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

CHEN TANGLONG, MA FENGCHAO, ZHOU YAN
College of Electrical Engineering
Southwest Jiaotong University
North Erhuan road 111#, Chengdu, Sichuan 610031, CHINA
CHINA
tl_chen@126.com

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].

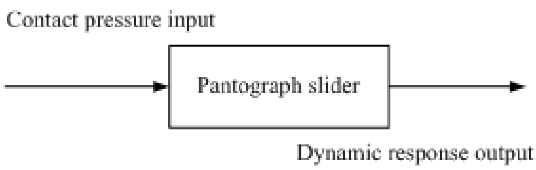

Fig.1 Analysis of system signal

Direct testing methods of pantograph-catenary contact force are commonly used in the world. However, because of fast motion, testing signals are vulnerable to the interference of electromagnetic sparks caused by pantograph-catenary contact vibration, and the installed pressure sensor increases the weight of slider and changes its shape. Therefore, the stability and safety of pantograph has been affected.

The testing method proposed in this paper is to install several laser range sensors symmetrically at the top of the locomotive, calculates the dynamic parameters of the pantograph-catenary contact force, pull-off value, hard spot and contact wire height according to testing the horizontal vibration displacement at the bottom of the pantograph slider.

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

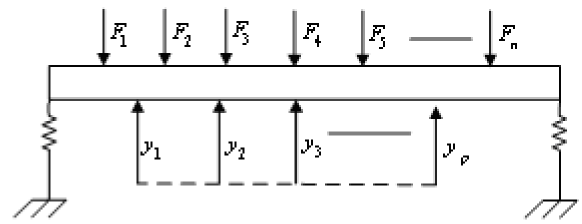

Fig.2 Response testing model of pantograph slider

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:

$$\begin{bmatrix} F_1 \\ F_2 \\ \cdots \\ F_n \end{bmatrix} = \begin{bmatrix} M_{11} & M_{12} & \cdots & M_{1n} \\ M_{21} & M_{22} & \cdots & M_{2n} \\ \cdots & \cdots & \cdots & \cdots \\ M_{n1} & M_{n2} & \cdots & M_{nn} \end{bmatrix} \begin{bmatrix} Y_1 \\ Y_2 \\ \cdots \\ Y_n \end{bmatrix} \quad (1)$$

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

$$P = \sum_{i=1}^{n} F_i = \sum_{j=1}^{n}\sum_{i=1}^{n} M_{ij} Y_i \quad (2)$$

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:

$$G = \max\{\frac{d^2 y(t,i)}{dt^2}\} \quad i = 1,\ 2,\ \cdots,\ p \quad (3)$$

$$H = h_0 + \frac{1}{p}\sum_{i=1}^{p} y(t,i) \quad (4)$$

$$Z = \sum_{i=1}^{p} W_i y(t,i) \quad (5)$$

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 ρ, 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 centriod as generalized coordinate(y,θ), analyzing the forces exerting on the slider, differential equation of forced vibration can be established as follows:

$$m\ddot{y} + 2ky = P_c(x - l_c) \quad (6)$$

$$I_c\ddot{\theta} + \frac{1}{2}kl^2\theta = P_c(x - l_c)(l_c - \frac{1}{2}l) \quad (7)$$

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:

$$\varpi_{n1} = \sqrt{\frac{2k}{m}} = \sqrt{\frac{2k}{l\rho}} \quad (8)$$

$$\varpi_{n2} = \sqrt{\frac{kl^2}{2I_c}} = \sqrt{\frac{6k}{l\rho}} \quad (9)$$

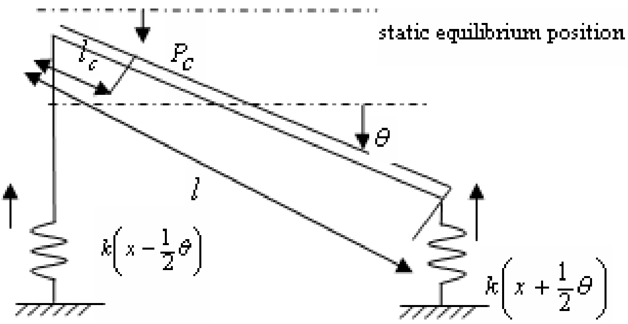

Fig.3 Mechanic model of slider's rigid beam imposed by external force

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

$$y = \frac{1}{\varpi_{n1}}\int_0^t P_c(x - l_c)\sin\varpi_{n1}(t-\tau)d\tau$$
$$= \frac{P_c(x - l_c)(1 - \cos\varpi_{n1}t)}{\varpi_{n1}^2} \quad (10)$$

$$\theta = \frac{1}{\varpi_{n2}} \int_0^t P_c(x - l_c)(l_c - \frac{1}{2}l) \sin \varpi_{n2}(t - \tau) d\tau$$
$$= \frac{P_c(x-l_c)(l_c - \frac{1}{2}l)(1 - \cos\varpi_{n2}t)}{\varpi_{n2}^2} \quad (11)$$

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.

$$y(x,t) = y - (l/2 - x)sin\theta$$
$$= \frac{P_c(x-l_c)(1-\cos\varpi_{n1}t)}{\varpi_{n1}^2}$$
$$-(l/2-x)\sin\left(\frac{P_c(x-l_c)(l_c-\frac{1}{2}l)(1-\cos\varpi_{n2}t)}{\varpi_{n2}^2}\right) \quad \ldots (12)$$

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 ρ, 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 $Q + (\partial Q/\partial x)dx$, moment is $M + (\partial M/\partial x)dx$. According to D'Alembert's Principle, the following equation can be obtained:

$$Q - \left(Q + \frac{\partial Q}{\partial x}dx\right) - \rho dx \frac{\partial^2 y}{\partial t^2} = 0 \quad (13)$$

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.

$$\left(M + \frac{\partial M}{\partial x}dx\right) - M - Qdx + \rho dx \frac{\partial^2 y}{\partial t^2}\frac{dx}{2} = 0 \quad (14)$$

Ignoring microdose of the second order $(dx)^2$ in the above equation, the relationship of moment and shearing force can be obtained:

$$\frac{\partial M}{\partial x} = Q \quad (15)$$

According to the knowledge of material mechanics, moment M corresponding to generalized displacement y can be expressed as follows:

$$M = EI\frac{\partial^2 y}{\partial x^2} \quad (16)$$

Putting (15) and (16) into (13), a four order homogeneous PDE of beam's horizontal vibration can be obtained:

$$EI\frac{\partial^4 y}{\partial x^4} + \rho\frac{\partial^2 y}{\partial t^2} = 0 \quad (17)$$

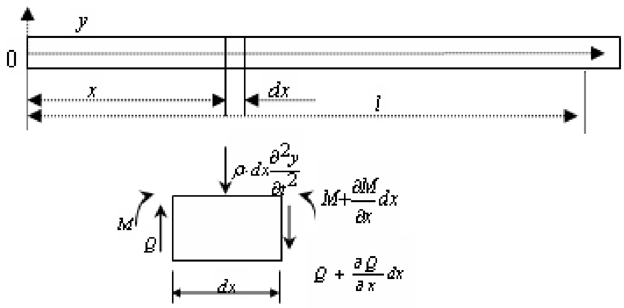

Fig.4 Horizontal free vibration of slider's elastic beam

The nature frequency and mode about slider's elastic beam can be solved by using variable separation method [1, 7]. Supposing

$$y(x,t) = \phi(x)a(t) \quad (18)$$

Differentiating the above equation and putting it into (17),the following equation is available:

$$\frac{EI}{\rho\phi(x)}\frac{d^4\phi(x)}{dx^4} = -\frac{a''(t)}{a(t)} = \omega_n^2 \quad (19)$$

Because ω_n^2 is a constant, the above equation can be expressed as follows:

$$\frac{d^4\phi(x)}{dx^4} - \frac{\rho\omega_n^2\phi(x)}{EI} = 0 \quad (20)$$

$$a''(t) + \omega_n^2 a(t) = 0 \quad (21)$$

ω_n^2 is the natural frequency of slider's beam, the solution about (21) is:

$$a(t) = A_1\cos\omega_n t + A_2\sin\omega_n t \quad (22)$$

A_1 and A_2 in the above equation can be determined by initiative condition.

Assuming $\lambda^4 = \frac{\rho}{EI}\omega_n^2$ (23)

The solution of (20) is:

$$\phi(x) = B_1\sin\lambda x + B_2\cos\lambda x + B_3 sh\lambda x + B_4 ch\lambda x \quad (24)$$

Substituting (22) and (24) into (18), the solution about horizontal free vibration of slider's beam is:

$$y(x,t) = (B_1 \sin \lambda x + B_2 \cos \lambda x + B_3 sh\lambda x + B_4 ch\lambda x) *(A_1 \cos\omega_n t + A_2 \sin \omega_n t) \quad (25)$$

There are six constants to be determined. A_1 and A_2 relies on the initiative condition of vibration. Three of the B_j s(j =1,2,3,4) and λ which is implicated in ω_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 Collenov function:

$$\phi(x) = C_1 S(\lambda x) + C_2 T(\lambda x) + C_3 U(\lambda x) + C_4 V(\lambda x) \quad (26)$$

Where $\frac{1}{2}(C_2 - C_4) = B_1, \frac{1}{2}(C_1 - C_3) = B_2,$

$\frac{1}{2}(C_2 + C_4) = B_3, \frac{1}{2}(C_1 + C_3) = B_4$

Finding the derivative of (26), the following equations are available:

$$\frac{d\phi}{dx} = \lambda[C_1 V(\lambda x) + C_2 S(\lambda x) + C_3 T(\lambda x) + C_4 U(\lambda x)] \quad (27)$$

$$\frac{d^2\phi}{dx^2} = \lambda^2[C_1 U(\lambda x) + C_2 V(\lambda x) + C_3 S(\lambda x) + C_4 T(\lambda x)] \quad (28)$$

$$\frac{d^3\phi}{dx^3} = \lambda^3[C_1 T(\lambda x) + C_2 U(\lambda x) + C_3 V(\lambda x) + C_4 S(\lambda x)] \quad (29)$$

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,

$$\phi|_{x=0} = 0, \quad \left.\frac{d\phi}{dx}\right|_{x=0} = 0 \quad (30)$$

$$\phi|_{x=l} = 0, \quad \left.\frac{d\phi}{dx}\right|_{x=l} = 0 \quad (31)$$

Putting (30) and (31) into (26) and (27), the following equations can be obtained:

$$C_1 = C_2 = 0 \quad (32)$$

$$C_3 U(\lambda l) + C_4 V(\lambda l) = 0 \quad (33)$$

$$C_3 T(\lambda l) + C_4 U(\lambda l) = 0 \quad (34)$$

C_3 and C_4 cannot be 0 so that the solution is not 0. Therefore, the following equation must be required.

$$\begin{vmatrix} U(\lambda l) & V(\lambda l) \\ T(\lambda l) & U(\lambda l) \end{vmatrix} = 0 \quad (35)$$

Putting Collenov function into the above equation:

$$(ch\lambda l - \cos \lambda l)^2 - (sh\lambda l - \sin \lambda l)(sh\lambda l + \sin \lambda l) = 0 \quad \ldots (36)$$

The following identical equations always exist:

$$ch^2 \lambda l - sh^2 \lambda l = 1 \quad (37)$$

$$\cos^2 \lambda l + \sin^2 \lambda l = 1 \quad (38)$$

Putting (37) and (38) into (36), frequency equation of horizontal vibration about slider's elastic beam can be obtained:

$$\cos\lambda l ch\lambda l = 1 \quad (39)$$

The following solution can be gained from solving this hyperbolic equation:

$$\lambda_i l \approx \frac{2i+1}{2}\pi \quad (i = 1,2,\cdots) \quad (40)$$

Putting the above equation into (23), computational expressions of natural frequency about slider's beam can be obtained:

$$\omega_{ni} = \left(\frac{2i+1}{2l}\pi\right)^2 \sqrt{\frac{EI}{\rho}} \quad (i = 1,2,\cdots) \quad (41)$$

For the facility of calculation, parameters of vibrating slider's beam can be chosen as Table1.

Table1 Calculation parameters of vibrating slider's beam

Line density of slider's beam ρ	2.5kg / m
Elastic modulus of slider's beam EI	1720 Nm^2
Length of slider's beam ℓ	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.

$$\phi(x) = C_3 U(\lambda x) + C_4 V(\lambda x) = D\left[ch\lambda x - \cos\lambda x - \frac{sh\lambda l + \sin\lambda l}{ch\lambda l - \cos\lambda l}(sh\lambda x - \sin\lambda x)\right] \quad \ldots (42)$$

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 $\lambda_i l$ corresponding to those natural frequencies is put into (42).

3.3 Dynamic impulse response to pantograph slider's elastic beam

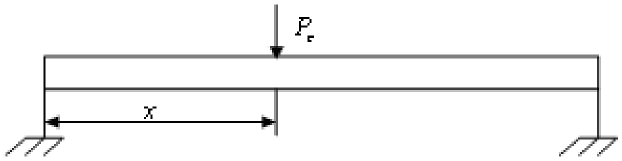

Fig.5 Slider's beam affected by unit impulse force

Supposing there exists a pantograph-catenary contact force P_c, the moving equation of free vibration can be obtained at $x = l_c$ spot of slider's beam:

$$EI\frac{\partial^4 y}{\partial x^4} + \rho\frac{\partial^2 y}{\partial t^2} = P_c\delta(x - l_c) \tag{43}$$

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

$$D^2\int_0^l\left[ch\lambda x - \cos\lambda x - \frac{sh\lambda l + \sin\lambda l}{ch\lambda l - \cos\lambda l}(sh\lambda x - \sin\lambda x)\right]^2 dx = \frac{1}{\rho} \quad \ldots (44)$$

Supposing that those natural frequencies are ω_{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:

$$y(x,t) = \sum_{r=1}^{\infty}\phi_r(x)q_r(t) \tag{45}$$

$q_r(t)$ 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:

$$\frac{d^2q_r(t)}{dt^2} + \omega_{nr}q_r(t) = Q_r(t)(r = 1,2,3,\cdots) \tag{46}$$

Where,

$$Q_r(t) = \int_0^l \phi_r(x)P_c\delta(x - l_c)dx = P_c\phi_r(l_c)$$

It can be solved by Duhamel integral method:

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

$$= \frac{P_c\phi_r(l_c)}{\omega^2{}_{nr}}(1 - \cos\omega_{nr}t) \tag{47}$$

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

$$y(x,t) = \sum_{r=1}^{\infty}\left[\frac{P_c\phi_r(l_c)}{\omega^2{}_{nr}}(1 - \cos\omega_{nr}t)\phi_r(x)\right] \tag{48}$$

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

To solve transfer function matrix [M_{ij}] in (1), the response matrix [D_{ij}] in the following equation (49) should be solved first, which is just as the calibration process of system detection.

$$\begin{bmatrix} Y_1 \\ Y_2 \\ \cdots \\ Y_n \end{bmatrix} = \begin{bmatrix} D_{11} & D_{12} & \cdots & D_{1n} \\ D_{21} & D_{22} & \cdots & D_{2n} \\ \cdots & \cdots & \cdots & \cdots \\ D_{n1} & D_{n2} & \cdots & D_{nn} \end{bmatrix}\begin{bmatrix} F_1 \\ F_2 \\ \cdots \\ F_n \end{bmatrix} \tag{49}$$

The relationship of transfer function matrix [M_{ij}] and response matrix [D_{ij}] is expressed by the following equation:

$$[M_{ij}] = [D_{ij}]^{-1} \tag{50}$$

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

(1) As Fig.2 shows, supposing a certain pantograph-catenary contact force F_1, imposing on the slider at the first certain spot from left to right, the displacement response values Y_1 , Y_2 ,…, Y_n corresponding to those laser sensors can be calculated separately from (12) and (48). D_{i1} can be calculated from (51):

$$Y_i = D_{i1} * F_1 \tag{51}$$

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:

$$Y_i = D_{ij} * F_j \tag{52}$$

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, 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. 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:

$$M_{ij} = \sum_{r=1}^{p} M'_{ij,r} \tag{53}$$

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.

$$F_i = \sum_{j=1}^{n} M_{ij} Y_j = \sum_{k=1}^{p} W_{ik} M_{ik} Y_k \tag{54}$$

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

$$Y_j = \sum_{l=1}^{p} \xi_l \phi_{lj} \tag{55}$$

Where, ϕ_{ij} is the i row and j column's element of the mode matrix, ξ_l is the displacement of the l_{th} mode. Putting (55) and (53) into (54), the following function is obtained.

$$\sum_{l=1}^{p} \xi_l \left(\sum_{j=1}^{n} \phi_{lj} \sum_{r=1}^{p} M'_{ij,r} - \sum_{k=1}^{p} W_{lk} \phi_{lk} \sum_{r=1}^{p} M'_{ij,r} \right) = 0 \tag{56}$$

The following function comes into existence as long as the above function is always available to discretional ξ_l.

$$\sum_{j=1}^{n} \phi_{ij} \sum_{r=1}^{p} M'_{ij,r} = \sum_{k=1}^{p} W_{ik} \phi_{ik} \sum_{r=1}^{p} M'_{ij,r} \tag{57}$$

Where W_{ik} 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 ρ is $2.5\,\mathrm{kg/m}$, EI is 1720Nm^2, ℓ is 0.8m, k is 2500 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, …, 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).

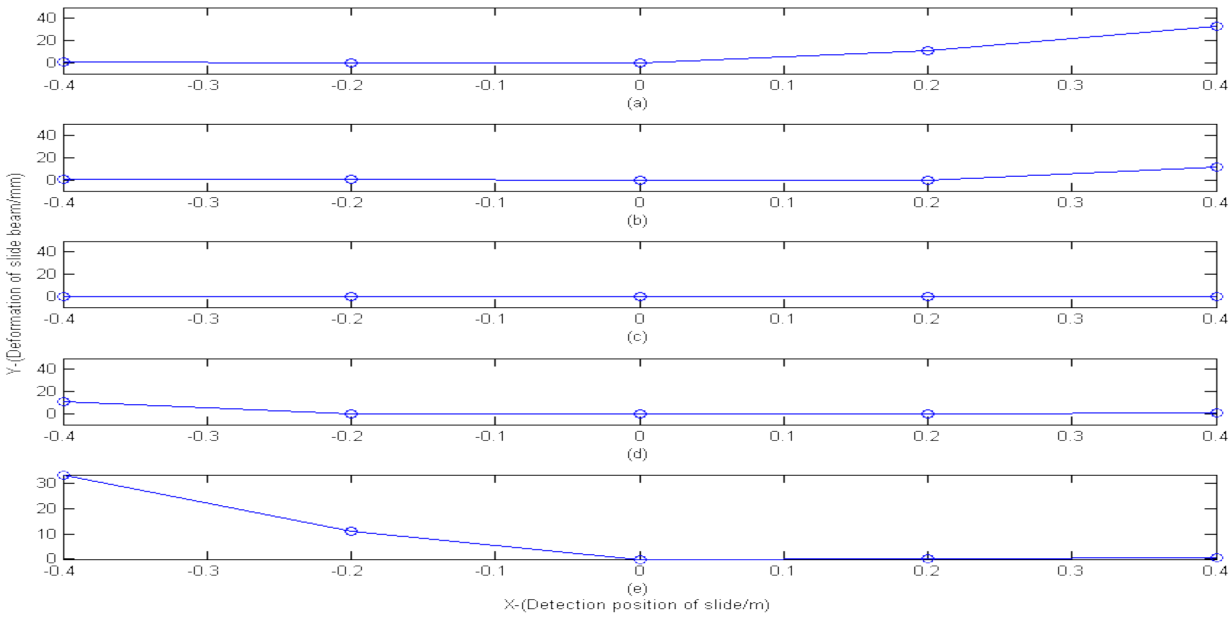

Fig.6 Slider beam's deformation when apply 150N contact force on the pantograph

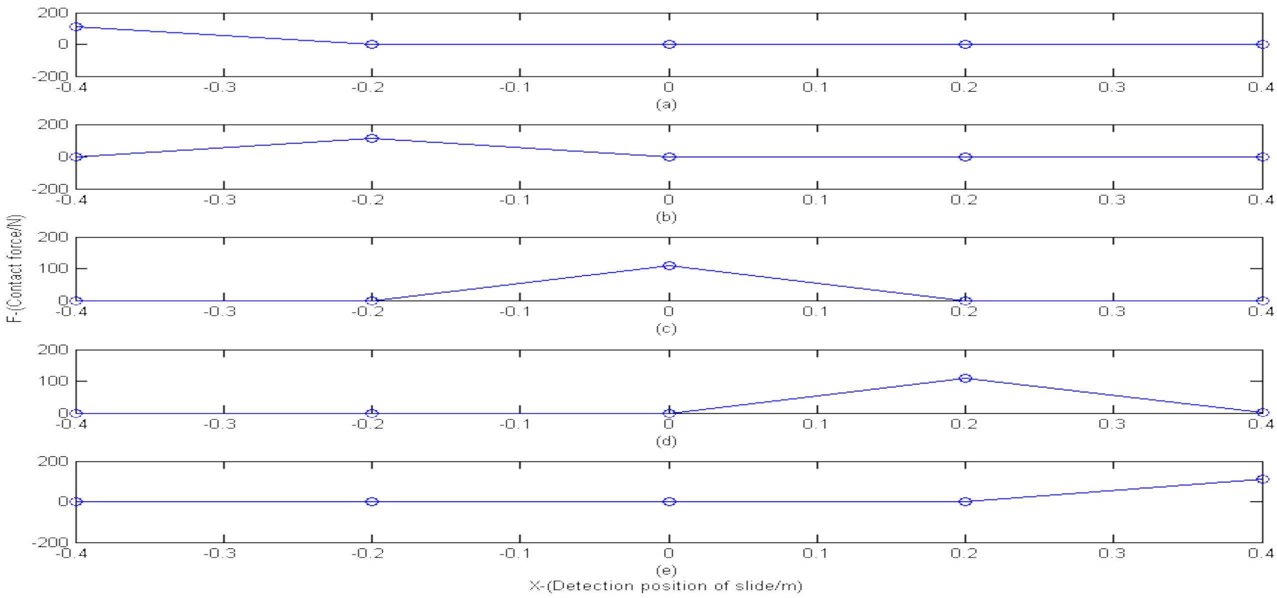

Fig.7 Contact forces of each corresponding points when apply 150N contact force on the pantograph

$$D=\begin{bmatrix} 3.7871e\text{-}006 & 2.5405e\text{-}006 & -8.9403e\text{-}008 & -3.9702e\text{-}007 & -6.394e\text{-}007 \\ 1.0888e\text{-}006 & 8.2842e\text{-}007 & -6.9536e\text{-}008 & -2.6592e\text{-}007 & -3.9702e\text{-}007 \\ -9.9337e\text{-}009 & -2.9801e\text{-}008 & -4.9668e\text{-}008 & -6.9536e\text{-}008 & -8.9403e\text{-}008 \\ 0.00010028 & -2.9547e\text{-}008 & -2.9801e\text{-}008 & 8.2842e\text{-}007 & 2.5405e\text{-}006 \\ 0.00030204 & 0.00010028 & -9.9337e\text{-}009 & 1.0888e\text{-}006 & 3.7871e\text{-}006 \end{bmatrix} \quad \ldots (58)$$

$$M=\begin{bmatrix} 2.6088e{+}005 & -3.6746e{+}005 & 36664 & 14829 & -3558.2 \\ -3.4185e{+}005 & 4.587e{+}005 & -7765.2 & -36735 & 14829 \\ -9.8061e{+}006 & 2.4319e{+}007 & -3.6532e{+}007 & -7765 & 36664 \\ 3.5312e{+}007 & -6.2916e{+}007 & 2.4319e{+}007 & 4.587e{+}005 & -3.6746e{+}005 \\ -2.1932e{+}007 & 3.5312e{+}007 & -9.8061e{+}006 & -3.4185e{+}005 & 2.6088e{+}005 \end{bmatrix} \quad \ldots (59)$$

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 abscissa 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 abscissa 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. Owning 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:

[1] Yu WanJu, Catenary system of high-speed electrified railway, Southwest Jiaotong University Press, 2003.

[2] Gukow, Kiessling Puschmann, Schmider, Schmidt, Fahrleitungen elektrtrischer, Banhnen.B.G.Teubner Stuttgart, 1997.

[3] Zhang WeiHua, The study of dynamic response about quais-high-speed railway catenary, Southwest Jiaotong University Press, 1997.

[4] Fujii Yasukazu, Theory on providing current for high-speed railway catenary, Railway and electrical technology, 1991.

[5] Xia YongYuan, Zhang Azhou, Calculation method of mechanical vibration, national defense industry press, 1993.

[6] Li GuoHao, Stability and vibration of bridge, China Railway publishing house, 2003.

[7] Xi ShaoZhong, Qiu BingQuan, Engineering mechanics, Southwest Jiaotong University Press, 2005.

[8] Mitsuru LKEDA, The conact force between pantograph and contact wire, QR of RTRI, Vol.45, No.2, May, 2004.

[9] Zhang Ce, Mechanical dynamics, Higher Education Press, 2000.

[10] Zhang Xianda, Modern signal processing, Tsinghua University Press, 2002.

[11] Chen AnHua, Liu DeShun, Dynamics theory and methods of vibration diagnosis, Machinery Industry Press, 2002.

[12] Zhang Jianmin, Mechanical vibration, China University of Geosciences Press, 1995.

[13] Chen Tanglong, Xiao Jian, Synthetical analysis of OCS dynamic testing data, WSEAS TRANSACTIONS on SYSTEM Issue 5, Volume 5, 2006, 705-710.

[14] Chen Tanglong, Xiao Jian, The OCS testing data analysis of hard spot based on data-mining technique, 6th Industrical conference on Data Mining ICDM'2006 Juny 14-15, 2006, Leipzig/Germany.

[15] Chen Tanglong, Xiao Jian, Novel detection methods for overhead contact system zigzag based on neural networks, Proceedings of the 9th WSEAS Int. Conf. on SYSTEM(CSCC 2005), 497-776.

[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.

[17] Chen Tanglong, Xiao Jian, Sysmatic analysis of OCS testing data, Proceedings of the WSEAS Int. Conf. On IMCAS'06, 531-593.

[18] Chen Tanglong, Xiao Jian, The study novel detection method for overhead contact system zigzag of high-speed electrified railway, WSEAS TRANSACTIONS on SYSTEM Issue 5, Volume 4, 2005, 529-536.