The Study of Novel Detection Approach for OCS Dynamic Parameters of High-speed Electrified Railway

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Abstract: This paper starts with dynamic response of pantograph slider, and computes the dynamic parameters of pantograph-catenary system such as contact force, hard spot, pull-off value and contact wire height by utilizing the transfer function matrix gained beforehand and the displacements of pantograph slider measured with the laser range sensors configured at the low voltage side. The paper also deducts the numerical algorithm of response matrix and transfer function matrix, demonstrates the feasibility of reducing the number of laser range sensors, and verifies its effectiveness by simulation with a simple example.

Key-Words: OCS(Overhead Contact System); Pantograph; System Response; Contact Force; Pull-off Value; Hard Spot; Contact Wire Height; Laser Testing Displacement

1 Introduction
Pantograph slider is moving fast under the overhead catenary when electric locomotive is running. Fig.1 shows the effect of pantograph-catenary contact force and dynamic response of horizontal vibration of pantograph slider [1-3, 4].

The study of this paper, which is helpful to construct the high-speed pantograph vibration test platform, can be verified and generalized by applying different frequencies, contact forces, and acting positions.

2 Testing principle of pantograph catenary system’s contact response

![Diagram](Image)

Fig.2 Response testing model of pantograph slider

The vibration of slider in the pantograph-catenary operation can be considered approximately as compound motion which includes horizontal bending vibration of elastic beam supported by fixed ends, vertical fluctuation and planary wheeling of rigid beam supported by elastic ends. Slider’s bending vibration mode can be solved by using Euler-Bernoulli beam [5].

$F_i$ expresses pantograph-catenary contact pulse force exerting at the $i_{th}$ spot of the slider’s beam. It
indicates changes of the pull-off value in different locations. \( Y_i \) represents testing value of displacement from the \( i_{th} \) high-speed laser sensor on the top of locomotive corresponding to the bottom of the pantograph slider. Their dynamic responses can be expressed as the following matrix form in terms of transfer function:

\[
\begin{bmatrix}
F_1 \\
F_2 \\
\vdots \\
F_n
\end{bmatrix} =
\begin{bmatrix}
M_{i1} & M_{i2} & \cdots & M_{in} \\
M_{21} & M_{22} & \cdots & M_{2n} \\
\vdots & \vdots & \ddots & \vdots \\
M_{n1} & M_{n2} & \cdots & M_{nn}
\end{bmatrix}
\begin{bmatrix}
Y_1 \\
Y_2 \\
\vdots \\
Y_n
\end{bmatrix}
\]

(1)

\( M_{ij} \) can be obtained from unit impulse response. Therefore, according to convolution principle, pantograph-catenary contact force \( P \) can be expressed as follows:

\[
P = \sum_{i=1}^{n} F_i = \sum_{j=1}^{n} \sum_{i=1}^{n} M_{ij} Y_i
\]

(2)

The impact acceleration of pantograph catenary \( G \), contact wire height \( H \) and pull-off value \( Z \) can be obtained instantly from discrete displacement signal \( y(t,i) \) tested by those laser sensors, which are expressed as follows:

\[
G = \max \left\{ \frac{d^2 y(t,i)}{dt^2} \right\} \quad i = 1, 2, \ldots, p
\]

(3)

\[
H = h_0 + \frac{1}{p} \sum_{i=1}^{p} y(t,i)
\]

(4)

\[
Z = \sum_{i=1}^{p} W_i y(t,i)
\]

(5)

Where

- \( h_0 \) the base height of sensors on the top of locomotive.
- \( p \) the number of laser sensors
- \( i \) the distributing order number of laser sensors
- \( W_i \) symmetrical weighting coefficients at geometric location of those laser sensors.

### 3 Kinetics analysis of slider’s beam

The model shown in Fig. 2 can be decomposed into a pantograph elastic slider’s beam supported by fixed ends and a pantograph rigid slider’s beam supported by elastic ends. After solving their dynamic response, horizontal response displacement \( y(t,i) \) can be added together at the same point of the axis under static equilibrium.

#### 3.1 Vibration of pantograph slider’s rigid beam in plane

Supposing that the bracing spring stiffness is \( k \), length of slider’s beam is \( l \), the line density is \( \rho \), mass is \( m \), centriod is \( c \), moment of inertia of slider’s rigid beam circling the centriod is \( I_c \), choosing centroid’s horizontal displacement \( y \) and angular displacement of slider’s rigid beam circling around the centroid as generalized coordinate \( y, \theta \), analyzing the forces exerting on the slider, differential equation of forced vibration can be established as follows:

\[
m y'' + 2ky = P_c(x-l_e) \quad (6)
\]

\[
I_c \theta'' + \frac{1}{2} k \theta' = P_c(x-l_e)(l_c - \frac{1}{2} l)
\]

(7)

Assuming \( P_c = 0 \), from which the natural frequency of horizontal vibration and the cycling frequency of rigid beam around its centroid can be obtained respectively:

\[
\omega_{n1} = \sqrt{\frac{2k}{m}} = \sqrt{\frac{2k}{\rho l}}
\]

(8)

\[
\omega_{n2} = \sqrt{\frac{k l_e^2}{2I_c}} = \sqrt{\frac{6k}{\rho l}}
\]

(9)

![Mechanic model of slider’s rigid beam imposed by external force](image)

Adopting Duhamel integral method to solve (6) and (7):

\[
y = \frac{1}{\omega_{n1}} \int_{0}^{t} P_c(x-l_e) \sin \omega_{n1}(t-\tau) d\tau
\]

\[
= \frac{P_c(x-l_e)(1 - \cos \omega_{n1}t)}{\omega_{n1}^2}
\]

(10)
\[ \theta = \frac{1}{\sigma_{x_2}} \int_0^1 P_e(x-l_c)(l_c - \frac{1}{2}t) \sin \sigma_{x_2}(t-\tau) d\tau \]
\[ P_e(x-l_c)(l_c - \frac{1}{2}t)(1-\cos \sigma_{x_2}t) \]
\[ \frac{-2}{\sigma_{x_2}} \] (11)

The composite horizontal vibration displacement \( y(x,t) \) caused by horizontal vibration and the wheeling around the centroid at \( x \) spot of slider’s rigid beam can be expressed as (12) when pantograph-catenary contact force affects on \( l_c \) spot as shown in Fig.3.
\[ y(x,t) = y - (l/2 - x) \sin \theta \]
\[ = \frac{P_e(x-l_c)(1-\cos \sigma_{x_2}t)}{\sigma_{x_2}} \]
\[ -\frac{P_e(x-l_c)(l_c - \frac{1}{2}t)(1-\cos \sigma_{x_2}t)}{\sigma_{x_2}} \]
\[ \cdots (12) \]

3.2 Bending vibration mode function of pantograph slider’s elastic beam

Considering horizontal displacement \( y \) of slider’s elastic beam supported by the fixed ends in the cross section’s symmetrical plane as generalized coordinate, supposing that line density of the beam is \( \rho \), the cross section’s bending stiffness is \( EI \), analysis of forces can be obtained as shown in Fig.4. The exerted forces on the element \( dx \) which is intercepted from section \( x \) on the beam can be analyzed. Supposing that the shearing force effected on section \( x \) is \( Q \), moment is \( M \); the shearing stress effected on section \( x + dx \) is \( Q + (dQ/dx)dx \), moment is \( M + (dM/dx)dx \). According to D’Alembert’s Principle, the following equation can be obtained:
\[ Q - \left( Q + \frac{dQ}{dx} \right) - \rho dx \frac{d^2 y}{dx^2} = 0 \] (13)

Moment equilibrium equation can be obtained by keeping arbitrary spot on section \( x + dx \) as the centroid of the moment on the right side of the unit:
\[ \left( M + \frac{dM}{dx} \right) - M = Qdx + \rho dx \frac{d^2 y}{dx^2} \]
\[ \cdots (14) \]

Ignoring microdose of the second order \( (dx)^2 \) in the above equation, the relationship of moment and shearing force can be obtained:
\[ \frac{dM}{dx} = Q \] (15)

According to the knowledge of material mechanics, moment \( M \) corresponding to generalized displacement \( y \) can be expressed as follows:
\[ M = EI \frac{d^2 y}{dx^2} \] (16)

Putting (15) and (16) into (13), a four order homogeneous PDE of beam’s horizontal vibration can be obtained:
\[ EI \frac{d^4 y}{dx^4} + \rho \frac{d^2 y}{dx^2} = 0 \] (17)

Fig. 4 Horizontal free vibration of slider’s elastic beam

The nature frequency and mode about slider’s elastic beam can be solved by using variable separation method [1, 7]. Supposing
\[ y(x,t) = \phi(x) \alpha(t) \] (18)

Differentiating the above equation and putting it into (17), the following equation is available:
\[ EI \frac{d^4 \phi(x)}{dx^4} \rho \phi(x) \frac{d^2 \phi(x)}{dx^2} = - \frac{a^2(t)}{a(t)} \]
\[ \omega_n^2 \] (19)

Because \( \omega_n^2 \) is a constant, the above equation can be expressed as follows:
\[ \frac{d^2 \phi(x)}{dx^2} - \rho \omega_n^2 \phi(x) \frac{d^2 \phi(x)}{dx^2} = 0 \]
\[ a(t) + \omega_n^2 \alpha(t) = 0 \] (20)
\[ \omega_n^2 \] is the natural frequency of slider’s beam, the solution about (21) is:
\[ a(t) = A_1 \cos \omega_n t + A_2 \sin \omega_n t \] (22)

Assuming \( \lambda = \frac{a^2(t)}{a(t)} \)
\[ a^2(t) + \omega_n^2 \alpha(t) = 0 \]
\[ \omega_n^2 \]

The solution of (20) is:
\[ \phi(x) = B_1 \sin \lambda_1 x + B_2 \cos \lambda_1 x + B_3 \sin \lambda_2 x + B_4 \cos \lambda_2 x \]
\[ A_1 \text{ and } A_2 \] in the above equation can be determined by initiative condition.

Substituting (22) and (24) into (18), the solution
about horizontal free vibration of slider’s beam is:
\[ y(x,t) = (B_1 \sin \lambda x + B_2 \cos \lambda x + B_3 \sin \lambda x + B_4 \cos \lambda x) \]
\[ \cdot \left( A_1 \cos \omega_n t + A_2 \sin \omega_n t \right) \]  
(25)

There are six constants to be determined. \( A_1 \) and \( A_2 \) relies on the initial condition of vibration. Three of the \( B_j \)s (\( j = 1,2,3,4 \)) and \( \lambda \) which is implicated in \( \omega_n \) can be determined by boundary conditions. Putting boundary conditions into the expression of vibration model function, the natural vibration model can be solved.

The expression of vibration model function can be shown as follows by using Colenov function:
\[ \phi(x) = C_1 \psi(\lambda x) + C_2 \theta(\lambda x) + C_3 \phi(\lambda x) + C_4 \phi(\lambda x) \]
Where
\[ \frac{1}{2}(C_2 - C_4) = B_1, \quad \frac{1}{2}(C_2 + C_4) = B_2, \]
\[ \frac{1}{2}(C_1 - C_3) = B_3, \quad \frac{1}{2}(C_1 + C_3) = B_4 \]
(26)

Finding the derivative of (26), the following equations are available:
\[ \frac{d\phi}{dx} = \lambda \left[ C_1 \psi(\lambda x) + C_2 \theta(\lambda x) + C_3 \phi(\lambda x) + C_4 \phi(\lambda x) \right] \]
(27)
\[ \frac{d^2 \phi}{dx^2} = \lambda^2 \left[ C_1 \psi(\lambda x) + C_2 \theta(\lambda x) + C_3 \phi(\lambda x) + C_4 \phi(\lambda x) \right] \]
(28)
\[ \frac{d^3 \phi}{dx^3} = \lambda^3 \left[ C_1 \psi(\lambda x) + C_2 \theta(\lambda x) + C_3 \phi(\lambda x) + C_4 \phi(\lambda x) \right] \]
(29)

Dynamic response of slider’s elastic beam shown in Fig.4 can be considered as horizontal bending vibration mode about elastic beam supported by fixed ends. For the elastic beam supported by fixed ends, its boundary condition is that ends’ displacement and corner of the beam are zeros. Namely,
\[ \phi\big|_{x=0} = 0, \quad \left. \frac{d\phi}{dx}\right|_{x=0} = 0 \]  
(30)
\[ \phi\big|_{x=\ell} = 0, \quad \left. \frac{d\phi}{dx}\right|_{x=\ell} = 0 \]
(31)

Putting (30) and (31) into (26) and (27), the following equations can be obtained:
\[ C_1 = C_2 = 0 \]
(32)
\[ C_2 \theta(\lambda \ell) + C_3 \phi(\lambda \ell) = 0 \]
(33)
\[ C_2 \theta(\lambda \ell) + C_3 U(\lambda \ell) = 0 \]
(34)
\[ C_3 \phi(\lambda \ell) + C_4 U(\lambda \ell) = 0 \]
\[ C_3 \text{ and } C_4 \text{ cannot be 0 so that the solution is not 0. Therefore, the following equation must be required:} \]
\[ \begin{bmatrix} U(\lambda \ell) & V(\lambda \ell) \\ T(\lambda \ell) & U(\lambda \ell) \end{bmatrix} = 0 \]
(35)

Putting Colenov function into the above equation:
\[ (c_h \lambda \ell - \cos \lambda \ell)^2 - (s_h \lambda \ell - \sin \lambda \ell)(s_h \lambda \ell + \sin \lambda \ell) = 0 \]  
... (36)

The following identical equations always exist:
\[ c_h^2 \lambda \ell - s_h^2 \lambda \ell = 1 \]
(37)
\[ \cos^2 \lambda \ell + \sin^2 \lambda \ell = 1 \]
(38)

Putting (37) and (38) into (36), frequency equation of horizontal vibration about slider’s elastic beam can be obtained:
\[ \cos \lambda \ell c_h \lambda \ell = 1 \]
(39)

The following solution can be gained from solving this hyperbolic equation:
\[ \lambda \ell \approx \frac{2i+1}{2} \pi \quad (i = 1, 2, \cdots) \]
(40)

Putting the above equation into (23), computational expressions of natural frequency about slider’s beam can be obtained:
\[ \omega_n = \frac{(2i+1)^2 \pi^2}{2 \nu \ell} \sqrt{\frac{EI}{\rho}} \quad (i = 1, 2, \cdots) \]
(41)

For the facility of calculation, parameters of vibrating slider’s beam can be chosen as Table 1.

Table 1 Calculation parameters of vibrating slider’s beam

| Line density of slider’s beam \( \rho \) | 2.5 kg/m |
| Elastic modulus of slider’s beam \( EI \) | 1720 Nm² |
| Length of slider’s beam \( \ell \) | 1.0 m |
| Elastic coefficient of springs on the ends of slider’s beam \( k \) | 2500 N/m |

The natural frequency of 1st order model is 94.5Hz, the natural frequency of 2nd order model is 258Hz, the natural frequency of 3rd order model is 505Hz, the natural frequency of 4th order model is 829Hz.

Putting (32), (33) and (34) into (26), the vibration mode function of the horizontal bending vibration of slider’s elastic beam can be obtained as (42) shows.
\[ \phi(x) = C_1 U(\lambda x) + C_2 V(\lambda x) \]
\[ = D c_h \lambda x - \cos \lambda x - \frac{s_h \lambda x + \sin \lambda x}{c_h \lambda x - \cos \lambda x} \]
(42)

In (42), \( D \) can be arbitrary constants. Main vibration mode of the corresponding order about horizontal bending vibration of slider’s elastic beam can be obtained as long as \( \lambda \ell \) corresponding to
3.3 Dynamic impulse response to pantograph slider’s elastic beam

![Image](image)

Fig. 5 Slider’s beam affected by unit impulse force

Supposing there exists a pantograph-catenary contact force \( P_r \), the moving equation of free vibration can be obtained at \( x = l_c \) spot of slider’s beam:

\[
EI \frac{d^4 y}{dx^4} + \rho \frac{d^2 y}{dt^2} = P_r \delta (x - l_c)
\]

(43) is the vibration mode function of slider’s elastic beam. Regularizing the main vibration mode, using its orthogonal features, the equation (44) is available:

\[
D^2 \left[ \frac{ch \lambda x - cos \lambda x - sh \lambda l + sin \lambda l}{ch \lambda l - cos \lambda l} (sh \lambda x - sin \lambda x) \right]^2 dx = \frac{1}{\rho} \int^x \int_y \cdots (44)
\]

Supposing that those natural frequencies are \( \omega_{nr} \), main vibration mode is \( \phi_r (x) \), where \( r = 1, 2, 3, \ldots \), the dynamic response to elastic beam can be expressed by modal superposition (coordinate transformation) as:

\[
y(x,t) = \sum_{r=1}^{\infty} \phi_r (x) q_r (t)
\]

(45) \( q_r (t) \) is the mode coordinate in the above equation. The response of each determined mode can be solved respectively; then pluses them together. Therefore, each determined mode changes into problem of single degree of freedom.

Using orthogonal feature of main vibration mode, \( r_{th} \) order mode can be expressed as follows:

\[
\frac{d^2 g_r (t)}{dt^2} + \omega_{nr}^2 g_r (t) = Q_r (t) (r = 1, 2, 3, \ldots)
\]

(46) Where,

\[
Q_r (t) = \int_0^t \phi_r (x) P_r \delta (x - l_c) dx = P_r \phi_r (l_c)
\]

It can be solved by Duhamel integral method:

\[
q_r (t) = \frac{1}{\omega_{nr}^2} \int_0^t Q_r (\tau) \sin \omega_{nr} (t - \tau) d\tau
\]

(47)

Putting (47) into (45), response to the generalized coordinates about slider’s elastic beam (48) can be obtained.

\[
y(x,t) = \sum_{r=1}^{\infty} \frac{P \phi_r (l_c)}{\omega_{nr}^2} (1 - \cos \omega_{nr} t) \phi_r (x)
\]

(48)

4 Solution of response matrix and transfer function matrix with numerical method

To solve transfer function matrix \([M_y]\) in (1), the response matrix \([D_y]\) in the following equation (49) should be solved first, which is just as the calibration process of system detection.

\[
\begin{bmatrix}
Y_1 \\
Y_2 \\
\vdots \\
Y_n
\end{bmatrix} =
\begin{bmatrix}
D_{i1} & D_{i2} & \cdots & D_{in}
\end{bmatrix}
\begin{bmatrix}
F_1 \\
F_2 \\
\vdots \\
F_n
\end{bmatrix}
\]

(49)

The relationship of transfer function matrix \([M_y]\) and response matrix \([D_y]\) is expressed by the following equation:

\[
[M_y] = [D_y]^{-1}
\]

(50)

Steps of computation based on analysis of system response are as follows:

(1) As Fig. 2 shows, suppressing a certain pantograph-catenary contact force \( F_j \), imposing on the slider at the first certain spot from left to right, the displacement response values \( Y_1, Y_2, \ldots, Y_n \) corresponding to those laser sensors can be calculated separately from (12) and (48). \( D_{ni} \) can be calculated from (51):

\[
Y_i = D_{ni} * F_j
\]

(51)

It needs to explain that the purpose on taking the maximum of the value \( Y_i \) is to avoid effects from time parameters in the function. Correspondingly, the maximum measured values of the laser sensors should be chose in unit sampling time.

(2) The method to solve the other elements \( D_{ij} \) of the matrix is similar to the way above-- supposing a certain pantograph-catenary contact force \( F_j \) imposed on the slider at the \( j_{th} \) certain spot. \( D_{ij} \) can be obtained by calculating from the following equation:
\[ Y_i = D_{ij} * F_j \]

(52)

\[ D_{ij} \] can also be calibrated and tested in lab.

(3) \[ [M_{ij}] \] can be calculated from (50).

(4) \[ F \] can be calculated from (1) and (2).

(5) Geometric parameters and dynamic parameters of catenary can be calculated from (3), (4), and (5) separately.

5 Feasibility study on reducing the number of laser range sensors

From previous discussion, it’s necessary to use laser sensors as much as possible to ensure the testing precision of dynamic parameters such as pantograph-catenary’s contact force etc. However, it’s also important to reduce the number of sensors under the precondition of keeping testing accuracy from the view of cost.

Generally speaking, dynamic characteristics of pantograph can be described through the vibration modal superposition principle. In the actual working condition of pantograph-catenary’s system, if the highest vibrating frequency is \( f_t \), then we regard the \( p_{th} \) order mode as the highest fundamental mode which is corresponding to the smallest natural frequency that is no less than \( f_t \). Then, \( p \) low-order modal independent parameters could be obtained and the problem be well solved by carrying out modal superposition. In other words, testing errors can be decreased by utilizing \( p \) laser sensors located at the sensitive displacement spots corresponding to the highest fundamental vibrating mode.

The transfer matrix elements in (1) can be expressed as follows:

\[ M_{ij} = \sum_{r=1}^{n} M_{ij,r} \]  

(53)

Where \( M_{ij,r} \) is the transfer relationship between the testing parameters sampled at the \( j_{th} \) laser sensors and mode of the \( r_{th} \) order corresponding to the \( i_{th} \) acting spot of contact force.

In case that contact force of \( i_{th} \) point can be expressed just by \( p \) sensors, \( W \) expresses the weight coefficient, \( n \geq p \), the following function is available.

\[ F_i = \sum_{j=1}^{p} M_{ij} Y_j = \sum_{k=1}^{p} W_k M_{ik} Y_k \]  

(54)

Pantograph slider and pantograph framework is an attached spring-damping system, so dynamic response of pantograph slider’s beam can be expressed as follows.

\[ Y_j = \sum_{i=1}^{p} \xi_i \phi_{ij} \]  

(55)

Where, \( \phi_{ij} \) is the \( i \) row and \( j \) column’s element of the mode matrix, \( \xi_i \) is the displacement of the \( i_{th} \) mode. Putting (55) and (53) into (54), the following function is obtained.

\[ \sum_{j=1}^{p} \sum_{i=1}^{p} \xi_i \left( \sum_{r=1}^{n} M_{ij,r} - \sum_{k=1}^{n} W_k \phi_{ik} \sum_{r=1}^{n} M_{ik,r} \right) = 0 \]  

(56)

The following function comes into existence as long as the above function is always available to directional \( \xi_i \).

\[ \sum_{j=1}^{p} \phi_{ij} \sum_{r=1}^{n} M_{ij,r} = \sum_{k=1}^{n} W_k \phi_{ik} \sum_{r=1}^{n} M_{ik,r} \]  

(57)

Where \( W_k \) is unknown variables, other parameters all can be solved from the system response relationship. So (57) is a \( p \times n \) matrix.

Therefore, dynamic response of the system can be tested accurately by \( p \) laser sensors, and from which, pantograph-catenary’s contact force can be estimated approximately.

6 System simulation of response testing

According to the process of backward analysis, configuring five laser testing displacement sensors symmetrically to test displacements of such five points as -0.4m, -0.2m, 0m, 0.2m, 0.4m at the bottom of pantograph slider, the response testing mode shown in Fig.2 can be simulated as shown in Fig.6. Where \( p = 2.5 \text{ kg/m} \), \( EI = 1720\text{Nm}^2 \), \( \ell = 0.8 \text{m} \), \( k = 2500 \text{ N/m} \).

Followed by the assumption that pantograph-catenary contact force 110N vertically imposed downward to the pantograph slider orderly at -0.4m, -0.2m, 0m, 0.2m, 0.4m, displacement response values \( Y_1, Y_2, \ldots, Y_n \) of the spots corresponding to those laser sensors can be calculated respectively from (12) and (48). Response relation matrix (58) can be obtained from (51).
Transfer function matrix (59) can be obtained from inversion of the matrix $D$.

In the case of using the pantograph-catenary contact force 150N again, imposing vertically on -0.4m and -0.2m spots downwards, from which $Y_0$ can be obtained, and then contact force 150N can be solved by putting $Y_0$ into (2) in turn. In the case of using the contact force 110N again, imposing vertically on -0.25m spots downward, from which $Y_0$ can be obtained, in turn, contact force 98.77N can be solved by putting $Y_0$ into (2). The error is...
about 10%, which is mainly created by configuring location of those sensors.

Supposing the pantograph-catenary contact force 150N imposed vertically on 0.4m spots downward, as Fig.7 (a) shows, where abcissa is detection point of slider’s beam and ordinate is contact force; displacement response to the sensors’ each spot is shown in Fig.6 (a), where abcissa is detection point of slider’s beam and ordinate is deformation of slider’s beam. Supposing the force imposed on -0.2m, 0m, 0.2m, 0.4m, the function chart of forces (Fig.7 (b) - (e)) is corresponding to displacement response chart (Fig.6 (b) - (e)).

The simulation results using transfer function computational method conform to the real situation and verify the effectiveness of this method. To distinguish from conventional mixed detection method, this approach make fully non-contact detection of OCS become reality, and therefore avoid the side effect on testing results by itself.

7 Conclusion

In practical application, the dynamic parameters such as contact force and geometric parameters such as pull-off value can be obtained through a series of calculations with the transfer function matrix and the measured data of pantograph slider’s displacements, which are sampled by the laser testing displacement sensors array installed at those vibration sensitive positions under slider’s beam.

The method of testing dynamic parameters of high-speed railway OCS based on the system response principle makes sense to take the testing sensors away completely from the pantograph slide, which is the goal of dynamic testing of high-speed railway OCS on locomotive. Owing to the limit of scan cycle and processing time, other non-contact detection such as image processing and laser radar can not meet the testing needs of dynamic characteristic under high-frequency condition. In actual application, the authors consider that data should be tested directly in the lab and disposed by recursive analysis, and computational model should be rectified and verified.

References:


[16] Chen Tanglong, Ma Fengchao, Zhou Yan, The study of OCS dynamic parameters' testing based on system response, Proceedings of the 7th WSEAS Int. Conf. on ISCGAV’07, 564-397.
