### Roller–rail parameters on the transverse vibration characteristics of super-high-speed elevators

<table>
<thead>
<tr>
<th>Journal:</th>
<th>Transactions of the Canadian Society for Mechanical Engineering</th>
</tr>
</thead>
<tbody>
<tr>
<td>Manuscript ID</td>
<td>TCSME-2018-0083.R1</td>
</tr>
<tr>
<td>Manuscript Type:</td>
<td>Article</td>
</tr>
<tr>
<td>Date Submitted by the Author:</td>
<td>04-Dec-2018</td>
</tr>
<tr>
<td>Complete List of Authors:</td>
<td>Zhang, Ruijun; College of mechanical and electrical engineering, Shandong Jianzhu University Zhang, Shuohua; College of mechanical and electrical engineering, Shandong Jianzhu University Qiao, Shuai; Shandong Jianzhu University, School of Mechanical and Electrical Engineering Cong, Dongsheng; College of mechanical and electrical engineering, Shandong Jianzhu University Dong, Mingxiao; College of mechanical and electrical engineering, Shandong Jianzhu University</td>
</tr>
<tr>
<td>Keywords:</td>
<td>Super-high-speed elevators, Guide-rails, Bernoulli-Euler theory, Coupled vibration model, Model equivalent method</td>
</tr>
<tr>
<td>Is the invited manuscript for consideration in a Special Issue?</td>
<td>Not applicable (regular submission)</td>
</tr>
</tbody>
</table>
Roller–rail parameters on the transverse vibration characteristics of super-high-speed elevators

Ruijun Zhang, Shuohua Zhang*, Shuai Qiao, Dongsheng Cong, Mingxiao Dong

School of Mechanical and Electrical Engineering, Shandong Jianzhu University,
Jinan 250101, Shandong Province, PR China.

Email address: 884799801@qq.com.
Abstract: Interaction and wear between wheel and rail become increasingly serious with the increase in elevator speed and load. Uneven roller surface, eccentricity of rollers, and the looseness of rail brackets result in serious vibration problems of high-speed and super-high-speed elevators. Therefore, the forced vibration differential equation representing elevator guide rails is established based on Bernoulli-Euler theory, and the vibration equation of the elevator guide shoes and the car is constructed using the Darren Bell principle. Then, the coupled vibration model of guide-rail, guide shoes, and car can be obtained using the relationship of force and relative displacement among these components. The roller-rail parameters are introduced into the established coupled vibration model by using the model equivalent method. Then, the influence of roller-rail parameters on the horizontal vibration of super-high-speed elevator cars is investigated. Roller eccentricity and the vibration acceleration of the car present a linear correlation, with the amplitude of the car vibration acceleration increasing with the eccentricity of the roller. A nonlinear relationship exists between the surface roughness of the roller and the vibration acceleration of the car. Increased continuous loosening of the guide-rail results in severe vibration of the car at the loose position of the support.

Keywords: Super-high-speed elevators; Guide-rails; Bernoulli-Euler theory; Coupled vibration model; Model equivalent method; Surface roughness.
1 Introduction

At present, the number of super-high-rise buildings increases on a daily basis. Thus, large load and high speed are inevitable in elevator development. Wear between rollers and guide rails becomes increasingly serious with the continuous improvement of the elevator running speed and increasing loads. Uneven wear and interaction between rollers and rails lead to uneven roller surface and loose rail brackets, which cause serious vibration problems to high-speed and super-high-speed elevators. Reduction of elevator comfort also shortens the life of its internal precision parts, thereby decreasing elevator safety.

Due to the operation of the product cannot be in the ideal environment, the study of the dynamic characteristics of the product under the non-ideal parameters remains an active area of research (Wang et al., 2018; Wang et al., 2016; Azzara et al., 2018). For the elevator, roller surface roughness, roller roundness, and the looseness of the rail brackets are related to the wear and strength of the roller rail. For investigating the change in roller–rail parameters (namely, uneven surface, deviation from roundness of rollers, and loosening of rail brackets) that influence the horizontal vibration of super-high-speed elevators, a mathematical model that can accurately describe the relations between lift systems should be constructed. Li et al. simplified the guide shoe system as a spring damper system, established a two-degree-of-freedom horizontal vibration model of an elevator car, simulated the vibration model based on the analysis of the incentive of elevator guide, and provided an effective simulation method for the horizontal vibration of an elevator (Li et al., 2002). Fu et al. derived the horizontal vibration differential equation, by using the conversion between the local and global coordinate systems, discussed the disturbance model of the guide rail, compared the acceleration response of the elevator under the sine, triangle, pulse, and step disturbances, and concluded that step disturbance of the guideway causes large horizontal vibration (Fu et al., 2003). Mei et al. considered the nonlinear rate dependent characteristics of rubber boot linings and established a 3D rolling contact model (Mei et al., 2009). Li et al. established a coupling dynamic model that included the elevator frame, roller, and guide rail starting from roller–rail contact stiffness but disregarded contact damping between roller and rail and contact stiffness between roller and car (Li et al., 2005). Guo et al. established a dynamic coupling model of elevator car, guide shoe, and guide rail and studied nonlinear factors between guide shoes and guide rails. The mathematical model of the elevator coupling system has not been widely used (Guo et al., 2011). Li et al. and Guo et al. considered the coupling relation of the elevator system (Li et al., 2005; Guo et al., 2011). However, all these studies did not consider contact damping between rollers and rails.

Xia et al. analyzed the effects of guide rail roughness, eccentricity of the guide wheel, and flat scar on the high frequency vibration of the elevator (Xia et al., 2012). The literatures regarding the influence of roller rail parameters on the horizontal vibration of elevators is limited, but relevant research on railway locomotive and track dynamic interaction is mature. Zhao et al. studied the influence of bridge stiffness on the dynamic characteristics of the vehicle–bridge coupling system and emphasized that the relative dynamic parameters of the vehicle and the bridge increase sharply with the decrease in stiffness (Zhao et al., 2017). Gao et al. analyzed the influence of structural parameters on orbit subsidence by using the axle–bridge coupling dynamic theory and identified vehicle running speed, axle load, and line traffic as main factors (Gao et al., 2007). Xiong et al. discussed the influence of each wear type on the safety and stability of the railway operation through the analysis of the typical wheel–rail wear form of heavy-duty railways (Xiong et al., 2014). Kouroussis et al. studied the influence of certain typical railway vehicle and track parameters on the
level of ground vibrations induced in the neighborhood (Kouroussis et al. 2012). Gernoth Götz et al. identified the influence of varying input parameters on the results of model validation (Gernoth Götz et al., 2017). Suarez et al. analyzed the influence of track quality and the properties of wheel–rail rolling contact on vehicle dynamics (Suarez et al., 2013).

In this paper, the close relationship between roller–rail parameters and elevator vibration under super-high-speed condition is considered. The physical model of wheel and rail parameters is established by referring to the research methods of railway locomotives and orbit dynamic interaction based on the dynamic model of super-high-speed elevator coupling. Then, roller–rail parameters are numerically discretized. The influence mechanism of the wheel–rail parameters on the horizontal vibration of super-high-speed elevators is studied by using step-by-step integration.

2 Model of coupling system of car, roller, and guide rail for super-high-speed elevator

The following assumptions and simplifications are developed considering the structure and motion law of super-high-speed elevators and the convenience of theoretical research:

(1) The car body is rigidly connected to the car frame;
(2) The mass center of the car coincides with the geometric center;
(3) The car and the roller are simplified as a mass spring damping system because the movement of the car relative to the roller and that of the roller relative to the guide rail on the normal rail surface are considerably small;
(4) The structure and parameters of each roller are the same;
(5) Guide rails on one side are simplified as a multi-span elongated straight beam (Zhu et al., 2013).

The horizontal vibration of an elevator system includes lateral vibration (y-direction) and front and rear vibration (x-direction). Lateral vibration is similar to the front and rear vibration mechanical model, and modal frequency is close to each other. Thus, the two vibration types are often studied separately. The vibration of guide rails exerts considerable influence on the vibration of an elevator system under the coupled state of the guide rail, guide shoe, and car system. Therefore, this research mainly studies the horizontal vibration of guide rails under the wind load that acts on a building (namely, horizontal vibrations of guide rails and vibrations that follow the passage, which occur in the y-direction). Fig. 1 shows the horizontal vibration model of a super-high-speed elevator that considers the wind-induced vibration of buildings.

Fig. 1 Horizontal vibration model of the rail, guide-shoe, and car coupling system
In Fig 1, \( l \) is the guide-rail length, \( m_0 \) is the mass of the guide-rail unit length, \( EI \) is the bending stiffness of the guide rail, \( c_0 \) is the damping of the guide rail, \( S \) is the distance in the \( x \) direction of the upper and lower rollers, \( y_1 \) is the elastic deformation of the \( t \) time of the guide rail 1, and \( y_2 \) is the elastic deformation of the \( t \) moment of the guide rail 2. \( y_{w1}, y_{w2}, y_{w3}, \) and \( y_{w4} \) are the horizontal displacements of rollers 1, 2, 3, and 4, respectively. \( m \) is the mass of the roller, \( k_1 \) is the contact stiffness between the roller and the guide rail, \( c_1 \) is the contact damping between the roller and the guide rail, \( k_2 \) is the connection stiffness between the roller and the car system, and \( c_2 \) is the connection damping between the roller and the car system. \( y_c \) and \( \theta \) are the horizontal and angular displacements of the car system, respectively. \( m_c \) is the quality of the car system, and \( J_c \) is the moment of inertia of the car system.

2.1 Horizontal vibration model of the super-high-speed elevator car, guide shoe, and guide rail coupling system

Guide rails are fixed to a building at a predetermined distance from the rail support. The guide rail is considered an Euler beam owing to the horizontal deflection of the guide rail, which is markedly smaller than the length. The Bernoulli–Euler theory indicates that under the coupled interaction of the guide shoe and the rail, the differential equations of the forced vibration of guide rails 1 and 2 can be expressed as follows:

\[
EI \frac{\partial^4 y_1(x, t)}{\partial x^4} + m_0 \frac{\partial^2 y_1(x, t)}{\partial t^2} + c_0 \frac{\partial y_1(x, t)}{\partial t} = -\delta[x - s_1(t)]f_1(x, t) - \delta[x - s_2(t)]f_2(x, t),
\]  

\[
EI \frac{\partial^4 y_2(x, t)}{\partial x^4} + m_0 \frac{\partial^2 y_2(x, t)}{\partial t^2} + c_0 \frac{\partial y_2(x, t)}{\partial t} = \delta[x - s_1(t)]f_1(x, t) + \delta[x - s_2(t)]f_2(x, t).
\]  

For the car system and the roller, the unevenness of the rail surface is an important excitation source that triggers vibration. The unevenness of the rail surface affects the vibration of the guide rail itself and thus influences the contact force of the roller and the rail. To conveniently study the relationship between the rail surface roughness and guide rail vibration, we assume identical surface roughness of guide rails 1 and 2 at the same lifting height. The surface roughness value of the guide rail at the contact point between the lower roller and guide rails 1 and 2 is \( r(s_1(t)) \). The surface roughness value of the upper roller and guide rails 1 and 2 is \( r(s_2(t)) \). Contact force can be expressed as follows:

\[
f_2(x, t) = k_1 \left[ y_{w2} - y_2(\ x, \ t) - r(s_1(t)) - \varepsilon(t) \right] + c_1 \left[ y_{w2} - y_2(\ x, \ t) - r(s_1(t)) - \varepsilon(t) \right],
\]  

\[
f_1(x, t) = k_1 \left[ y_1(\ x, \ t) - y_{w1} + r(s_1(t)) + \varepsilon(t) \right] + c_1 \left[ y_1(\ x, \ t) - y_{w1} + r(s_1(t)) + \varepsilon(t) \right],
\]  

\[
f_3(x, t) = k_1 \left[ y_1(\ x, \ t) - y_{w2} + r(s_2(t)) + \varepsilon(t) \right] + c_1 \left[ y_1(\ x, \ t) - y_{w2} + r(s_2(t)) + \varepsilon(t) \right],
\]  

\[
f_4(x, t) = k_1 \left[ y_{w4} - y_2(\ x, \ t) - r(s_2(t)) - \varepsilon(t) \right] + c_1 \left[ y_{w4} - y_2(\ x, \ t) - r(s_2(t)) - \varepsilon(t) \right].
\]  

where \( \varepsilon(t) \) is the roller–rail parameter and \( \varepsilon(t) \) is the derivative of \( \varepsilon(t) \). The models are presented in Section 2.2.

Damping force between the car system and the roller is determined by instantaneous horizontal
velocity, rotational angular velocity of the car system, and instantaneous horizontal velocity of the wheel. Elastic force between the car system and roller is determined by the instantaneous level of the car system displacement, rotation angular displacement, and the momentary displacement of the roller. Considering that the outer edge of the roller is a vibration damping rubber and the relative displacement between the roller and the guide rail is small, we simplify the roller as a mass spring damping system connected to the guide rail by a spring and a damper. The Dalangel principle is used to establish the motion equations of four rollers as follows:

\[ m \ddot{y}_w + c_1[y_w - y_c] + k_1[y_w - y_c + \frac{1}{2} h \theta] = f_1, \]  
\[ m \ddot{y}_w + c_2[y_w - y_c] + k_2[y_w - y_c + \frac{1}{2} h \theta] = -f_2, \]  
\[ m \ddot{y}_w + c_3[y_w - y_c] + k_3[y_w - y_c + \frac{1}{2} h \theta] = f_3, \]  
\[ m \ddot{y}_w + c_4[y_w - y_c] + k_4[y_w - y_c + \frac{1}{2} h \theta] = -f_4. \]

Similarly, the pattern of connection between the car and the roller is simplified as a spring and damping system owing to a shock-absorbing rubber between the car and the car frame, a shock-absorbing spring between the shoes, and small relative displacement between the car and roller. The car system is considered as a rigid body, and its horizontal movement includes translation along the \(y\)-axis and rotation around the centroid. The Dalangel principle is used to establish the following differential equations for the overall horizontal and rotational displacements of the car system under the coupling of the car system and the roller:

\[ m \ddot{y}_c + 4c_2 \ddot{y}_c(t) + 4k_2 \dot{y}_c(t) = c_3[y_w + \dot{y}_w(t) + y_w(t)] + c_2[y_w(t) + y_w(t)] + k_3[y_w(t) + y_w(t) + y_w(t) + y_w(t)] + k_4[y_w(t) + y_w(t) + y_w(t) + y_w(t)] \]
\[ J \ddot{\theta} + c_2 h^2 \dot{\theta}(t) + k_2 h^2 \dot{\theta}(t) = \frac{1}{2} c_3 \dot{y}_c(t) + \dot{y}_c(t) - \dot{y}_c(t) - \dot{y}_c(t) \]
\[ + \frac{1}{2} k_4 \dot{y}_c(t) + \dot{y}_c(t) - \dot{y}_c(t) - \dot{y}_c(t) \]

2.2 Roller–rail Parameter Excitation Analysis

Roller eccentricity and the unevenness of the roller surface can be expressed by the equivalent roughness of the rail. The equivalent model is shown in Fig. 2.

Equivalent roller radius

\[ \tilde{R} = R - r, \]

where \(R\) is the radius of the roller and \(r\) is the roller eccentricity.

Roller eccentricity is usually expressed by the trigonometric function (Zhang et al., 2005),

\[ r_e(t) = r \cos(\theta), \text{ such that } \theta = \frac{v}{R} t. \]

The surface roughness of the roller (SR) is equivalent to the roughness of the rail surface, which is simulated by a random number.
Loosening of rail brackets leads to the decrease in the supporting strength of the guide rail joint. This study simulates the lack of support caused by rail-bracket looseness with the increase in guide-rail span. The change in rail span directly affects the vibration mode function and vibration mode damping of the rail. This change also conforms to the change in rail structure and damping caused by rail-bracket looseness. The specific relationship is given by the following:

The damping of the $i^{th}$ order vibration of the guide rail is as follows:

$$c_i = 2\xi m\omega_i$$  \hspace{1cm} (7)
where $\xi$ is the damping ratio and $\omega_i$ is the circular frequency of order $i$, such that $\omega_i = \frac{\sqrt{E}}{l^2} \sqrt{\frac{m}{I}}$.

3 Solution to Coupled Equations

Any reasonable displacement of the structure can be represented by the superposition of the individual structure modes with the corresponding amplitudes. This study uses mode decomposition to transform the partial differential equation of the guide rail into an ordinary differential equation as follows:

$$y_j(x, t) = \sum_{i=1}^{n} q_{ij}(t) \alpha_i(x), \quad j = 1, 2,$$

(8)

where $q_{ij}(t)$ refers to the generalized mode coordinates of the guide rail, $\alpha_i(x) = \sqrt{\frac{2}{n l}} \sin \left(\frac{\pi i x}{l}\right)$ is the vibration mode function of the guide rail, and $n l$ is the total length of one side of the guide rail.

To obtain the $n$th order general coordinate motion equation of the guide rail, we substitute Equation (8) into Equations (1) and (2). Then, the two sides of Equations (1) and (2) are multiplied by $\alpha_i(x)$ and integrated on the interval $[0, l]$. The $n$th order general coordinate motion equation of the guide rail can be obtained using mode shape orthogonality as follows:

$$m_{ij1}(t) + c_{ij1}(t) + k_{ij1}(t) = f_{i1},$$

(9)

$$m_{ij2}(t) + c_{ij2}(t) + k_{ij2}(t) = f_{i2},$$

(10)

where $k_{ij}$ is the calculation stiffness of the guide rail under a generalized coordinate, and $f_{i1}$ and $f_{i2}$ are the generalized load vectors of guide rails 1 and 2, respectively.

This study considered the generalized coordinate motion Equations (10) and (11) of the guide rail under coupling action and motion Equations (3)–(6) of the car and rollers. After consolidation, the coupling dynamic equation of the system can be merged as follows:

$$M \{ \ddot{Y} \} + C \{ \dot{Y} \} + K \{ Y \} = \{ F(t) \},$$

(11)

where $M$, $C$, and $K$ are the mass, damping, and stiffness matrices, respectively, of the coupled system. $\{ Y \} = [y_1, \theta_1, y_2, \theta_2, y_3, \theta_3, q_1, q_2]^T$ is the generalized displacement vector of the system, and $\{ F(t) \}$ is the generalized load vector of the system that considers the wheel and rail preload.

The coefficient of the system dynamic equilibrium Equation (11) changes with the change in position of the guide shoe on the guide rail. Thus, the equation comprises second-order differential equations with time-varying coefficients. This group of time-varying coefficient differential equations can be solved only through a stepwise numerical method (Kai et al., 2013). A step-by-step integral program with discrete variables is compiled based on incremental method to solve the system. The program considers the influence of the structural parameters on the dynamic characteristics of the guide rail and the nonlinearity of the equation group (Rezaiee-Pajand et al., 2016; Muñoz et al., 2017).
4 Simulation results and analysis

4.1 Calculation parameter analysis

The large-span discrete values of the roller–rail parameter are selected to completely investigate the influence of the parameter on the vibration of super-high-speed elevators.

A super-high-speed elevator with a rated running speed of 20 m/s is the research object of the simulation. Table 1 shows the main input parameters of the simulation.

4.2 Influence of roller surface unevenness on elevator vibration

The dynamic response of the car under the wear of the roll guide and roller surface is simulated using the method of the third section based on the model established in the second section. The simulation results are shown in Fig. 4 as follows:

Fig. 4 shows that the vibration acceleration of the car increases with the increase in the wear degree of the roller surface. When the surface of the roller is considered as a smooth surface without wear, the vibration acceleration amplitude of the car is small at $10^{-3}$ m orders of magnitude. When the wear degree of the roller surface reaches the mean value of $1 \times 10^{-4}$ m, the vibration acceleration amplitude of the car increases significantly at $10^{-2}$ m orders of magnitude. Compared with that of smooth rollers, the vibration acceleration amplitude of the car increases by...
approximately 10 times. With increasing degree of roller wear, the amplitude of vibration acceleration of the car gradually increases.

4.3 Influence of roller roundness deviation on elevator vibration

Fig. 5 shows that the acceleration amplitude of the car gradually increases with increasing deviation from roundness of the roller in a linear relationship. However, no tendency toward
stability is observed. Fig. 5e. shows a partial enlarged view of Fig. 5d. The curve of car vibration acceleration shows a periodicity change, which is attributed to the non-roundness of rollers. Two opposite direction peaks in each period are observed.

4.4 Influence of rail-bracket looseness on elevator vibration

Fig. 6. Vibration acceleration curve of car with different kind of rail-bracket looseness

(a) The middle one of three consecutive rail-brackets is loose

(b) The middle two of four consecutive rail-brackets is loose

(c) The middle three of five consecutive rail-brackets is loose

Compare Fig. 6(a) and Fig. 4(a), the change in the car vibration acceleration curve is not evident if only the middle one among three successive guide brackets is considered loose. Severe vibration will be observed when the car is operating near the loose position of the guide brackets, given two occurrences of loosening in the middle two of the four guide brackets or three loosening occurrences in the middle three of the five rail brackets. In addition, the peak value of the vibration acceleration increases with the increase in the number of rail brackets with continuous looseness.

5 Conclusion

The three aspects in this study, namely, roller surface, roller eccentricity, and rough guide bracket loosenings, were explored considering the growing problem of elevator vibration because of improvements in elevator speed and loading, which increase the interaction and wear between rollers and rails. Simulation results show that the vibration acceleration of the car initially increases and then tends toward stability with the increase in the unevenness of the roller surface. Thus, for
satisfactory dynamic performance of the elevator, the unevenness of the roller surface should be controlled within the $1 \times 10^{-3}$ order of magnitude. Car vibration acceleration and the non-roundness of the rollers show linear relationship. Therefore, car vibration owing to the excessive non-roundness of the rollers should be prevented. The instantaneous increase in car vibration caused by the loosening of several continuous rail brackets should be avoided to prevent rail-bracket loosening. These conclusions can provide certain theoretical guidance for the super-high-speed elevator to maintain good dynamic performance in the design of speed and load capacity.

Acknowledgements

The authors would like to thank the Shandong Province Natural Science Foundation, China (GRANT NO.ZR2017MEE049) and the Key Research Development Project of Shandong Province (GRANT NO. 2018GSF122004) for their financial support for this work.

References


### Tables

Table 1 Main input parameters of the simulation

<table>
<thead>
<tr>
<th>Parameter name</th>
<th>value</th>
<th>Parameter name</th>
<th>value</th>
</tr>
</thead>
<tbody>
<tr>
<td>$v$ (m·s$^{-1}$)</td>
<td>20</td>
<td>$m_c$ /kg</td>
<td>1515</td>
</tr>
<tr>
<td>$m_o$ (kg · m$^{-1}$)</td>
<td>17.8</td>
<td>$J_o$ (kg · m$^2$)</td>
<td>3667</td>
</tr>
<tr>
<td>$EI$/Nm$^2$</td>
<td>$3 \times 10^3$</td>
<td>$S$ /m</td>
<td>3</td>
</tr>
<tr>
<td>$C_o$ (Ns · m$^{-1}$)</td>
<td>1500</td>
<td>$SR2$ /m</td>
<td>$1 \times 10^{-4}$</td>
</tr>
<tr>
<td>$l$ /m</td>
<td>2.5</td>
<td>$SR3$ /m</td>
<td>$1.5 \times 10^{-4}$</td>
</tr>
<tr>
<td>$m$ /kg</td>
<td>3.1</td>
<td>$SR4$ /m</td>
<td>$2 \times 10^{-4}$</td>
</tr>
<tr>
<td>$k_1$ /Nm$^2$</td>
<td>$7 \times 10^5$</td>
<td>$r1$ /m</td>
<td>$1 \times 10^{-4}$</td>
</tr>
<tr>
<td>$C_1$ (Ns · m$^{-1}$)</td>
<td>134</td>
<td>$r2$ /m</td>
<td>$1.5 \times 10^{-4}$</td>
</tr>
<tr>
<td>$k_2$ /Nm$^2$</td>
<td>$5 \times 10^4$</td>
<td>$r3$ /m</td>
<td>$2 \times 10^{-4}$</td>
</tr>
<tr>
<td>$C_2$ (Ns · m$^{-1}$)</td>
<td>920</td>
<td>$r4$ /m</td>
<td>$2.5 \times 10^{-4}$</td>
</tr>
<tr>
<td>$SR1$ /m</td>
<td>0</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>
## List of Figures

<table>
<thead>
<tr>
<th>Figure No</th>
<th>Caption</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Horizontal vibration model of the guide rail, guide-shoe, and car coupling system</td>
</tr>
<tr>
<td>2</td>
<td>Roller Eccentricity and Roller Surface Unequal Equivalent</td>
</tr>
<tr>
<td>3</td>
<td>Rail brackets Loosening Equivalent</td>
</tr>
<tr>
<td>4(a)</td>
<td>SR=0</td>
</tr>
<tr>
<td>4(b)</td>
<td>SR=1×10^{-4}m</td>
</tr>
<tr>
<td>4(c)</td>
<td>SR=1.5×10^{-4}m</td>
</tr>
<tr>
<td>4(d)</td>
<td>SR=2×10^{-4}m</td>
</tr>
<tr>
<td>5(a)</td>
<td>r1=1×10^{-4}m</td>
</tr>
<tr>
<td>5(b)</td>
<td>r2=1.5×10^{-4}m</td>
</tr>
<tr>
<td>5(c)</td>
<td>r3=2×10^{-4}m</td>
</tr>
<tr>
<td>5(d)</td>
<td>r4=2.5×10^{-4}m</td>
</tr>
<tr>
<td>5(e)</td>
<td>A partial enlarged view of Fig. 5(d)</td>
</tr>
<tr>
<td>6(a)</td>
<td>The middle one of three consecutive rail-brackets is loose</td>
</tr>
<tr>
<td>6(b)</td>
<td>The middle two of four consecutive rail-brackets is loose</td>
</tr>
<tr>
<td>6(c)</td>
<td>The middle three of five consecutive rail-brackets is loose</td>
</tr>
</tbody>
</table>
Fig. 1 Horizontal vibration model of the guide rail, guide-shoe, and car coupling system
Fig. 2. Roller Eccentricity and Roller Surface Unequal Equivalent

205x132mm (120 x 120 DPI)
Fig. 3. Rail brackets Loosening Equivalent

169x182mm (120 x 120 DPI)
(a) The middle one of three consecutive rail-brackets is loose

(b) The middle two of four consecutive rail-brackets is loose

(c) The middle three of five consecutive rail-brackets is loose

249x235mm (120 x 120 DPI)