Integrated Simulation between Flexible Body of Catenary and Active Control Pantograph for Contact Force Variation Control*

Mohd Azman ABDULLAH**, Yohei MICHITSUJI***, Masao NAGAI**
and Naoki MIYAJIMA**

** Department of Mechanical Systems Engineering, Tokyo University of Agriculture and Technology,
2-24-16 Naka-cho, Koganei-shi, Tokyo, 184-8588, Japan

E-mail: mohdazman@utem.edu.my

*** Department of Mechanical Engineering, Ibaraki University
4-12-1 Nakanarusawa-cho, Hitachi-shi, Ibaraki, 316-8511, Japan

Abstract

Railway transport has been developed for a variety of requirements with a diversity of studies and technologies in recent years. In particular, the intercity railway transport that can be operated at speed of more than 350 km/h is the goal for the railway industry. Due to vibration and drag forces at high speed, contact force variation occurs between pantograph and catenary. This variation also causes instability in the pantograph and catenary interaction. In this study, multibody dynamics analysis is used to model the catenary. The integration of the catenary model and the pantograph model in the simulation flow produces contact force variations. A sinusoidal feed forward force and a simple feedback control force are applied to control the wave-like contact force fluctuations by means of active dampers. Evaluation of the combination of active control forces will produce optimized forces that may be able to maintain, thus improve the contact force variations.

Key words: Integrated Simulation, Flexible Multibody Dynamics, Absolute Nodal Coordinate Formulation, Active Control

1. Introduction

In fulfilling the global needs for high-speed intercity railway transport, a variety of researches have been continuously conducted. Such developments include improved comfort for passengers, environmental performance and high speed transport system which can reduce the time consumed for traveling. In particular, high-speed railway transports that can be operated at speed of more than 350 km/h has been the objective and goal for the railway industry. However, at high speed operating condition, the most essential element is the stability of current collection. Due to vibration of the car body and external drag force produced during high speed, contact force variation occurs between the current collector, the pantograph and overhead power supply line, the catenary. This variation causes instability in the interaction between pantograph and catenary.

Several European and Japanese researchers have investigated many problems associated with the complex interactions ⁽¹⁾⁽²⁾. In order to sustain sliding contact with catenary, adequate upward contact force must be exerted by pantograph ⁽³⁾⁽⁴⁾⁽⁵⁾. Large contact force causes mechanical damage to the pantograph system while zero contact force generates electrical arcs that increase the sliding wears and tears to the contact strip of the

*Received 15 Sep., 2009 (No. 09-0519)
[DOI: 10.1299/jmtl.3.166]

pantograph. As the result, the performance of the current collecting system is reduced and the maintenance cost is increased. Some researchers attempted to control the contact force variations analytically, for example fuzzy variable structure control at 170 km/h ⁽⁶⁾ and sliding modes control at 320 km/h ⁽⁷⁾.

In the previous dynamic analysis approach, the springs are used to model the interactions ⁽⁸⁾⁽⁹⁾⁽¹⁰⁾. This approach is too simple and not effective for the effect of the nonlinear deformation of the cable. Some analyses use partial differential equations to model the catenary as a continuous string or beam ⁽⁴⁾⁽¹¹⁾⁽¹²⁾. While partial differential equations are applicable for wave propagations study, they are not suitable for multibody dynamics. The use of linear finite element formulation in modeling the catenary cables ⁽¹³⁾ does not account for the coupling effects and nonlinearities.

In this study, the latest analysis of multi-body dynamics, Absolute Nodal Coordinate Formulation (ANC) ⁽¹⁴⁾⁽¹⁷⁾⁽¹⁸⁾ with damping force formulation ⁽¹⁵⁾ is used to model the overhead wire. The flexible body modeling approach using ANC law with damping forces ⁽¹⁶⁾ is applied. The catenary model based on multibody dynamics of complex systems for precise modeling and three degree of freedom (DOF) pantograph dynamic analysis is conducted. A simple sinusoidal feed forward control force and proportional feedback control are applied to the pantograph system to control the force variations. An optimum setting of the control force is produced from the comparison of evaluation functions.

2. Beam Analysis Technique

The damping force formulated by ANC is the new technique proposed to enhance the ordinary ANC ⁽¹⁵⁾. The amount of the beam deformation is expressed in a complex formula. Takahashi and Shimizu have proposed a simple equation to represent the motion of the small deformation in the stringing beam. This has led to shortening computational time in the beam simulation ⁽¹⁵⁾. In their research, the beam height and length are considered in the shape function, and the impact of the moment of inertia is derived from the mass matrix. Moreover, the simple equation of axial strain is represented by elastic force and bending deformation. The axial strain energy and bending deformation energy are derived from the damping force method at the same time.

In formulation of ANC model with damping force, they have considered Rayleigh damping equation. In this study, the same damping parameter is introduced in conjunction with drag coefficient parameter and the formulation of fluid resistance. The damping force, depending on the material of the beam, is produced by the axial strain energy U_{ID} , and bending deformation energy U_{ID} .

$$U_{ID} = \frac{1}{2} \int_0^{l_e} Da \dot{\epsilon}^2 dx \quad (1)$$

$$U_{ID} = \frac{1}{2} \int_0^{l_e} DI \dot{\kappa}^2 dx \quad (2)$$

Where $\dot{\epsilon}$ and $\dot{\kappa}$ are the nonlinear stiffness vectors and l_e is the length of one element, a is the cross section area of the wire and I is the second moment of inertia. The damping forces are derived as follows,

$$\mathbf{F}_{ID} = \left(\frac{\partial U_{ID}}{\partial \dot{\mathbf{e}}} \right)^T \quad (3)$$

$$\mathbf{F}_{ID} = \left(\frac{\partial U_{ID}}{\partial \dot{\mathbf{e}}} \right)^T \quad (4)$$

Where D is the introduced damping parameter and $\dot{\mathbf{e}}$ is the velocity of nodal coordinate vector. The drag force acting on the beam is derived from the following formulas,

$$\mathbf{Q}_{Dx} = C_D a_x \frac{\rho_f \cdot v_x^2}{2} \begin{bmatrix} 0 & -\frac{1}{2} & 0 & -\frac{l_e}{12} & 0 & -\frac{1}{2} & 0 & \frac{l_e}{12} \end{bmatrix}^T \quad (5)$$

$$\mathbf{Q}_{Dy} = C_D a_y \frac{\rho_f \cdot v_y^2}{2} \begin{bmatrix} -\frac{1}{2} & 0 & -\frac{l_e}{12} & 0 & -\frac{1}{2} & 0 & \frac{l_e}{12} & 0 \end{bmatrix}^T \quad (6)$$

Where a_x and a_y are the projected areas, while v_x and v_y are the speeds. Thus, the damping force formulated ANC law equation of motion with mass matrix $\mathbf{M}_a$, is as follows,

$$\mathbf{M}_a \ddot{\mathbf{e}} = \mathbf{Q} - \mathbf{F}_l - \mathbf{F}_t - \mathbf{F}_{ID} - \mathbf{F}_{ID} + \mathbf{Q}_{Dx} + \mathbf{Q}_{Dy} \quad (7)$$

2.1 Numerical analysis tool

As the beam equation of motion is formulated, Matlab/Simulink is used for analysis. In order to shorten the simulation time, the main part of the S-function programming will be using C-language. S-function with C-language increases the Matlab simulation speed and eases the control system design. In addition, Simulink acquires fast computing speed to calculate with the acceleration mode setting.

3. Catenary and Pantograph System Model

3.1 Catenary modeling

In this research, a heavy but simple catenary model used for high-speed train was constructed. Figure 1(a) shows the structure of the simple catenary and Fig. 1(b) shows the analysis model. In this model, the trolley line elasticity is supported by spring damper located in the catenary and hanger. The entire catenary is modeled using the ANC method with a small number of elements in order to reduce the analysis time. The model used an absolute coordinate system, with y-axis as height and x-axis as line direction. The starting point is at the trolley line standard height as $y=0$, and support point as $x=0$.

There is a tendency to which vertical stiffness of the trolley line is great when the extent is close to the support point and become weaker in the center of the span. So, the spring constant of the hanger is set high near the support point but low at the center of the span due to the influence of span cycle. The impact on the hanger is considered as an elastic body which is replaced by a spring-damper model. The hanger separation is about 5 m, therefore, in ANC model, the trolley line spring-damper interval is 5 m with elastic support. Since the hanger is restrained with the nodal element of the trolley, each element i is attached to the line with external force Q .

The tension of the trolley line leading to one end of the whole model is fixed with the spring-damper. Axial power and tension on x direction are acting on the other end of the line. The tension acting on n number of element of external force is Q^n . Next, the contact force effect from pantograph is considered. The contact force is designated as P and the element to which pantograph contacts is designated as i .

(a) Structure of heavy simple catenary

(b) Analysis model of heavy simple catenary

Fig.1 Model of catenary system

The distance from nodal element to the contact point is designated as x and the external force which affects the trolley line is designated as Q . Since the pantograph is traveling at a certain speed, the position of the contact point is also traveling at the same speed. From above, the equation of motion for the stringing is proposed as,

$$M\ddot{e} = Q - F + Q_H^i + Q_T^n + Q_P^i \quad (8)$$

3.2 Pantograph modeling

In the conventional pantograph model, the 2 and 3 mass systems have been replaced with simple model of contact strip and pan head analysis. However, using numerical analysis model, the design of pantograph parameter and control similar to an actual vehicle model is necessary. Therefore, in this study, a model considering the framework of a single arm pantograph is utilized as shown in Fig. 2. The contact strip and pan head are two mass system frameworks. Since they are restrained to the links of the pantograph, the framework is considered as 3DOF with one direction of rotation angle. The torque τ occurs at the lower frame due main spring k_f and damper c_f installed in the framework which produces the upward force. The damper in the actual design pulls the brake for the framework when the pantograph descends to a certain point. The control force CF is applied to the lower frame,

$$\tau = (CF)l_{f2} \quad (9)$$

Where l_{f2} is the length of the lower pantograph frame, which is equal to 1 m.

3.3 Catenary-pantograph system modeling

The pantograph model and catenary model interact together to generate contact force. The contact force acting on a point on top of the contact strip, with spring and damper models. This contact point moves together with the pantograph. Due to the relative displacement and relative speed of the top and bottom of the wear plate as well as flexible body of catenary, contact force occurs. If the relative speed is less than zero, than the pantograph is considered in bounce condition and is restrained with the spring and damper. Considering the contact point between pantograph and catenary, the free body diagram

force equation can be written as,

$$P = (y_c - y_{pl})k_c + (\dot{y}_c - \dot{y}_{pl})c_c \quad (10)$$

Where P is the contact force, y_c is the contact point displacement, y_{pl} is the contact strip displacement, k_c and c_c are the contact strip spring constant and damping parameter respectively. The analysis flow of catenary-pantograph system is shown in Fig. 2.

Fig.2 Analysis flow of catenary-pantograph system

3.4 Numerical analysis

The actual trolley line is constructed with a fixed height depending on the hanger in order to obtain steady contact with the pantograph. In the simulation, a spring is installed to replace the hanger and the damper is raised to an initial point where the trolley line becomes horizontal. The spring constant of the hanger⁽²⁰⁾ is shown in Table 1. The large spring constant of the hanger is close to the support point (hanger number 1 and 10) and low spring

constant at the center of the span (hanger number 5 and 6). Traveling speeds v for the analysis is set at 300 and 350 km/h. The parameters of catenary and pantograph are detailed in Table 2.

Table 1 Spring constant

Hanger number	Spring (N/m)	Damper (Ns/m)
1,10	8000	80
2,9	4500	45
3,8	3000	30
4,7	2500	25
5,6	2500	25

Table 2 Catenary and pantograph parameters

Parameter	Symbol	Value and unit
Length of span	s	50 m
Number of span	n_s	2
Number of halder	n_h	10/span
Number of element	n	100/span
Wire diameter	d	15.49×10^{-3} m
Young's modulus	E	130×10^9 Ns/m ²
Mass density	ρ	8920 kg/m ³
Drag coefficient	C_D	0.8
Fluid density	ρ_f	1.2929 kg/m ³
Gravity	g	9.8 m/s ²
Wire tension	T	19600 N
Mass of contact strip	m_1	6.8 kg
Mass of pan head	m_2	9 kg
Contact spring	k_c	39000 N/m
Contact strip spring	k_{p1}	39000 N/m
Pan head spring	k_{p2}	39000 N/m
Contact damper	c_c	120 Ns/m
Contact strip damper	c_{p1}	120 Ns/m
Pan head damper	c_{p2}	120 Ns/m
Mass of upper frame	m_{f1}	5 kg
Mass of lower frame	m_{f2}	5 kg
Length of upper frame	l_{f1}	1 m
Length of lower frame	l_{f2}	1 m
Frame spring	k_f	120 Nm/rad
Frame damper	c_f	60 Nms/rad

4. Control Force Approach

4.1 Feed forward

From the previous research, it is discovered that the contact force variations are repetitions of wave-like patterns ⁽¹⁹⁾. Since the nominal and required contact force is 54 N, lower and higher contact forces need pushing and pulling control forces correspondingly. Contact force peaks which are at zero need to be eliminated and at most need to be reduced. In this research a simple sinusoidal control force is applied to overcome the variations.

$$CF_{FF} = A \sin\left(\frac{3\pi x}{2s} - \theta\right) \quad (11)$$

Where, CF_{FF} is the feed forward (FF) control force (Fig. 3), A is the amplitude, θ is the phase angle. The distance of travel x and catenary span s are fixed. Amplitude A and phase angle θ are varied accordingly in order to find the optimum result. A simple evaluation function (EF) is used to evaluate the performance of sinusoidal FF control forces with different amplitude and phase angle. This evaluation function is also used to compare all three control force approaches. The lower the value of the evaluation function the better the performance of the control forces.

$$EF = P_{\max} - P_{\min} \quad (12)$$

Fig.3 FF control

4.2 Feedback

A simple proportional or direct feedback (FB) is applied to the system to compare performances. In this control approach, the same amount of force is fed into the system directly without additional features (Fig. 4).

Fig.4 Proportional FB control

4.3 Combination feed forward and feedback

The next control approach is by a combination of feed forward and feedback control forces (FFFB). However since the disadvantage of FB may reduces the performance, an additional 40 N offset force is fed into the combination for better improvement (FFFB2). This offset, also a type of FF, is required to push the contact point of the pantograph to its nominal 54 N contact force as shown in Fig. 5. This approach is believed to be necessary in order to compare the pantograph performance with different control approaches.

Fig.5 FFFB2 control

5. Numerical Analysis Result

The main objectives of this study are to reduce the maximum force whilst eliminate contact loss so that the range of force between maximum and minimum forces is small. Figure 6 shows the performance of proportional or direct FB control for $v = 350$ km/h. The high frequency about 20 Hz contact force repetition with amplitude about 25 N is due to the holder of the catenary and low frequency about 2 Hz contact force with amplitude about 50 N is due to the supporter of the catenary. In Fig. 6, the maximum peaks of the contact force are reduced significantly; however, the contact losses or zero forces are not eliminated. This is due to hard positive control forces that pull the pantograph downward continuously. Figure 7 illustrates the FF control force evaluation function analysis. The optimum values for FF control force has amplitude of 20 and phase angle of $\pi/6$. As shown in Fig.8, the use of this control force has greatest effect where it reduces the maximum peaks as well as eliminates the contact loses.

A combination of FF and FB is illustrated in Fig. 9. Even though the FF part seems effective, the FB part deteriorates the combination where few contact losses are noticed. In order to improve the combination, a 40 N offset force is applied to the system. The result as shown in Fig. 10, demonstrates a significant improvement in term of contact force performance where there is none contact loss. The additional 40 N offset force has improve the system performance by pushing the pantograph contact point to its nominal 50 N force. Figure 11 summarizes the evaluation of control approaches as compared to original contact force without control for speeds 300 and 350 km/h and for extended evaluation, Fig. 12 illustrates the contact loss ratio for each control approach. Contact loss ratio is the percentage ratio of total distance of loss contact or zero force over the total distance travel which is 100 m. Both figures are required in assessing the control approaches. From Fig. 11, all control approaches are in favored of improving the system when compared to the system without control. Even though the FF control for the case of 350 km/h is seem the worse compared to others, the range of the force between maximum and minimum forces are still acceptable for the pantograph to operate which is under 150 N. Figure 12 is the important level of evaluation. FB is the worse even compared to the system without control. This is because; FB control approach applied the contact force value directly to the system. Without any modification, high value in force fed into the system causes pulling effect to the pantograph contact point and therefore increases the loss. As a result, proportional FB control force approach is not suitable for this system. For both cases of speed, the FF and FFFB2 have definitely eliminated the contact losses, thus improving the system performance extensively.

Fig.6 Direct FB control for $V = 350$ km/h

Fig.7 Evaluation function for $V = 350\text{km/h}$

Fig.8 FF control for $V = 350\text{km/h}$

Fig.9 FFFB control for $V = 350\text{km/h}$

Fig.10 FFFB2 control for $V = 350 \text{ km/h}$

Fig.11 Contact force controls evaluation

Fig.12 Contact loss ratio

6. Conclusion

In this research, the design of the pantograph based on multibody dynamics catenary model technique is proposed. The ANC method with damping force formulation can be used to model the flexible body of catenary. The contact force fluctuation analysis of the catenary-pantograph system verified that high speed operation produces high amplitude of contact force frequency. The sinusoidal feed forward control force is excellent for high speed cases, while a combination of feed forward and feedback control is appropriate for taking advantage of proven performance feed forward approach. Both control approaches are appropriate for the system. In this research, only proportional feedback control approach is considered. Perhaps for the next step, other feedback methods such as on-off control and proportional-integral-derivative (PID) control can be applied and validated via experimentation.

From the result, it is possible to develop the controller design for the pantograph considering the parameter of the pantograph for passive control and active dampers for active control via simulation model. In the future, it is expected that the parameter identification experiment can be performed in order to model the catenary-pantograph accurately, and force variation control experiment to validate the simulation result. The development of the active pantograph control depends on the availability of the active damper with high frequency response to produce such control forces.

Acknowledgement

The author (Mohd Azman Abdullah) would like to acknowledge financial support from the Ministry of Higher Education of Malaysia and Universiti Teknikal Malaysia Melaka under the SLAI financial scheme.

References

- (1) Gostling, R. J. and Hobbs, A. E. W.: The Interaction of Pantograph and Overhead Equipment --- Practical Applications of a New Theoretical Method ---, *Proc. Institution of Mechanical Engineers*, Vol.197C, 61-69 (1983).
- (2) Makino, T., Yoshida, K., Seto, S. and Makino, K.: Running Test on Current Collector with Contact Force Controller for High-Speed Railways, *JSME International Journal, Series C*, Vol.40, No.4, 671-680 (1997).
- (3) Thompson, A. G. and Davis, B. R.: An Active Pantograph with Shaped Frequency Response Employing Linear Output Feedback Control, *Vehicle System Dynamics*, Vol.19, 131-149 (1990).
- (4) Arnold, M. and Simeon, B.: Pantograph and Catenary Dynamics --- A Benchmark Problem and Its Numerical Solution ---, *Applied Numerical Mathematics*, Vol.34, 345-362 (2000).
- (5) O'Connor, D. N., Eppinger, S. D., Seering, W. P. and Wormley, D. N.: Active Control of a High-Speed Pantograph, *Journal of Dynamic Systems, Measurement, and Control*, Vol.119, 1-4 (1997).
- (6) Huang, Y. J.: Discrete Fuzzy Variable Structure Control for Pantograph, *Electrical Engineering*, Vol.86, 171-177 (2004).
- (7) Pisano, A. and Usai, E.: Output-Feedback Regulation of the Contact Force in High-Speed Train Pantograph, *ASME Journal of Systems, Measurement, and Control*, Vol.126, 82-87 (2004).
- (8) Poetsch, G., Evans, J., Meisinger, R., Kortum, W., Baldauf, W., Veitl, A. and Wallascheck, J.: Pantograph/Catenary Dynamics and Control, *Vehicle System Dynamics Measurement, and Control*, Vol.28, 159-195 (1997).

- (9) Manabe, K.: High-Speed Contact Performance of a Catenary-Pantograph System, *JSME International Journal*, Vol.32, 31-40 (1989).
- (10) Yagi, T., Stensson, A. and Hardell, C.: Simulation and Visualization of the Dynamic Behaviour of an Overhead Power System with Contact Breaking, *Vehicle System Dynamics*, Vol.25, 31-49 (1996).
- (11) Wu, T., X. and Brennan, M. J.: Basic Analytical Study of Pantograph-Catenary System Dynamics, *Vehicle System Dynamics*, Vol.30, 443-456 (1998).
- (12) Zhai, W., M. and Cai, C. B.: Effect of Locomotive Vibrations on Pantograph-Catenary System Dynamics, *Vehicle System Dynamics*, Vol.29, 47-58 (1998).
- (13) Lesser, M., Karlsson, L. and Drugge, L.: An Interactive Model of a Pantograph-Catenary System, *Vehicle System Dynamics*, Vol.25, 397-412 (1996).
- (14) Shabana, A. A., Hussein, H. A. and Escalona, J. L.: Application of the Absolute Nodal Coordinate Formulation to Large Rotation and Large Deformation Problems, *Journal of Mechanical Design*, Vol.120, 188-195 (1998).
- (15) Takahashi, Y. and Shimizu, N.: Study on the Elastic Force for the Deformed Beam by means of the Absolute Nodal Coordinate Multibody Dynamics Formulation, *Transactions of the Japan Society of Mechanical Engineers, Series C*, Vol.67, No.655, 626-632 (2001).
- (16) Miyajima, N., Michitsuji, Y. and Takehara, S.: Swing-up Control of Mass Body Interlinked Tether, *Transactions of the Japan Society of Mechanical Engineers, Series C*, Vol.73, No.735, 2955-2961 (2007).
- (17) Seo, J-H., Kim, S-W., Jung, I-H., Park, T-W., Mok, J-Y., Kim, Y-G. and Chai, J-B.: Dynamic Analysis of a Pantograph-Catenary System using Absolute Nodal Coordinates, *Vehicle System Dynamic*, Vol.44, No.8, 615-630 (2006).
- (18) Shabana, A. A.: *Dynamics of Multibody Systems*, (3rd Edition), Cambridge University Press, pp.309-322 (2005).
- (19) Kusumi, S., Fukutani, T. and Nezu, K.: Diagnosis of Overhead Contact Line based on Contact Force, *QR of RTRI*, Vol.47, No.1, 39-41 (2006).
- (20) Electric Railways Handbook Editorial Committee Compilation, *Electric Railways Handbook (Denki Tetsudo Handbook)*, Corona Company (2007).