Study on Curving Characteristic of Vehicles with Non Linear Air Suspension*

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This paper presents the dynamic analysis of curving behavior of railway vehicles. Curving simulation of vehicles running at low speed is important from a point of view of safety. It haven't often been done and results of simulation usually differ from measured them. Though the influence of leveling valves and differential pressure valves can be generally ignored in the case of simulation of vehicles running at high speed, the effects of them must be considered at low speed. Therefore we improved A'GEM which is multibody dynamics software to add non linear air suspension elements with leveling valves and differential pressure valves. The results of curving simulation were compared with experiments with good agreement.

Key Words: Railway, Dynamics of Machinery, Modeling, Transient Response, Railway Vehicle, Curving Simulation, Non Linear Air Suspension, Leveling Valve, Differential Pressure Valve, Multibody Dynamics

1. Introduction

While increase of speed and reduction of maintenance of railway vehicles have recently become an important problem, an active study and development of body tilting devices, steering trucks and like for the purpose of improving curving speed and reducing lateral force specially for the conventional lines with many curves are being carried out together with curving simulation of railway vehicles when they are running at the speed exceeding the balanced speed.

Commuter trains, however, are compelled to travel at low speed during the rush hours in the morning and evening due to overcrowded railway schedule. Viewing from the safety standpoint, it is considered that curving simulation at low speed is important because large lateral force generates even at low speed and derailment coefficient becomes high. However, there are few studies for this case, more over, actual measurement results specially of sharp

curves are difficult to illustrate. Railway vehicles are provided with height adjustment devices, so called the leveling valves: LV, for automatically maintaining the floor height to a certain level. Safety devices, so called the differential pressure valves: DPV, are also provided to keep the car body from tilting to one side when one of the air springs blows out and the right and left floor height changes. Though the influence of leveling valves and differential pressure valves can be generally ignored in the case of simulation of vehicles running at balanced speed or more, the effect of them should not be disregarded at low speed. In the usual multibody dynamics software packages, however, LV and DPV are not included as linear spring and damper oscillation model are being used for air suspension models of curving simulation program. This is considered as one of the reasons for the calculated values and the measured values not being easily met.

The multibody dynamics software [A'GEM]⁽¹⁾ was accordingly improved by changing the air suspension model from the existing linear spring and damper parallel model to a model having non linear characteristic including leveling valves and differential pressure valves. Curving simulation was carried out by using the improved [A'GEM] and the calculated values were compared with measured results.

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2. Symbols

Symbols used in this paper are as follows :

- A : Choke opening area
- A_s : Effective area of air spring
- C_d : Flow Coefficient at choke
- F : Air spring vertical force
- g : Gravity acceleration
- M_B : Car body mass
- P_0 : Pressure in air spring at neutral height
- P_1 : Pressure fluctuation in air spring
- P_2 : Pressure fluctuation in auxiliary air tank
- P_{AT} : Atmospheric pressure
- P_D : Choke downstream (low pressure side) pressure
- P_s : Pressure in air spring
- P_U : Choke upstream (high pressure side) pressure
- Q_1 : Air mass flowing from auxiliary air tank into air spring
- Q_2 : Air mass flowing from LV into air spring
- Q_3 : Air mass flowing from differential pressure valve into auxiliary air tank
- q : Air mass flow rate passing through choke
- V_A : Volume of air spring at neutral height
- V_B : Volume of auxiliary air tank
- Z_s : Air spring vertical deflection
- κ : Ratio of specific heat of air
- ρ_0 : Air density of air spring at neutral height
- ρ_U : Choke upstream air density

3. Calculation Model

3.1 Vehicle model

The vehicle prepared for the simulation is a commuter type with bolsterless trucks. The vehicle model is a single vehicle and the number of rigid body is a car body, two trucks and four wheel sets, total of seven bodies. The number of degrees of freedom is 17 as shown below :

Car body, truck : Right and left, yawing and rolling

Wheel set : Right and left, and yawing

All springs and dampers except ones of the air spring in the vertical direction were linear. Regarding the model of the air spring in the vertical direction, the linear damper was left without change among the existing linear spring and damper parallel system but the spring was newly added to the model as an external force provided with non linear characteristic, as shown in Fig. 1.

3.2 Air suspension model

3.2.1 Air suspension system The outline of the air suspension system model is as shown in Fig. 2. It is composed of an air spring and an auxiliary air tank and leveling valve (LV) is provided to keep the

Fig. 1 Dynamic model of vehicle

Fig. 2 Air suspension system model

deflection of the air spring within a certain range. When the air spring deflection exceeds a set value, LV acts to return it within the set value by opening its supply or discharge valve and supply or discharge air. It is provided with the dead band and a time-delay characteristic so that it may not response to vibrations. Air is supplied or discharged only when air spring continues to expand or contract in excess of the dead band for a period longer than the action time-delay. On the contrary, air supply or discharge action is instantly stopped when the air spring deflection enters the dead band while supplying or discharging air, and time-delay does not occur. The right and left air springs are connected together through a differential pressure valve (DPV) which functions to maintain the pressure difference between the two air springs below the DPV set value by opening the valve when the pressure difference between the springs exceeds the set value and allowing air to flow from the high pressure side to the low pressure side.

Thus air spring force of the existing spring and damper parallel model has been determined by solely the air spring deflection using a certain spring constant but, by incorporating LV and DPV in the air suspension system model, non linear air spring force parameterized by air spring deflection, pressure in air spring and time can be calculated and a more realistic air suspension behavior can be reproduced.

3.2.2 Pressure in air spring and air spring vertical force Volume, air mass and the pressure in air spring at a neutral height are to be V_A , $\rho_0 V_A$, and P_0 , respectively. It is assumed that air mass Q_1 flows into

the air spring from the auxiliary air tank and air mass Q_2 flows in from LV when the air spring expands by Z_s as shown in Fig. 3. The pressure in air spring $P_0 + P_1$ at this time is presented by the following formula^{(2),(3)}:

$$P_0 + P_1 = P_0 \left\{ \frac{\rho_0 V_A + Q_1 + Q_2}{\rho_0 (V_A + A_s Z_s)} \right\}^\kappa \quad (1)$$

Air of mass Q_3 flows into the auxiliary air tank from DPV so the pressure in the auxiliary air tank at this time is presented by the following formula:

$$P_0 + P_2 = P_0 \left\{ \frac{\rho_0 V_B - Q_1 + Q_3}{\rho_0 V_B} \right\}^\kappa \quad (2)$$

By putting it $P_0 + P_1 = P_0 + P_2 = P_s$ because pressure in air spring equal pressure in auxiliary air tank under stationary condition, pressure in air spring P_s is:

$$P_s = P_0 \left\{ \frac{V_A + V_B + (Q_2 + Q_3)/\rho_0}{V_A + V_B + A_s Z_s} \right\}^\kappa \quad (3)$$

From this pressure in air spring, air spring vertical force F is found by the following formula.

$$F = A_s (P_s - P_{AT}) - M_B g / 4 \quad (4)$$

3.2.3 Air flow rate flowing through LV and DPV Air flow rate flowing through LV and DPV is dealt with as air flowing through a choke as shown in Fig. 4. Air flow rate passing through the choke is determined by the ratio P_D/P_U between choke upstream pressure P_U and downstream pressure P_D . At this time, mass flow rate q is found by the following formula⁽⁴⁾:

(a) In case of $\frac{P_D}{P_U} \geq \left(\frac{2}{\kappa + 1} \right)^{\kappa/(\kappa-1)}$

$$q = C_d A \sqrt{P_U \cdot \rho_U \cdot \frac{2\kappa}{\kappa-1} \left\{ \left(\frac{P_D}{P_U} \right)^{2/\kappa} - \left(\frac{P_D}{P_U} \right)^{(\kappa+1)/\kappa} \right\}} \quad (5-a)$$

(b) In case of $\frac{P_D}{P_U} \leq \left(\frac{2}{\kappa + 1} \right)^{\kappa/(\kappa-1)}$

$$q = C_d A \sqrt{P_U \cdot \rho_U \cdot \kappa \left(\frac{2}{\kappa + 1} \right)^{(\kappa+1)/(\kappa-1)}} \quad (5-b)$$

Fig. 3 Air spring model

Fig. 4 Air flow of LV and DPV

4. Curving Simulation

4.1 Calculation conditions

Curving simulations were carried out after improving the [A'GEM] and adding the non linear air suspension system model as indicated in section 3.2. Calculations were made by setting the conditions of the track, vehicle running speed, LV and DPV actions as mentioned below and the principal details of the vehicle used for the calculation is listed in Table 1:

(1) Track conditions:

0 to 20 m: straight track

20 m to 80 m: Transient curve track

80 m to 150 m: Circular curve track

Radius of curvature: 403 m, cant: 100 mm, slack: 5 mm

(Balanced speed: 69 km/h)

50 N rail was used for rail profile and the value estimated from the measured value of the derailment coefficient of the first axle inner rail side was used for the friction coefficient between wheel and rail. The measured value of the lateral force later referred to is the running test data of a vehicle that have actually ran on the track of quite the same conditions as mentioned above.

(2) Vehicle running speed: 20 km/h, 40 km/h, 70 km/h

(3) LV dead band: ± 5 mm, LV action time-delay: 2 sec., DPV set pressure: 0.08 MPa

Table 1 Specification of calculation model

Name	Unit	Value
Carbody mass	kg	18535
Truck mass	kg	1589.5
Wheelset mass	kg	1104
Inertial radius of car body mass in roll	m	1.391
Inertial radius of car body mass in yaw	m	5.687
Inertial radius of truck mass in roll	m	0.635
Inertial radius of truck mass in yaw	m	1.135
Inertial radius of wheelset mass in yaw	m	0.54
Height of center of gravity of car body mass	m	1.788
Height of center of gravity of truck mass	m	0.6
Wheel radius	m	0.43
Height of secondary suspension	m	0.948
Height of secondary lateral damper	m	0.85
Height of trailing equipment	m	0.41
Wheelbase	m	2.1
Center distance between trucks	m	13.8
Distance between contact points of wheels and rail	m	1.12
Distance between primary suspensions	m	1.64
Distance between secondary suspensions	m	1.93
Longitudinal stiffness of primary suspension per wheelset	N/m	3.6×10^7
Lateral stiffness of primary suspension per wheelset	N/m	3.4×10^7
Vertical stiffness of primary suspensions per wheelset	N/m	1.52×10^8
Longitudinal stiffness of secondary suspension per truck	N/m	1.6×10^5
Lateral stiffness of secondary suspension per truck	N/m	2.7×10^5
Lateral damping of secondary suspension per truck	Ns/m	8.0×10^4
Vertical damping of secondary suspension per truck	Ns/m	4.82×10^4
Longitudinal stiffness of trailing equipment per truck	N/m	7.65×10^8
Wheel conicity		1/20
Volume of air spring	m ³	0.022
Volume of auxiliary air tank	m ³	0.047
Effective area of air spring	m ²	0.222
Pressure in main reservoir	Pa	8×10^5
Flow coefficient at choke of leveling and differential valves		0.7
Choke area of leveling and differential valves	m ²	7.07×10^{-6}

4.2 Simulation results

4.2.1 With LV and DPV One example of the curving simulation result conducted by applying the calculation conditions as mentioned in 4.1 above is shown in Fig. 5. Seven calculation results at the speed of 20 km/h are shown. They are the simulation results in order of air spring deflection (positive in contraction direction), pressure in air spring, air mass flow rate from LV, air mass flow rate from DPV, change of air mass in air spring, car body and truck rolling angle (positive in inner direction of curve), and wheel loads of the first and the third axles. Among the legends in the figures, No. 1 and No. 2 represent the front truck and the rear truck, and #1 and #3 represent the first and the third axles, respectively. The words, inside and outside respectively mean the inner

rail side and the outer rail side.

Regarding the air spring deflection, expansion and contraction of those on the front truck change places as the vehicle leaves an transient curve and enters an circular curve (around 80 m) by the influence of the track distortion in the transient curve. On the other hand, expansion and contraction of those on the rear truck change places in the circular curve and deflection in such section is smaller than that of those on the front truck. Air supply/discharge by LV is observed on air springs on both the front and rear trucks when the vehicle is running on an transient curve because deflection exceeds the dead band but, on the circular curve, LV action is seen only on the front truck suspension. As the vehicle enters an transient curve, the pressure in air spring changes diagonally

Fig. 5 Simulation results (with LV and DPV)

and the front truck outer rail side and the rear truck inner rail side becomes higher, while the front truck inner rail side and the rear truck outer rail side becomes lower. The internal pressure difference between the right and left air springs on the rear truck becomes large, as compared with that of the front truck, and exceeds DPV set pressure so the DPV opens for a short time. When the vehicle enters a circular curve, the internal pressure difference between the right and left air springs on the front truck becomes smaller but the internal pressure difference between the springs on the rear truck is left large. The wave form of the car body rolling angle becomes irregular by the influence of LV and DPV, and tilts to the inner side of the curve due to overcant. On the other hand, the front and the rear trucks roll to the opposite sides in the transient curve, the front truck rolling to the outer side of the curve, while the rear truck rolls to the inner side of the curve. As the vehicle enters a circular curve, the front truck also starts to roll to the inner side of the curve. As to wheel load, the change of wheel load in the inner and outer rails reverses in the transient curve with the first axle (front truck). With the third axle (rear truck), the wheel load unbalance between the inner and outer rails becomes large, specially on the outer rail side where there is about 40% wheel load reduction.

4.2.2 Without LV and DPV An example of simulation results in the case of setting enough LV dead band and DPV set pressure so that the LV and the DPV may not operate is shown in Fig. 6. The figure indicates the calculation results when the vehicle is operated at the speed of 20 km/h, same as with the LV and the DPV acting, but results pertaining to air flow rate and air mass are omitted because air does not flow through LV or DPV.

Under this calculation conditions, the air suspension system model is confined air spring because air does not flow through LV or DPV. Accordingly, the air suspension system model in this case can be considered equivalent to the existing linear spring and damper parallel model.

Air spring deflection is much larger as compared with the case of LV and DPV operating, and the front and rear truck spring deflections are the same in the circular curve. In spite of the large air spring deflection, the change of pressure in air spring is small as compared with the case of LV and DPV operating because air mass in the air spring does not change. The car body rolling angle is observed to be considerably large due to absence of height adjustment function by an LV. On the other hand, the rolling angles of the front and rear trucks become the same in the circular curve, as in the case of the air spring

deflection. As to wheel load, wheel load unbalance between the first and the third axles, or in other words, front and rear trucks is gone in the circular curve. For this reason, wheel load unbalance of the rear truck between the inner and the outer rails reduces and the wheel load reduction rate of the third axle on the outer rail side becomes about 30%. This indicates that LV is effective to prevent wheel load of the front truck from reducing but it reversely increases wheel load reduction of the rear truck. In order to prevent this phenomenon, it is generally effective to act the differential pressure valve low.

5. Discussions

The simulation results were compared with the measured values in order to judge the validity of the

Fig. 6 Simulation results (without LV and DPV)

Fig. 7 Comparison of lateral force

simulation. Comparison of the calculation values of lateral force of the first axle outer rail side with the measured values is shown in Fig. 7. The calculation value indicates the cases of LV and DPV operating (LV, DPV-ON) and LV and DPV not operating (LV, DPV-OFF). The measured value of the lateral force to be compared is the running test data taken when the vehicle was actually run on quite the same track conditioned for the simulation, as previously mentioned.

There is almost no difference between the calculation model and the measured value at the balanced speed (70 km/h=19.4 m/s). This is because LV functioning time becomes short due to the short vehicle passing time through a curve thus resulting less influence of LV and DPV. At low speed, however, the difference between the two becomes large and, at a very low speed, the value of the LV-DPV operating model becomes closer to the measured value thus makes it easier to explain the measured value. With the LV-DPV operating model, it can be explained that lateral force does not increase at a very low speed similarly to the measured value because the influence of the non linear air spring caused by the function of LV and DPV greatly reflects on the calculation results of the wheel load and the like the more the vehicle is operated at low speed. This makes it clear that a non linear air suspension including an LV and a DPV should be taken into consideration in making a curving simulation at low speed.

Though comparison of the measured value was

made only with the lateral force in this paper, for future development, it is considered necessary to make further running tests and compare other items such as pressure in air spring and car body rolling angle with the measured values as well. This may lead to possibility of making decisions of coefficient of air flow rate and the like. It can also be considered to apply control algorithm to vehicles provided with car body tilt control device including an active suspension by incorporating it into a program.

6. Conclusions

The following results are obtained in this study :

(1) By the fact that multibody dynamics software [A'GEM] was improved and a non linear air suspension system was added to the model, possibility of making models of an active suspension and the like was confirmed.

(2) By the fact that non linearity was taken in consideration, actual vehicle behavior is now physically explainable.

(3) It was made apparent that a calculation using a non linear air suspension system model including LV and DPV is necessary for curving simulation at low speed by means of comparing it with the measured values.

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