Mechanical model of yaw damper and its application in the simulation of vehicle system dynamics

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Abstract. Considering the limitation and inaccuracy of conventional simplified Maxwell model of yaw damper in the analyses of vehicle system dynamics, this paper established an accurate mechanical model of both one-way oil flow yaw damper and two-way oil flow yaw damper according to the analyses of their inner structure, and also contrasted two mechanical models of different oil flow direction and simplified Maxwell model in the vehicle system dynamics and analyzed the reason, especially in hunting stability, ride quality and small curve passability. The results show that the mechanical models of AMESim is superior than simplified Maxwell model in the accuracy of simulation.

1. Introduction

Yaw dampers are installed between the bogie and carbody, which dissipates the mechanical energy by its hole throttle principle. These yaw dampers can reduce lateral vibration of vehicle system and depress hunting motion [1]. Traditional used simplified Maxwell model has some limitation and errors in the simulation of vehicle system dynamics and cannot meet the need of accurate simulation.

The static and dynamic character and simulation of damper is always the research hotspot of scholars from both home and abroad. Zeng Jing [2] considered the influence of critical velocity from rubber joints of the yaw damper and secondary suspension lateral damper; Duan Yanwen [3] analyzed the dynamic three-dimensional flow field in damper and the influence of damping character from the velocity of opening valve and oil temperature; Ma Tianfei [4] established simulation model of double cylinder superposition valve block damper by AMESim and contrasts its static and dynamic character with experimental data; Lewandowski R [5], Weigel M [6] established nonlinear numerical model of damper using neural network method. Stefano Bruni [7], Kasteel RV [8] constructed mechanical model of damper by fluid method after analyzing the structure of damper; Beghi A [9], Pracny V [10] established the combination of half mechanical model and half blackbox model and verified the nonlinear character of single cylinder.

Based on the research of formers, this paper established the accurate mechanical model of one-way oil flow yaw damper and two-way oil flow yaw damper by AMESim after analyzing the inner structure of yaw damper. And contrasted these two AMESim models with simplified Maxwell model in the simulation of vehicle system dynamics.

2. Establishment of Accurate Mechanical Models of Yaw Damper

Commonly used yaw dampers are included one-way oil flow yaw damper shown as Figure 1 and two-way oil flow yaw damper shown as Figure 2.
The mechanical model mainly consists of the property of oil, flow loss and damping valves [11,12].

2.1. Property of Oil
The evaluating indicators of oil property includes fluidity, compressibility and density. Fluidity is evaluated by coefficient of dynamic viscosity and compressibility is evaluated by bulk modulus.

In the working condition, dynamic viscosity isn’t constant and is influenced by real-time pressure $P$, temperature $T$ and the proportion of mixed air $ε$. The relationship is shown in Eq.1.

$$\mu(P, T, ε) = \mu_0 (1 + 1.5ε) \exp\left(\alpha(P - P_0) - \lambda(T - T_0)\right)$$

In Eq.1, $\mu_0$ is initial dynamic viscosity; $P_0$ is initial pressure; $\alpha$ is coefficient between viscosity of oil and pressure.

Compressibility of oil can be evaluated by bulk modulus $\beta_y$ which is relevant to instantaneous pressure $P$ and the proportion of mixed air $ε$. The relationship is shown in Eq.2.

$$\beta_y = \left[\frac{1 + ε_0(P_0/P)}{1 + ε_0β_y(P_0/P)^2}\right]β_{y0}$$

In Eq.2, $β_{y0}$ is initial bulk modulus of pure oil.

The density of oil is relevant to instantaneous pressure $P$, temperature $T$ and the proportion of mixed air $ε$. The relationship is shown in Eq.3.

$$\rho(P, T) = \frac{P_0}{1 + ε_0P_0/P} \cdot \exp\left[β(P - P_0) - \lambda(T - T_0)\right]$$

In Eq.3, $β = 1/β_y$.

2.2. Flow Loss
Inner flow loss of damper includes the loss of leak and the loss of compress $Q_c$. And the loss of leak
includes the leak loss between guide lid and piston $Q_g$, the leak loss between piston and pressure cylinder $Q_p$ and the leak between pressure cylinder and bottom valves $Q_b$. The loss of leak is shown in Eq.4 and the loss of compress is shown in Eq.5.

$$Q_g + Q_p + Q_b = \begin{cases} \frac{\pi d^4 \sigma_1^3 \Delta P}{12 \mu l} + \frac{\pi D^2 \sigma_2^3 \Delta P}{12 \mu l} + \frac{\pi D^2 \sigma_3^3 \Delta P}{6 \mu \ln(R_2/R_1)}, & x' > 0 \\ \frac{\pi D^2 \sigma_2^3 \Delta P}{12 \mu l} + n \frac{\pi D^2 \sigma_3^3 \Delta P}{6 \mu \ln(R_2/R_1)}, & x' < 0 \end{cases}$$ (4)

In Eq.4, $d$ is diameter of piston rod; $D$ is diameter of piston; $l$ is the length of seal area of guide lid; $L$ is the length of seal area of piston; $\sigma_1$ is the interval between guide lid and piston; $\sigma_2$ is the interval between slave cylinder and piston; $\sigma_3$ is the interval between guide lid or bottom valve and piston $\Delta P$ is pressure differential between stretching cylinder and storage cylinder; $R_i$ is inner radius of pressure cylinder; $R_2$ is outer radius of pressure cylinder; $x'$ is velocity of piston; For one-way oil flow damper $n = 2$ and for two-way oil flow damper $n = 1$.

The loss of compress can be calculated by Eq.5.

$$Q_c = \frac{-V_{oil} \rho_0}{\rho} \exp\left[\beta_{1}(P_{\Delta P}) - \lambda_{1}(T_{\Delta T})\right] \left(\chi \Delta P - \frac{-\lambda_{1}P}{P + \varepsilon_{0}P_{\Delta T}}\right)$$ (5)

In Eq.5,

$$\chi = \frac{\beta P_{P}^{2} + \beta \varepsilon_{0}P_{P} + \varepsilon_{0}P_{0}}{(P + \varepsilon_{0}P_{0})^{2}} + \frac{(P_{\Delta P})(\beta_{1}\varepsilon_{0}P_{P} + 2\beta_{1}^{2}\varepsilon_{0}P_{P} - \beta_{1}\varepsilon_{0}P_{P})}{(P + \varepsilon_{0}P_{0})^{3}} + \frac{\beta_{1}^{2}\varepsilon_{0}P_{P}}{P + \varepsilon_{0}P_{0}}.$$ (6)

2.3. Damping Valves

There are stretching process and compressing process existing in damper. In stretching process, the relationship between the flow $Q_R$ which crosses damping valves and pressure drop $\Delta P$ is shown in Eq.6.

$$Q_R = \begin{cases} \frac{\pi d^4 \Delta P}{64 \mu l}, & 0 < x' < v_1 \\ \frac{\pi d^4 \Delta P}{64 \mu l} + 3C_d A_h \sqrt{2\Delta P / \rho}, & v_1 < x' \leq v_2 \\ \frac{\pi d^4 \Delta P}{64 \mu l} + 3C_d (A_{h1} + A_{h2}) \sqrt{2\Delta P / \rho}, & x' > v_2 \end{cases}$$ (6)

In compressing process, the relationship is shown in Eq.7.
In Eq. 6 and Eq. 7, \( A_{hi} \) is open area of changeable throttle hole of damping valves; \( C_d \) is coefficient of flow; \( d_b \) is the diameter of through-hole of bottom damping valve; \( l_b \) is the length of through-hole of bottom damping valve.

According to the analysis above and the relevant parameters of yaw damper used in high-speed railway, this paper established mechanical simulation model by AMESim. The model of one-way oil flow damper model is shown in Figure 3 and two-way oil flow damper is shown in Figure 4. These models will be used for comparison with Maxwell model and experimental data in the following paragraph.

### 3. The Application of Vehicle System Dynamics

Commonly used model in the simulation of vehicle system dynamics is simplified Maxwell model. When the relative velocity of two points of damper is bigger than the biggest velocity of F-v curve, the model uses epitaxial interpolation to find optimum velocity. This method may bring errors. Therefore, this part conducted co-simulation of vehicle system dynamics in some aspects by using Simpack-Simulink-AMESim data interaction model. This part is the contrast of one-way oil flow damper AMESim model, two-way oil flow damper AMESim model and simplified Maxwell model.

#### 3.1. Response of Sine Excitation

For high-speed rail way on service with one-way oil flow yaw damper and two-way oil flow yaw damper, the same sinusoidal track excitation is given and the results are shown in Figure 5 and Figure 6.
It can be concluded from the results that under the same track excitation on the vehicle, the stretching damping force is far greater than compressing damping force in one-way oil flow yaw damper, but the stretching force is near to the compressing force in two-way oil flow yaw damper. This character is the reflect of dynamic character of yaw damper.

3.2. Hunting Motion Stability
Hopf bifurcations curve is a kind of evaluation criterion of hunting motion. Figure 7 is the hopf curve of simplified Maxwell model, AMESim model of one-way oil flow damper and AMESim model of two-way oil flow damper.

It can be concluded from the results that three models are all subcritical bifurcations. The critical speed of simplified Maxwell model is lower than AMESim model of one-way oil flow damper. The stability of simplified Maxwell model is lower than AMESim model of one-way oil flow damper. This is because in the initial stage the AMESim model provides small damping force but provides big damping coefficient, and this will consume lots of vibration energy but cannot make a higher critical speed. The critical speed of simplified Maxwell model is higher than AMESim model of two-way oil flow damper. The stability of simplified Maxwell model is higher than AMESim model of two-way oil flow damper. This is because simplified Maxwell model look for damping force by the relative speed of the two points of damper using epitaxial interpolation, and this will lead to damping force and damping coefficient are far greater than simplified model when the relative speed is high. Thus the motion of wheel will quickly converge after large amplitude excitation and make a higher stability of AMESim model.

3.3. Ride Quality
Ride quality mainly considers lateral ride quality, vertical ride quality and comfortability. This paper
uses sperling index to evaluate ride quality. The results of contrast are shown in Figure 8, Figure 9, Figure 10.

![Figure 9. Vertical sperling index](image1)

![Figure 10. Comfortability index](image2)

It can be concluded from the results that simplified Maxwell model is superior than the two AMESim model s in lateral ride quality, vertical ride quality and comfortability. Considering the hunting stability of vehicle, a superior hunting stability usually means a superior ride quality and comfortability. That is because the computing acceleration value of AMESim model is larger than the simplified model when the frequency is smaller than the maximum weight frequency (usually 5.4Hz) lateral ride quality. When passing through the maximum weight frequency, the situation is opposite. Because of the weighting coefficient of frequency in AMESim model is larger than that of simplified Maxwell model, that may lead to a higher index of lateral sperling and comfortability.

3.4. Curve Pass Ability
For small radius curve, the radius is 250m, the length of transition curve is 15m, the length of circle curve is 100m, and without any superelevation. The main criterions are vertical force of wheel-rail, lateral force of wheel-axle, rate of wheel load reduction and coefficient of derailment. The results of contrast are shown in Figure 11, Figure 12, Figure 13 and Figure 14.

![Figure 11. Vertical force of wheel-rail](image3)

![Figure 12. Lateral force of wheel-axle](image4)
It can be concluded from the results that for one-way oil flow damper, simplified Maxwell model has a poorer performance in small curve passability than AMESim model. Because simplified Maxwell model causes larger damping coefficient than AMESim model when crossing the small curve, that will lead to the left yaw damper and right yaw damper of the same bogie providing larger secondary suspension gyroscopic moment, and that also result in a poorer performance of curve passability of simplified Maxwell model.

For two-way oil flow damper, the simplified Maxwell model has a superior performance than AMESim model in small curve passability. In the simulation process, AMESim model can fully reflect the character of two-way oil flow yaw damper which the compressing damping force is larger than stretching damping force in large stroke when crossing the small curve. And the simplified Maxwell model cannot accurately reflect this character. Therefore, AMESim model provides larger secondary suspension gyroscopic moment by its left and right yaw damper of bogie, and that will lead to a poorer performance of passability of AMESim model.

4. Conclusion
(1) Dynamic viscosity, bulk modulus and density of oil in damper are influenced by the inner instantaneous pressure $P$, temperature of oil $T$ and the proportion of mixed oil $\varepsilon$. The flow loss in working process mainly includes leaking loss and compressing loss.

(2) One-way oil flow damper and two-way oil flow damper show their difference in the vibration response of vehicle system.

(3) There are some limitations and errors for commonly used simplified Maxwell model as input force element in vehicle system dynamics. That mainly because this model looks for damping value by the relative speed of the two point at both end of damper using epitaxial interpolation. That will lead to a larger error of damping force and damping coefficient between model and actual value.

Acknowledgement
This research was financially supported by National Key R&D Program of China (2016YFB1200505)

References
[1] Lou Ren, Shi Huailong. Vehicle system dynamics and its application [M]. Chengdu: Southwest Jiaotong University Press, 2018
[2] Zeng Jing, Wu pingbo. The influence of rubber joint stiffness of damper for the critical velocity of railway passenger vehicle [J]. China Railway Science, 2008,29(2):94-98.
[3] Duan yanwen, Li Renxian, Wu jianbin. Analysis of influencing factors damping character of damper in high-speed railway [J]. Journal of Transportation Engineering and Information, 2014(4):91-96.
[4] Ma Tianfei, Cui Zefei, Zhang Minmin. Modelling and simulation of double cylinder superposition valve block damper basing on AMESim. Journal of Mechanical Engineering, 2013, 49(12):123-130.
[5] Lewandowski R. Identification of the parameters of the Kelvin–Voigt and the Maxwell fractional models, used to modeling of viscoelastic dampers [J]. Computers & Structures, 2010, 88(1–2):1-17.

[6] Weigel M. Nonparametric Shock Absorber Modelling Based on Standard Test Data [J]. Vehicle System Dynamics, 2002, 38(6):415-432.

[7] Stefano Bruni. Modelling of suspension components in a rail vehicle dynamics context [J]. Vehicle System Dynamics, 2011, 49(7):1021-1072.

[8] Richard Van Kasteel. A new shock absorber model for use in vehicle dynamics studies [J]. Vehicle System Dynamics, 2005, 43(9):613-631.

[9] Beghi A. Grey-box modeling of a motorcycle shock absorber for virtual prototyping applications [J]. Simulation Modelling Practice & Theory, 2007, 15(8):894-907.

[10] Pracny V. Full vehicle simulation using thermo-mechanically coupled hybrid neural network shock absorber model [J]. Vehicle System Dynamics, 2008, 46(3):229-238.

[11] Chen Zhuoru. Engineering Fluid Mechanics, third edition [M]. Beijing: Higher Education Press, 1994.

[12] Yang Guozhen, Wang Futian. hydraulic damper of rolling stock [M]. Beijing: China Railway Publishing House, 2003.