Research of the Active Control for High-Speed Train Pantograph

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Abstract—**One of the main problems which limit the trains run at a higher speed is the vibration of the contact force between the pantograph and the catenary. In this paper, three kinds of stiffness curves are fitted by using the non-linear least square method and a unified fitting formula is presented. Based on this, a 3-DOF time-variant model of the pantograph / catenary is established, with optimal control strategy , the contact force under different train speeds is controlled, the simulations show that this method can reduce the fluctuations of the contact force, the controller is *effective*.**

Keywords—**Pantograph, Active, control, Catenary, Simulation**

I. INTRODUCTION`

The vibration between pantograph collector head and catenary causes the vehicle's current collection quality gets badly, damages the locomotive, accelerate wear of catenary, makes an enormous economic losses in particular. So the study on how to reduce pantograph vibration becomes more and more important.

A common request is that of having an almost constant contact force in all operating condition, which is the goal that we pursue, but it is not easy to realize. The counteraction of the variations of the contact force magnitude could be obtained either by increasing the tension of the contact wire or by increasing contact force modulus, actually the first solution is quite expensive, and it would need that connected railway systems have the same standards, while the latter implies the fast wearing of the equipment[1][5].

An interesting approach to the problem is that of using active pantographs, in which either the lower or the upper frame of the arm supporting the current collector is actuated. In [1] a linear time-variant model for the pantograph dynamics is established, and the catenary is considered as environmental disturbance modeled by an uncertain spring with time-varying stiffness. The contact force is the measurable system output to regulate along a prescribed constant value, which only the lower frame is actuated. In [2] the dynamics pantograph/catenary system are examined. Active pantograph elements are introduced to reduce contact force variation, the number of states measured is reduced, velocity between the frame and the head, without significantly affecting system performance. But the controller is not suitable at a higher speed. In [3] the control strategy is based on Extended Kalman Filter technique, used to get a contact force estimation available for control feedback. In [10], wu xuejie etc. adopt the fuzzy control technology and study active control of the pantograph SS 7, and has carried on the model test.

In this paper, three kinds of stiffness curves are fitted by using the non-linear least square method and a unified fitting formula is presented. Based on this, a 3-DOF time-variant model of the pantograph / catenary system is established, with optimal control strategy, the contact force under different train speeds is controlled.

II.CATENARY MODELS AND STIFFNESS FORMULA

In order to research active pantograph, the equivalent stiffness of catenary system must be handled well. High speed vehicle catenary structure has been sort out three types, i.e., the simple chain suspension, the elastic chain suspension and double chains suspension, the parameters of the three types catenarys is shown in table 1, and the finite element model is shown in Fig. 1.

Every type suspension has difference stiffness, in fact, the stiffness changes, ranging from high values in correspondence of the towers to low values in the center of a span, and the velocity of such variation depend on the vehicle speed. In [1][6][8], catenary stiffness is regarded as a time-varying stiffness spring system, which is simplified as,

$$k(t) = k_0(1 + \varepsilon \cos(\frac{2\pi v}{L}t) \qquad (1)$$

where

v - speed of vehicl, m/s.

L – span length, m.

k_0 - average equivalent stiffness, N/m.

ε *-- stiffness variation coefficient in a span.*

$$\varepsilon = \frac{k_{\max} - k_{\min}}{k_{\max} + k_{\min}}, \quad k_0 = \frac{k_{\max} + k_{\min}}{2}$$

Kmax, kmin—the maximum, minimum stiffness value in a span，N/m。

Table 1. Characteristics of the Baseline catenary system

TERMS	SIMPLE CHAIN	ELASTIC CHAIN	DOUBLE CHAIN
Tower spacing	63 m	65mm	50mm
Messenger wire tension	14 KN	15KN	24KN
Contact wire tension	20KN	15KN	14.7KN
auxiliary messenger wire tensio		2.8KN	14.7KN
Contact wire density	2.0kg/m	1.8kg/m	4.4kg/m
Contact wire area	150mm2	120mm2	170mm2
Messenger wire area	65 mm2	70mm2	180mm2

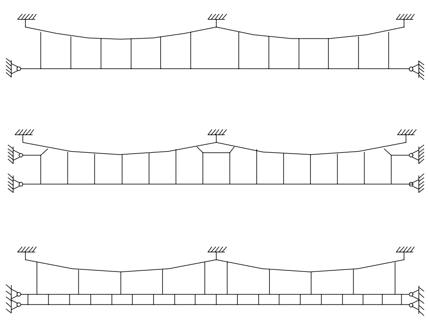


Fig. 1 The FEM models of the Catenary

Formula (1) is only very approximate description of the catenary stiffness, using it to control vibration of pantograph-catenary system would in sequel lead to further error simulation, and it can not reflect the actual operating model correctly. [8][11] give the measure result of the simple chain suspension and the elastic chain suspension. [9] presents the calculating result of three types catenary stiffness using finite element method(FEM), the result is the same with [8][11]. In order to use the FEM result much better, and simplify calculating, it is necessary to formularize the FEM result.

In this paper, three types of stiffness curves are fitted by using the non-linear least square method, and a unified fitting formula of catenary stiffness is presented. The fitting result as follow

$$k(t) = k_0(1 + \alpha_1 f_1 + \alpha_2 f_2 + \alpha_3 f_1^2 + \alpha_4 f_3^2 + \alpha_5 f_4^2) \quad (2)$$

where $f_1 = \cos(\frac{2\pi v}{L}t)$, $f_2 = \cos(\frac{2\pi v}{L_1}t)$,

$f_3 = \cos(\frac{\pi v}{L}t)$, $f_4 = \cos(\frac{\pi v}{L_1}t)$

L1—dropper spacing， m

other sign is homologous with (1).

For the simple chain catenary, the coefficient in the formula is computed as

K0=3694.5，κμαξ=5326, κμιν=2063, ε=44%,

α_1 = 0.4665，α_2 = 0.0832， α_3 = 0.2603， α_4 = –0.2801， α_5 = –0.3364

For the elastic chain catenary, the coefficient in the formula is computed as

K0=1925，κμαξ=2100, κμιν=1550, ε=15%,

α_1 = 0.0755，α_2 = –0.0735，

α_3 = –0.1459， α_4 = –0.0575， α_5 = 0.0699

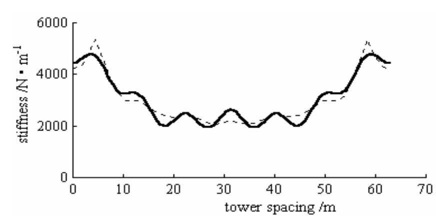
For the double chain catenary, the coefficient in the formula is computed as

K0=5017，κμαξ=6364, κμιν=3670, ε=27%,

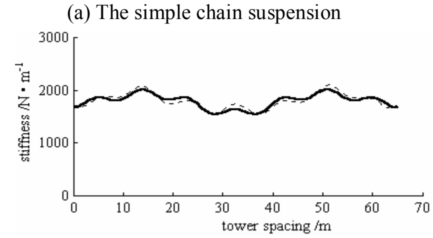
α_1 = 0.2124，α_2 = –0.0588， α_3 = 0.0674， α_4 = –0.1168， α_5 = –0.0937

Fitting curves and FEM result are compared in Fig. 2, the dotted lines denote the FEM results, and the full lines denote the fitting result. Analyzing the fig. 2, we can find that fitting result is very close to the FEM results, and its fitting precision can be even satisfied requests of calculation or design.

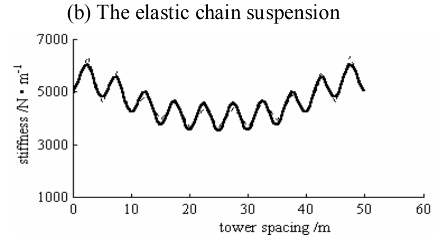
The computation speed using least square fitting model is higher than using FEM model, the computation time is reduced.



(a) The simple chain suspension



(b) The elastic chain suspension



（c）The double chain suspension

Fig. 2 Stiffness curve of the catenary

- III.PANTOGRAPH MODEL

THE MATHEMATICS MODEL OF PANTOGRAPH SYSTEM IS VERY COMPLICATED. IN [7], IT SHOWS THAT THE PANTOGRAPH SYSTEM IS A NON-LINEAR SYSTEM CONTAINING square TERMS. BEING COMPLICATED AND DIFFICULT TO SOLVE, SO IT IS USUALLY LINEARIZED, AND USE 2 DEGREE OF FREEDOM(DOF) OR 3DOF MODEL, AS FIG. 2 SHOWS.

The objective of a pantograph control law is to minimize fluctuations in contact force from the desired value. This will be done here with a controller which applies its input force to the frame. This frame-actuated design was chosen for several reasons. First it would be easy installed, many pantographs already use a pneumatic cylinder on the frame to provide uplift force. such a cylinder could be chosen to be compatible with pneumatic servo system hardware. Such hardware is readily available from an array of commercial vendors. A head-actuated design would require the actuator to move with the pantograph frame. Increasing the overall mass and thus degrading performance[2].

Analyzing the pantograph model (fig. 3 (b)), the dynamics equation of the pantograph-catenary system can be written as follow :

$$\begin{cases} m_1\ddot{z}_1 + k_1(z_1 - z_2) + c_1(\dot{z}_1 - \dot{z}_2) + k(t)z_1 = 0 \\ m_2\ddot{z}_2 + k_1(z_2 - z_1) + k_2(z_2 - z_3) + c_1(\dot{z}_2 - \dot{z}_1) \\ + c_2(\dot{z}_2 - \dot{z}_3) = 0 \\ m_3\ddot{z}_3 + k_2(z_3 - z_2) + c_2(\dot{z}_3 - \dot{z}_2) + c_3(\dot{z}_3 - \dot{z}_r) = F \end{cases} \quad (3)$$

where

$\dot{z}_r$ —the interfering signal from locomotive.

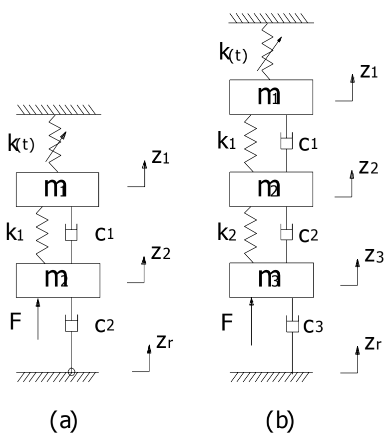


Fig. 3 The pantograph reference model

F — active control force, N.

Equation (3) can't be used to calculate directly by active control method. For analyzing conveniently, vary equation (3) and make the state variables as follow

$$X = [x_1, x_2, x_3, x_4, x_5, x_6]^{\mathrm{T}} = [z_1, \dot{z}_1, z_2, \dot{z}_2, z_3, \dot{z}_3]^{\mathrm{T}}$$

so the equation (3) will be written as

$$\dot{X} = AX + Bu + DP$$

$$Y = k(t)x_1 \quad (4)$$

where

$$A = \begin{bmatrix} 0 & 1 & 0 & 0 & 0 & 0 \\ -\frac{k_1 + k(t)}{m_1} & -\frac{c_1}{m_1} & \frac{k_1}{m_1} & \frac{c_1}{m_1} & 0 & 0 \\ 0 & 0 & 0 & 1 & 0 & 0 \\ \frac{k_1}{m_2} & \frac{c_1}{m_2} & -\frac{k_1 + k_2}{m_2} & -\frac{c_1 + c_2}{m_2} & \frac{k_2}{m_2} & \frac{c_2}{m_2} \\ 0 & 0 & 0 & 0 & 0 & 1 \\ 0 & 0 & \frac{k_2}{m_3} & \frac{c_2}{m_3} & -\frac{k_2}{m_3} & -\frac{c_3 + c_2}{m_3} \end{bmatrix}$$

$$B = \begin{bmatrix} 0 \\ 0 \\ 0 \\ 0 \\ 0 \\ \frac{1}{m_3} \end{bmatrix}, \quad D = \begin{bmatrix} 0 \\ 0 \\ 0 \\ 0 \\ 0 \\ \frac{c_3}{m_3} \end{bmatrix}, \quad P = \dot{z}_r = w(t).$$

u—input signal.

$W(t)$—white noise.

Equation (4) is established as the active control model of the pantograph in this paper. In the following, it will be used for simulation.

IV.ACTIVE CONTROL STRATEGY

While the locomotive runs, the pantograph gets the interfering force from the locomotive can be regarded as an integrated white noise:

$$\dot{z}_r = w(t),$$

$$E[w(t)w(t-t)] = \phi_0\delta(t)$$

that is a gauss white noise signals with zero mean value.

In order to improve the pantograph performance, reduce the contact force oscillation, and improve the current collection quality, the pantograph head vibration should be reduced during designing active pantograph. In addition, from the view point of realizing control, should need the necessary control energy relatively little. So the pantograph – catenary system performance index can be written as

$$J(u) = E\left[\int_0^\infty (Ru^2(t) + X^T QX)dt\right] \quad (5)$$

where

R-- the weight coefficient.

Q -- the right matrix of the state.

Suppose that all of the state vector can be measured, the system (4) is satisfied equation (5), so the optimizing controlling ratio is expressed as

$$u = -Kx$$
$$K = R^{-1}B^T P^T \quad (6)$$

where

$$K = [k_1^* \;\; k_2^* \;\; k_3^* \;\; k_4^*],$$

P can be solved from the riccati's equation of following.

$$A^T P + PA - PBR^{-1}B^T P + Q = 0 \quad (7)$$

the active control force as following

$$F=-R^{-1}B^T P^T + F_0 \quad (8)$$

where F_0 is the desired contact force, N.

V.SIMULATION

The state weighting matrix Q and the control weighting fator R are

Q=διαγ(0.15,0.04,0.15,0.15,0.04,0.15), P =10–6.

Table 2 Baseline pantograph parameters

Head mass	m1	6.4 kg
Upside frame mass	m2	7.0 kg
Downside frame mass	m3	12.0 kg
Stiffness between the head and upside frame	k1	2650 N/m
Stiffness between the upside and downside frame	k_2	10000 N/m
Stiffness between the downside frame and base	k_3	0
Damping between the head and upside frame	c_1	0
Damping between the upside and downside frame	c_2	0
Damping between the downside frame and base	c3	70 Ns/m
Static uplift force	F0	90 N

In this paper only the simple chain catenary and the elastic chain catenary are simulated.

A Simulation under the Simple Chain Catenary

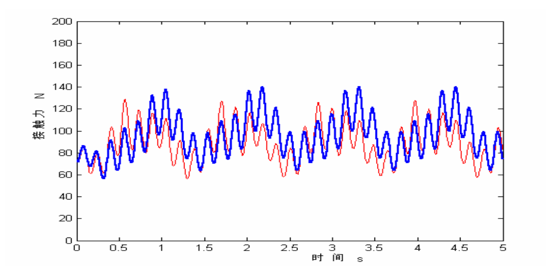
The values of the gains are

κ1*=−46 N/μ，κ2*=−57 Nσ/μ，

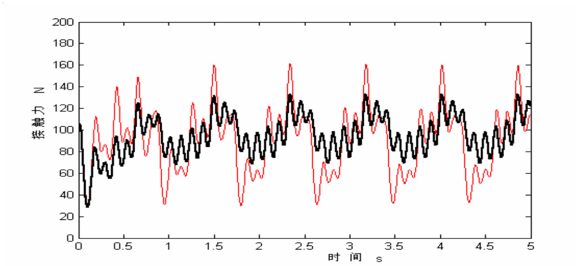
κ3*=−10215 N/μ，κ4*=−53 Nσ/μ，

κ5*=8929 N/μ，κ6*=538 Nσ/μ。

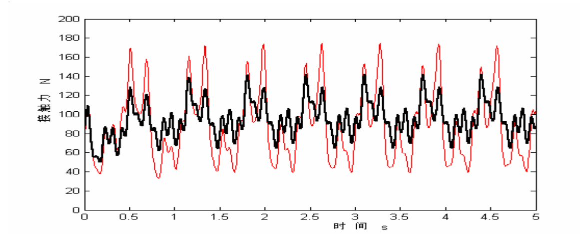
Simulation results as fig. 4~5 show，the dotted lines represent the, and the full lines represent the contact force time history of the active control pantograph system. （the following as the same）.



（a） v=200 km/h



(b) v=270 km/h



（c） v=350 km/h

Fig. 4 Time history of the contact force under the simple chain catenary

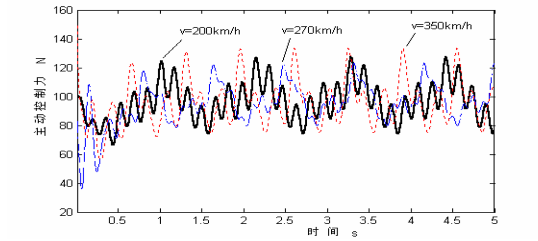


Fig. 5 Time history of the active control force under the simple chain catenary

B Simulation under the Elastic Chain Catenary

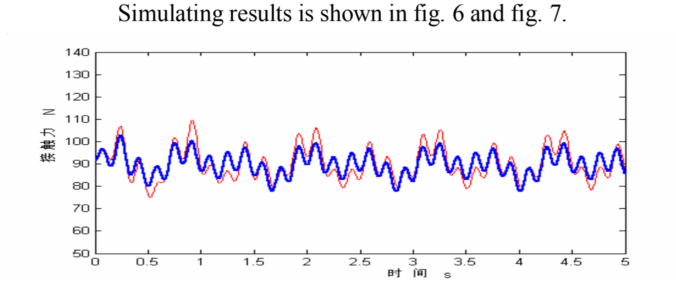
The values of the gains are

κ1*=−145 N/μ, κ2*=−36 Nσ/μ,

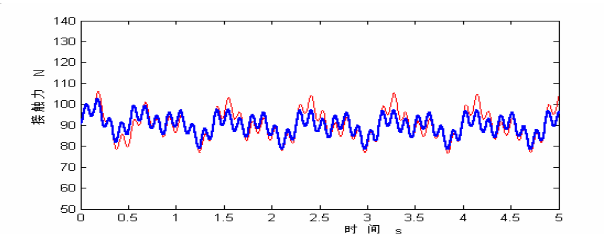
κ3*=−9981 N/μ, κ4*=−44 Nσ/μ,

κ5*=9103 N/μ, κ6*=541 Nσ/μ.

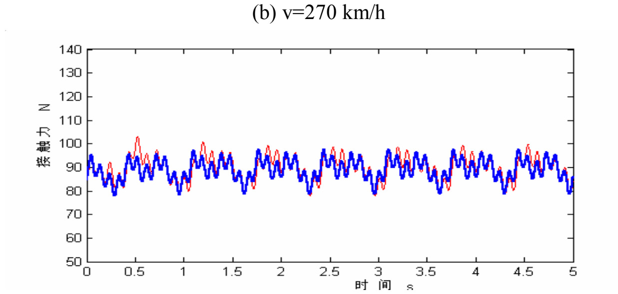
Simulating results is shown in fig. 6 and fig. 7.



(a) v=200 km/h



(b) v=270 km/h



(c) v=350 km/h

Fig. 6 Time history of the contact force under the elastic chain catenary

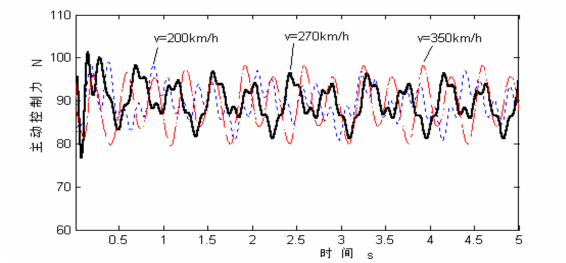


Fig. 7 Time history of the active control force under the elastic chain catenary

- *C Only partial state variables can be measured*

Because in the actual project uses， the state variables of the pantograph–catenary system can not be measured totally，or some of them is very difficult to measure，such as the displacement and velocity of the pantograph head. For this reason , we study the situation which only partial variables can be measured , So suppose that

$$\kappa 1* = \kappa 2* = \kappa 4* = 0$$

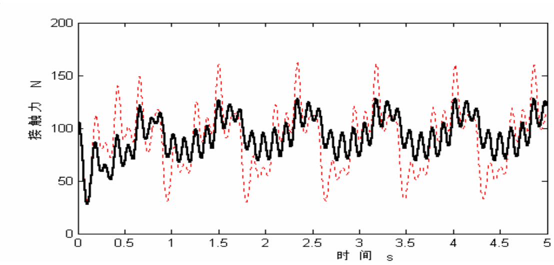
Then the active control force as,

$$F = -(k_3^* x_3 + k_5^* x_5 + k_6^* x_6) + F_0 \quad (9)$$

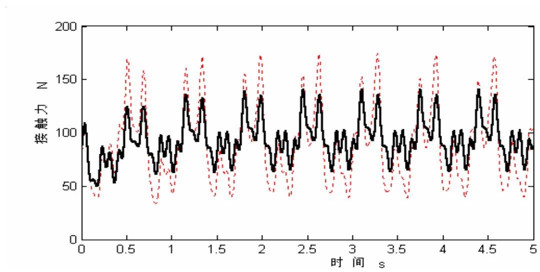
Simulation result is shows as fig. 8.

Analyzing above figures, can get the following conclusion.

- after exerting active control force，the contact force oscillation between catenay and pantograph has been restrained in all cases, especially for the simple chain suspension the control effectivity is more obvious，the magnitude which has been reduced in contact's force vibration is nearly 60N.
- increasing the run speed，contact force fluctuating will aggravate，for the simple chain type becomes more badly.



(a) v=270 km/h



(b) v=350 km/h

- Fig. 8 Time history of the contact force while partial state variables can be measured
- comparing the two types of catenaries ， the dynamic performance of elasticity chain suspension is better.
- while part of state variables can not be examined, active control pantograph can also get better control result.

VI CONCLUSION

In order to improve power collecting system for high speed trains, an active control strategy of the pantograph system is proposed based on the optimize control theory to the contact force in this paper.

Using the non-linear least square method, three kinds of stiffness curves are fitted by and a unified fitting formula is presented. The formula has a high precision, using it would simplified calculating process and reduce production design period .

The frame-actuated controller obt0ained a reduction in contact force variation and allowed an increase in train speed.

As the railway develop to high-speed, the investigation of active control pantograph would be a valid method to solve the problem of the stabilized power collections .

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