The Study of OCS Dynamic Parameters' Testing based on System Response

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Abstract: This paper starts with dynamic response of pantograph slider, and computes the dynamic parameters of pantograph-catenary systems 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, and verifies its effectiveness by simulation with a simple example.

Key-Words: OCS; 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].

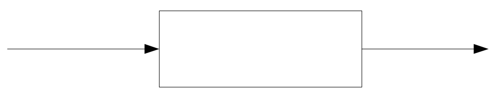

Fig.1 Analysis of system signal

Direct testing methods of pantograph-catenary contact force are commonly used domestically and abroad. 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.

2 Testing principle of pantograph catenary system's contact response

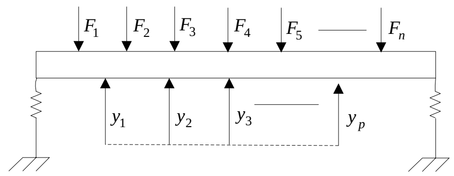

Fig.2 Response testing model of pantograph slider

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 and fluctuation drive and planar wheel 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 slide 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 the bottom of the pantograph slider. Their dynamic response 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:

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

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:

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

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

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

The parameters in (3), (4), (5) are as follows, where:

h_0 the base height of sensors on the top of locomotive.

p the number of laser sensors

i the distributing ordinal number of laser sensors

W_i symmetrical weighting coefficients at geometric location about 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 fix~~ed~~ 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 circled the centriod as generalized coordinate y, θ, analyzing the forces exerting on the slider, Differential equation of forced vibration can be established as follows:

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

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

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:

$$\varpi_{n1} = \sqrt{\frac{2k}{m}} = \sqrt{\frac{2k}{l\rho}} \qquad 8$$

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

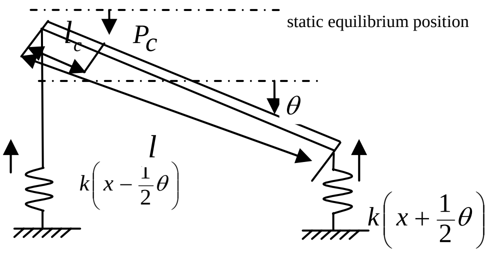

Fig.3 Mechanic model of slider's rigid beam imposed by external force

Adopting Duhamel integral method to solve (6) and (7), 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 (10) when pantograph-catenary contact force affects on l_c spot as Fig.3 shows.

$$y(x,t) = \frac{P_c(x - l_c)(1 - \cos\varpi_{n1}t)}{\varpi_{n1}^2} - (l/2 - x)\sin(\frac{P_c(x - l_c)(l_c - \frac{1}{2}l)(1 - \cos\varpi_{n2}t)}{\varpi_{n2}^2}) \qquad 10$$

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 symmetry plane as generalized coordinate, supposing that line density of the beam is ρ, bending stiffness of it is *EI*, analysis of forces can be obtained as shown in Fig.4. Four order homogeneous PDE of slider's beam's horizontal vibration can be obtained based on D'Alembert's Principle and Torque Equilibrium Principle:

$$EI\frac{\partial^4 y}{\partial x^4} + \rho\frac{\partial^2 y}{\partial t^2} = 0 \qquad (11)$$

Formula of natural frequency (12) and horizontal bending vibration mode function of slider's beam (13) can be obtained by using separation of variables and Collenov function to (11):

$$\omega_{ni} = \left(\frac{2i+1}{2l}\pi\right)^2 \sqrt{\frac{EI}{\rho}} \qquad (i = 1,2,\cdots) \qquad 12$$

$$\phi(x) = D\left[ch\lambda x - \cos\lambda x - \frac{sh\lambda l + \sin\lambda l}{ch\lambda l - \cos\lambda l}(sh\lambda x - \sin\lambda x)\right] \quad \dots \ 13)$$

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	$2.5\,\mathrm{kg/m}$
Elastic modulus of slider's beam EI	$1720\ \mathrm{Nm}^2$
Length of slider's beam ℓ	1.0 m
Elastic coefficient of springs on the ends of slider's beam k	2500 N/m

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

In (13), 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 (13).

3.3 Dynamic impulse response to pantograph slider's elastic beam

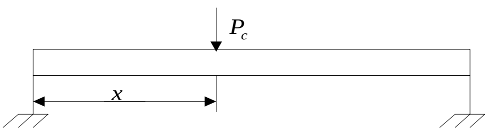

Fig.5 Slider's beam affected by unit impulse force

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:

$$EI\frac{\partial^4 y}{\partial x^4} + \rho\frac{\partial^2 y}{\partial t^2} = P_c\delta(x - l_c) \qquad 14$$

(13) is the vibration mode function of slider's elastic beam. Regularizing the main vibration mode, using its orthogonal features, the equation (15) is available.

$$D^2\int_0^l\left[ch\lambda x - \cos\lambda x - \frac{sh\lambda l + \sin\lambda l}{ch\lambda l - \cos\lambda l}(sh\lambda x - \sin\lambda x)\right]^2 dx = \frac{1}{\rho} \quad \dots \ 15$$

Supposing that those natural frequencies is 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:

$$y(x,t) = \sum_{r=1}^{\infty}\phi_r(x)q_r(t) \qquad (16)$$

Using orthogonal feature of main vibration mode, it can be solved by Duhamel integral method:

$$q_r(t) = \frac{P_c\phi_r(l_c)}{\omega^2_{nr}}(1\text{-}\cos\omega_{nr}t) \qquad 17$$

Putting (17) into (16), response to the generalized coordinates about slider's elastic beam (18) can be obtained.

$$y(x,t) = \sum_{r=1}^{\infty}\left[\frac{P_c\phi_r(l_c)}{\omega^2_{nr}}(1 - \cos\omega_{nr}t)\phi_r(x)\right] \qquad 18$$

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 (19) should be solved first:

$$\begin{bmatrix} Y_1 \\ Y_2 \\ \cdots \\ Y_n \end{bmatrix} = \begin{bmatrix} D_{11} & D_{12} & \cdots & D_{1n} \\ D_{21} & D_{22} & \cdots & D_{2n} \\ \cdots & \cdots & \cdots & \cdots \\ D_{n1} & D_{n2} & \cdots & D_{nn} \end{bmatrix}\begin{bmatrix} F_1 \\ F_2 \\ \cdots \\ F_n \end{bmatrix} \qquad 19$$

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

$$[M_{ij}] = [D_{ij}]^{-1} \qquad 20$$

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 (10) and (18). D_{i1} can be calculated from (21):

$$Y_i = D_{i1} * F_1 \qquad 21$$

(2)The method to solve the other elements D_{ij} of the matrix is similar to the way above. D_{ij} can be obtained by calculating from the following equation:

$$Y_i = D_{ij} * F_j \qquad 22$$

(3) [M_{ij}] can be calculated from (20).

(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 System simulation of response testing

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 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 (10) and (18). Response relation matrix (23) can be obtained from (21).

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

… 23

Transfer function matrix (24) can be obtained from inversion of the above matrix D.

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

… 24

In the case of using the pantograph-catenery 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 10%, which is mainly created by configuring location of those sensors.

Supposing the pantograph-catenery contact force 150N imposed vertically on -0.4m spots downward, as Fig.7 (a) shows, displacement response to the sensors' each spot is shown in Fig.6 (a). 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)).

This shows that simulation using transfer function computational method conforms to the real situation.

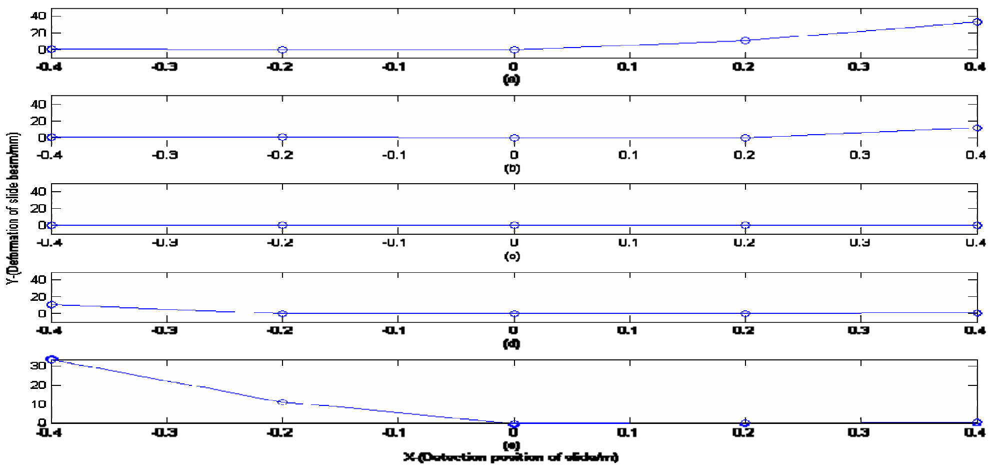

Fig.6 Slider beam's deformation when apply 150N contact force on the pantograph

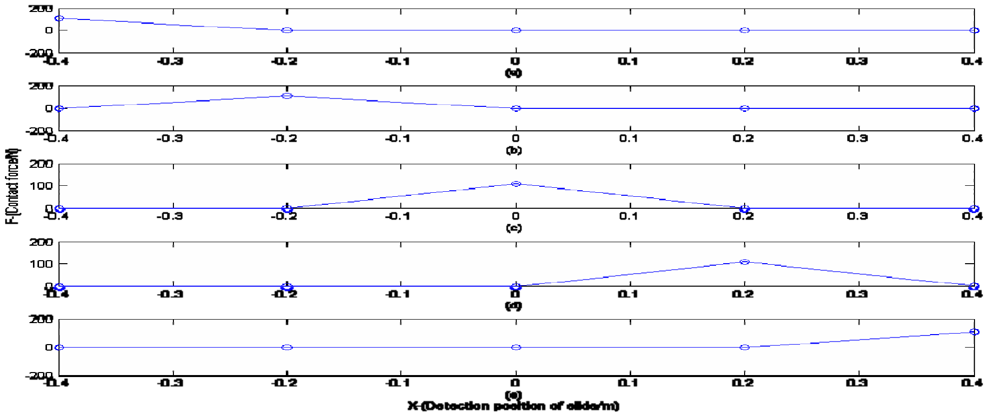

Fig.7 Contact forces of each corresponding points when apply 150N contact force on the pantograph

6 Conclusion

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 practical 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.

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