYSIS

3.1 Investigation of Analytical Methods

Firstly, the working group investigated the methods of rope vibration analysis that have already been proposed. According to the literature survey, the following 4 methods have been mainly proposed.

The first one is a method that expresses the rope vibration in the wave equation with time-varying length [4, 5]. The wave equation is calculated numerically by using the finite difference method. The second is a method that uses a single-degree-of-freedom model derived from the wave equation [6]. The vibration modal shape is assumed to be a sinusoidal wave, and the equation of motion is calculated numerically using the Newmark β method. Recently, an improved method of this one using the single-degree-of-freedom model has been proposed, which takes into account the coupling with the vertical motion of the cage [7]. The third one is a method using a multi-body dynamics model [8], which expresses a rope in many rigid elements connected with rotational springs and dampers. The fourth is a method using a finite element method [9].

Table 1 shows the comparison of the above methods for rope vibration analysis. The accuracy of the methods based on the wave equation, multi-body dynamics and finite element method is very good because they can represent the motion of each part of the rope, tension distribution, and interaction between vertical motion in detail. However, the method replaced with a single-degree-of-freedom model has a slightly lower accuracy compared with other methods, because the tension distribution and higher-order modes cannot be considered in exchange for the simplicity of the model. Methods based on multi-body dynamics and a finite element method require special software or computer programs to solve the vibration response. In addition, it is necessary to set the spring constants and the damping coefficients of a rope, which are difficult to determine from the rope specifications prior to numerical analysis. On the other hand, methods based on the wave equation and the single-degree-of-freedom model are easy to apply to vibration response analysis and have sufficient accuracy. Therefore, the working group specially focused on the methods based on the wave equation and the single-degree-of-freedom model.

Table 1 Comparison of rope vibration analytical methods

	Wave Equation	Single Degree of Freedom	Multi-body Dynamics	Finite Element Method
Accuracy	Very good	Good	Very good	Very good
Difficulty	Easy, but relatively difficult if car movement and tension distribution are considered	Easy	Difficult, calculation of spring constant and so on are required	Difficult, calculation of spring constant and so on are required
Generality	High	High	Special software is required	Special software is required

3.2 Investigation of Damping Ratio

The damping ratio of ropes is a very important parameter that directly determines the response amplitude. For example, if the damping ratio doubles, the response amplitude at resonance is halved. Therefore, the damping ratio of the transverse vibration of the rope was investigated in a literature survey.

Kimura et al. conducted free vibration experiments on full-scale hoist ropes to determine the damping ratio of the ropes [10]. The results showed that the damping ratio was generally 0.2%, although there were differences depending on the length and diameter. Kaneko et al. also conducted free vibration experiments on full-scale hoist ropes and the damping ratio of the ropes was 0.38% [8]. Crespo et al. used 0.3% of the modal damping ratio for their simulation analysis [11].

In addition, Utsuno [12] found that the damping of ropes obtained by experiment was smaller than the actual damping and investigated the effect of air resistance on rope vibration.

4 VIBRATION RESPONSE ANALYSIS

4.1 Analytical Procedure

Based on the above investigations, a transverse vibration response analysis was carried out in the working group. The working group adopted the following wave equation method, which is accurate and does not require special software.

$$\rho A\left(\frac{\partial^2 u}{\partial t^2} - 2V\frac{\partial u}{\partial t\partial z} + V^2\frac{\partial^2 u}{\partial z^2}\right) + C\left(\frac{\partial}{\partial t} - V\frac{\partial}{\partial z}\right)u - \frac{\partial}{\partial z}\left[T(z,t)\frac{\partial u}{\partial z}\right] = 0 \tag{1}$$

Where t is time, z is a spatial coordinate whose origin is the top of the building, $u(z,t)$ is the transverse displacement of the rope, ρA is a linear density of the rope, V is the velocity of the cage and C is the damping coefficient of the rope. As shown in Fig. 1, the tension $T(z,t)$ takes into account not only the distribution due to the weight of the rope itself, but also the variation due to the vertical motion of the car. The parameters shown in Table 2 were used. Theoretical natural frequencies f_n in Table 2 were calculated using the following formula for natural frequency of simple string vibration that takes into account the average tension due to the rope's own weight.

$$f_n = \frac{n}{2l}\sqrt{\frac{T_0 + \rho A g \frac{l}{2}}{\rho A}} \tag{2}$$

Where n is an order of vibration, l is the length of the rope, T_0 is the vertical load by weight of the car and g is the gravitational acceleration. The numerical analysis was carried out using the finite difference method.

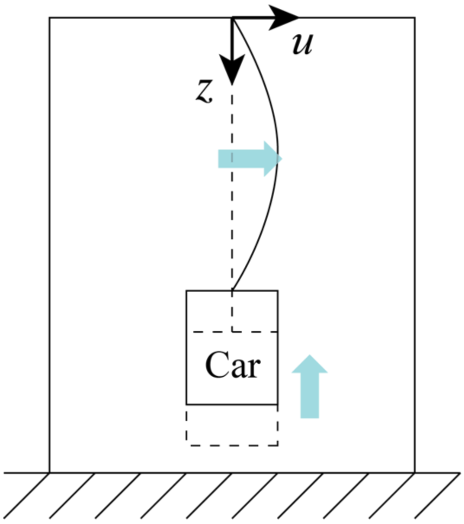

Figure 1 Comparison of resonance curve with the single-degree-of-freedom model

Table 2 Parameters of rope

Vertical load by car T_0 [N]	1960
Length l [m]	50
Linear density ρA[kg/m]	0.494
Damping ratio ζ[%]	0.2
Theoretical natural frequency f_n [Hz]	0.65, 1.30, …

4.2 Comparison with Single Degree of Freedom Model

To verify the accuracy of the simple method using the single-degree-of-freedom model, a comparison with the wave equation method was conducted. The single-degree-of-freedom model has the same natural frequency and damping ratio as the rope, namely 0.65Hz and 0.2%, respectively. Figure 2 shows the resonance curves for each method when a sinusoidal waveform is input. The result of the modal analysis, which extended the single-degree-of-freedom model to 5th order mode, is also shown in Fig. 2. The maximum response displacement of the rope decreases as the vibration mode increases for the same input acceleration, so vibration modes should be considered up to 4th or 5th order modes. Note that the vertical motion of the car is not taken into account here, because nonlinear vibration occurs if it is taken into account as described below.

As shown in Fig. 2, the result by the single-degree-of-freedom model for the first-order mode around 0.65 Hz is in agreement with the result by the wave equation and is sufficiently accurate. The result by the modal analysis also accurately represents the second-order mode around 1.30 Hz, although the amplitude is smaller than the result by the wave equation because the tension distribution is not considered. As discussed below, the response of the ropes becomes smaller when the vertical motion of the car is taken into account, so the single degree of freedom method estimates a larger response than the actual response, because the single degree of freedom model does not take into account the vertical motion of the car.

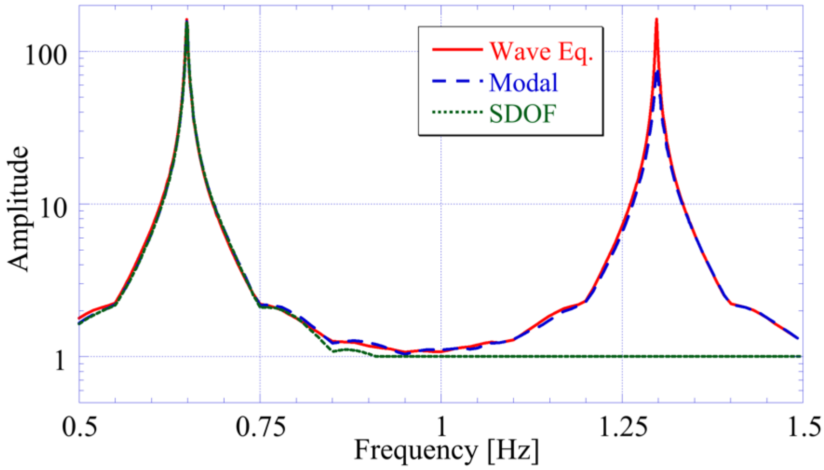

Figure 2 Comparison of resonance curve with the single-degree-of-freedom model

4.3 Influence of Vertical Motion of Car

In the vibration analysis of ropes, the amplitude at resonance is particularly important. Generally, the amplitude of a vibration system strongly depends on the damping ratio of the system, but according to experiments in the past, the actual response amplitude tends to be smaller than that estimated by the damping ratio obtained from free vibration tests. Assuming that the elongation of the rope is very small, the car moves upwards by the transverse vibration of the rope. The inertia force based on such a cage movement should cause variation in the rope tension. Therefore, the vertical motion of the car was taken into account in this section, and the effect of the vertical motion of the car on the response was investigated.

In this section, the vertical elongation of the rope and its attachment was ignored. This means that the displacement of the car z_{car} is calculated geometrically using the following equation.

$$z_{car} = l - \sum_{i=1}^{N-1} \sqrt{\Delta z^2 - (u_{i+1} - u_i)^2} \tag{3}$$

Where l is the length of the rope, Δz is the length of each element of the rope in the finite difference method, N is the number of the rope elements and u_i is the transverse displacement of the ith element of the rope. Then the vertical acceleration of the car $\ddot{z}_{car}$ was calculated from the second time derivative of the displacement z_{car}, and the inertia force by the vertical acceleration $\ddot{z}_{car}$ was considered in the tension of the rope at each step.

Figure 3 shows the influence of the vertical motion of the car on the resonance curve. As shown in Fig. 3, the amplitude and frequency at resonance decrease as the input displacement increases. This tendency is similar to that of nonlinear vibration, and the variation of tension caused by the vertical movement of the cage acts as a soft spring.

Figures 4 and 5 show the influence of the vertical motion of the car on the maximum amplitude and resonance frequency. As shown in Fig. 4, the maximum amplitude decreases rapidly and then converges as the input displacement increases. As shown in Fig. 5, the resonance frequency slightly decreases as the input displacement increases.

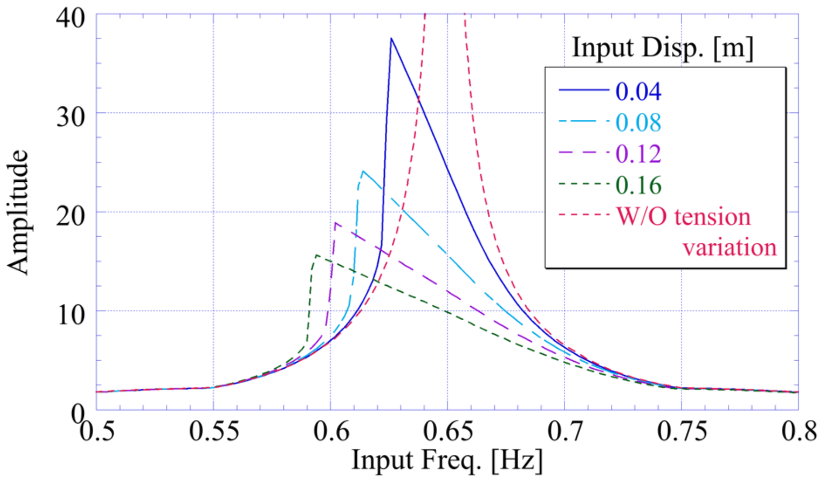

Figure 3 Influence of vertical motion of car on resonance curve

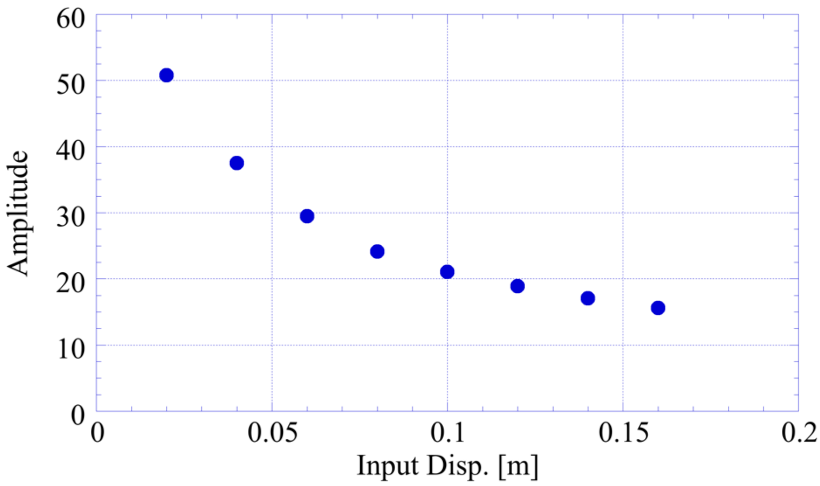

Figure 4 Influence of vertical motion of car on the maximum amplitude

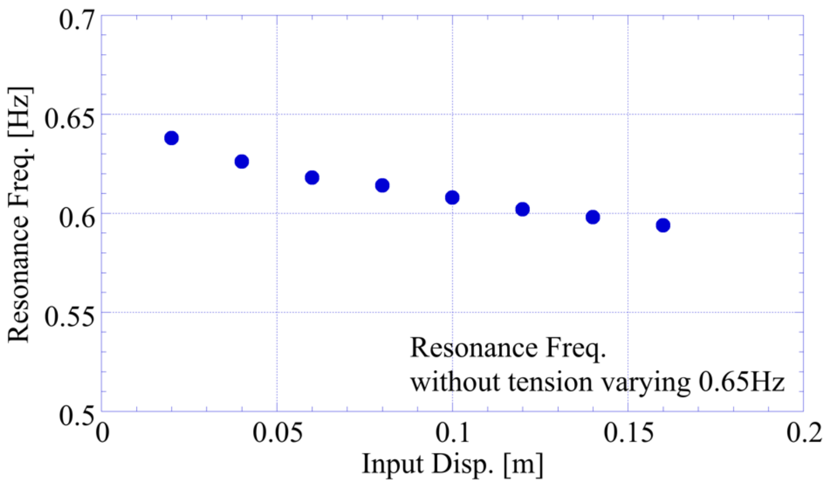

Figure 5 Influence of vertical motion of car on resonance frequency

4.4 Seismic Response Analysis

The seismic response analysis of the rope was carried out in this section. A response analysis using sinusoidal input waves was also performed to compare the seismic response with the response at resonance. Firstly, the seismic response of a building with a height equal to the rope length was analysed using a single degree of freedom model, and the resulting response was used as input to the rope. The input seismic waves to the building were El Centro NS waves, JMA Kobe NS waves and Hachinohe EW waves with maximum velocities of 0.25 and 0.50 m/s. The height of the building, i.e. the length of the rope, was 20, 40, 60, and 80m. The natural period of the building was 0.025 times the building height and the damping ratio was 2%. For the sinusoidal analysis, the top of the rope was vibrated with a sinusoidal wave.

The relationship between input and response displacement is shown in Fig. 6. The horizontal axis is the ratio of the maximum input displacement to the rope length, and the vertical axis is the ratio of the maximum response displacement to the input. As shown in the legend on the right-hand side of Fig. 6, each plot represents analytical results for the seismic response. The solid plots are results that take into account the tension fluctuations caused by the vertical movement of the car, and the open plots do not consider this. For comparison, the analytical results for the sinusoidal wave response are shown as lines.

From Fig. 6, the maximum response displacement is generally the same regardless of the rope length when a sinusoidal wave is input, and the maximum response displacement decreases as the input displacement increases. This is because the response displacement of the rope increases as the input displacement increases, and the vertical vibration of the cage also increases, so the nonlinearity of the vibration consequently increases.

Even when seismic waves were input, the maximum response displacement decreased as the input displacement increased. In addition, the maximum response displacement was smaller than that of a sinusoidal wave input because the time of excitation at the resonant frequency was short and the response did not grow. Therefore, the response due to seismic waves is on the safe side if the resonant response in sinusoidal waves is evaluated.

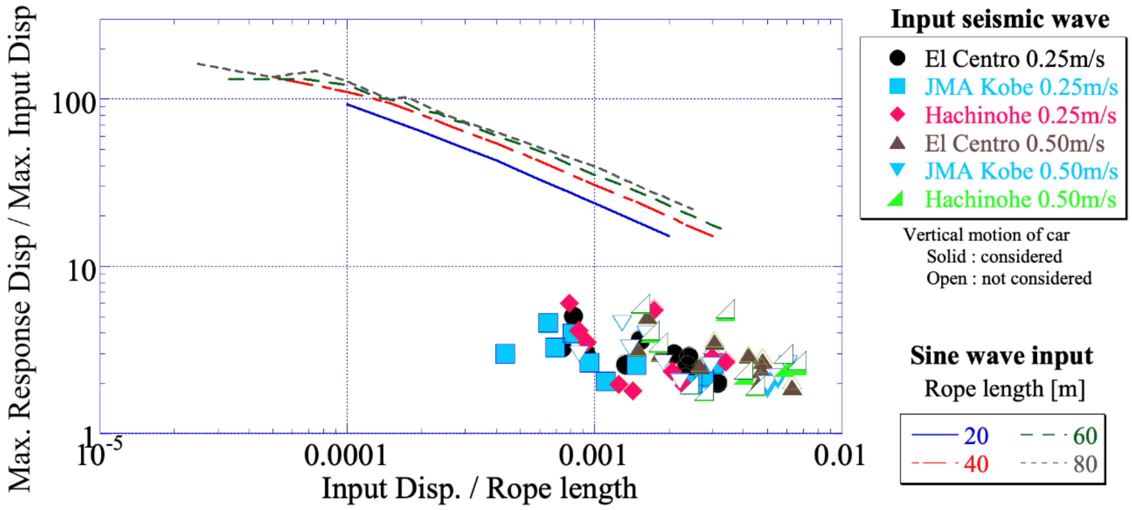

Figure 6 Maximum response displacement of seismic response

5 CONCLUSION

This paper described the activities and results of the Rope Vibration Analysis Working Group of the Japan Society of Mechanical Engineers. In the working group, previously proposed rope vibration analysis methods and rope damping were investigated through a literature review and vibration response analysis was performed. As a result of the working group's discussion, analytical methods based on the wave equation and the single-degree-of-freedom model are suitable for rope vibration analysis. The damping ratio of lift ropes is generally small, i.e. about 0.2%, but the amplitude will be small due to the non-linear vibration caused by the vertical motion of the car. It is expected that the analysis methods of the rope vibration will be further improved in the future based on the results of the working group.

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BIOGRAPHICAL DETAILS

Dr. Keisuke Minagawa is an associate professor at the Saitama Institute of Technology and is a chair of the Panel of Elevator Safety and Reassurance held in JSME (Japan Society of Mechanical Engineers). He has been an evaluator of lift systems and mechanical car parking systems in Japan since 2015. He is also an expert in seismic isolation and vibration control.

Prof. Satoshi Fujita, a JSME (Japan Society of Mechanical Engineers) Fellow, has ten years of management experience as a director, a dean of the school of engineering and a vice-president of Tokyo Denki University. He has been engaged in engineering research and development of seismic isolation systems and vibration control systems for buildings or key industrial facilities for over 35 years at both the University of Tokyo and Tokyo Denki University. In recent ten years, he has been a committee member of the Panel on Infrastructure Development of the Japanese Ministry of Land, Infrastructure and Transport (MLIT), and a chair of the Special Committee on Analysis and

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