Construction of Railway Vehicle Simulation Model
Taking into Account Air Springs Deflation

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When the air springs on a railway vehicle go flat, wheel load is decreased by track irregularity. In order to examine the running safety of the railway vehicle under the air spring deflation, running tests were carried out, and various data such as wheel load, lateral force, force acting on the air spring and the amount of climbing of the wheel were obtained. Furthermore, we constructed a numerical simulation model of a railway vehicle in consideration of an air spring deflation, and its validity was confirmed through the comparison with the test results.

Keywords: air spring deflation, vehicle running safety, wheel load reduction, amount of climbing of the wheel, curving simulation

1. Introduction

Air springs used on railway vehicles are designed to insulate car bodies against vibrations from the bogies underneath by means of low vertical rigidity of the springs, to improve ride comfort.

However, air springs may deflate if the air pipe is damaged in a collision with a motor vehicle, or the air spring may need to be deflated in a rescue operation as a safety measure against vertical buckling [1]. When deflated, the spring’s vertical rigidity is known to increase, making the spring less capable of following twists, and significantly reducing wheel load.

To examine the running safety of vehicles with deflated air springs, running tests were conducted on a vehicle with deflated air springs, in which measurements were taken of the wheel load, lateral force, forces acting on the air spring and wheel climb. Stationary tests were also conducted to measure the forces acting on the deflated air springs and, based on the obtained data, a simulation program involving a deflated air spring model was created. Simulation results were compared with the running test data to verify the validity of the model.

2. Running test

Running tests were conducted using a test vehicle owned by the Railway Technical Research Institute on a curved section of the test track at the institute.

2.1 Test vehicle

Figure 1 shows a rough representation of the test vehicle. RTR235 bolsterless bogies for conventional lines owned by the Railway Technical Research Institute were used in the test. The first wheelset of the vehicle was designed to measure wheel/rail contact force. In the running test, wheel load and lateral force were measured using the wheelset according to a new continuous measuring method. The force acting on the air springs was measured using three-component force sensors inserted beneath the laminated rubber. Wheel climb was measured by radiating laser beams from the laser displacement meters installed underneath by means of low vertical rigidity of the springs, to insulate car bodies against vibrations from the bogies un

Direction of measurement

Three-component force sensor
Measuring wheelset of wheel/rail contact force (New continuous measuring method)
Laser displacement meter (Amount of climbing of the wheel)

Fig. 1 Rough representation of the test vehicle

| Table 1 Key specifications of the test vehicle |
|----------------|----------------|
| **Mass (kg)** | **Car body** 19930 |
| **Bogie frame** | **1390** |
| **Wheelset** | **1240** |
| **Wheelbase (m)** | **2.1** |
| **Bogie center to center distance (m)** | **13.8** |
| **Height of car body’s center of gravity (m)** | **1.7** |
| **Rigidity of axle spring (MN/m/axle)** | **11.8** |
| **Lateral** | **10.4** |
| **Vertical** | **2.8** |
| **Rigidity of air spring (MN/m/spring)** | *** 0.11** |
| **Lateral** | *** 0.11** |
| **Vertical** | *** 0.43** |
| **Vertical rigidity of laminated rubber (MN/m/piece)** | **8.9** |

* Static value with the air spring in proper working condition and under a load of 7200 kg.

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on the axle box aimed at the top surface of the rails.

The key specifications of the test vehicle are listed in Table 1. The air spring’s vertical rigidity when deflated was 8.9 MN/m (which is in fact the vertical rigidity of the laminated rubber), about 20 times of the air spring’s vertical rigidity of 0.43 MN/m when in proper working condition. The test vehicle was left empty to become a higher decrease ratio of wheel load. The average wheel load was about 34 kN, which corresponds to that of a relatively light commercial vehicle when empty. The static wheel load imbalance of the first wheelset designed to measure wheel/rail contact force was 4.9 % when the air springs were in proper working condition and 4.8 % when deflated. In both cases, the wheel load on the outside rail was lighter than that on the inner rail.

### 2.2 RTRI test track

The curved section on the test track had a radius of 160 m, a cant of 90 mm, and a cant transition rate of about 1100 at the entrance and 400 at the exit. Figure 2 shows the 5-meter twist. The negative areas are where the outside wheel load of the first axle decreases. The test track had a heavy twist, due to either track irregularity or cant transition, at around the 260-meter point in the circular curved section and at around the 320-meter point in the transition curve section near the exit.

Due to the conditions described above, it can be said that the test curve presented tougher vehicle running conditions in terms of safety than a typical curve found on commercial lines.

**Fig. 2** 5-meter twist on RTRI test track

### 2.3 Test conditions

The running tests were carried out with the air springs in two modes: proper working mode and deflated mode. In both cases, the test vehicle was run at 10 km/h. The ratio of lateral force to wheel load for the inside rail/wheel, which corresponds to the coefficient of wheel/rail friction when running through a sharp curve, was 0.37 in both modes. The running tests were conducted on dry rails.

### 2.4 Test results

The results (in waveforms) of the running test in both proper working and deflated modes are shown together in Fig. 3. For ease of comparison, time-history data gathered at intervals of 1 ms was put through one-second moving average processing. The force acting vertically on the air spring shows variations from the reference value representing a stationary test vehicle. The moments producing rotational constraint of the bogie indicate moments around the bogie center produced by the force acting longitudinally on the air spring. Positive values indicate moments that act to increase the lateral force on the outside rail/wheel of the first wheelset.

**1. Wheel climb**

In proper working mode, the wheel did not show any tendency to climb the rail. In deflated mode however, the wheel climbed about 6 mm at around the 260-meter point in the circular curved section with a heavy twist, then fell, and climbed up to around 17 mm in the transition curve section near the exit. Based on the above, it is presumed that the wheel/rail contact point reached the flange straight section at around the 260-meter point and 14R on the flange tip in the transition curve section near the exit.

**2. Wheel load**

In deflated mode, a significant reduction in wheel load was observed in sections with a heavy twist, i.e. at around the 260-meter point and in the transition curve near the exit. That is because the vertical rigidity of the air spring increase when it is deflated, making the bogie less capable of following the twist. In the deflated mode, the wheel load decreased by a constant rate down to around 8 kN up to around 10 m into the transition curve near the exit. The wheel load then stabilized there while the wheel stayed in contact with the rail at the flange straight section before increasing further by about 3 kN when the contact point moved to 14R on the flange tip. Then, the wheel load decreased to nearly zero the moment the wheel started falling.

**3. Lateral force**

The lateral force showed similar behavior in both air spring modes. It is worth noting that the lateral force tended to be weaker in deflated mode than in proper working mode where the moments producing rotational constraint of the bogie were in the negative range. That presumably is because the negative moments acted to decrease the lateral force on the outside rail/wheel of the first wheelset.

**4. Derailment quotient**

Wheel load can fluctuate more where there is heavier twist. In those sections, the coefficient of derailment tends to be greater in the deflated mode than in the proper working mode. The derailment quotient in the deflated mode (after moving average processing) was around 2 at around the 255-meter point with heavy twist and around 2.5 in the transition curve section near the exit where the wheel climbed more than 10 mm. The maximum coefficient of derailment in the deflated mode was 4.3, which occurred when the wheel load decreased to nearly zero the moment the wheel started falling after climbing.

**5. Force acting on the air spring**

In proper working mode, the force acting vertically on the air spring decreased by as much as around 10 kN in the transition curve section near the exit. In deflated mode, the force decreased by around 22 kN near the 255-meter point and by around 30 kN in the transition curve section near the exit. In proper working mode, the moments producing rotational constraint of the bogie obtained from the force acting longitudinally on the air spring, formed a waveform that is nearly proportional to the curvature of
the track. In deflated mode, the moments steadied at nearly zero between fluctuations, some large, in the circular curve section. Those fluctuations all stayed below the moments observed in the proper working mode. The moments also stayed in the negative range in the transition curve section near the exit, indicating that the moments in the deflated mode did not act to increase lateral force as much as the moments in the proper working mode.

The force acting laterally on the air spring in the circular curve section is about 2 kN in the proper working mode and about 1 kN in the deflated mode. The force in the deflated mode was roughly the same as or less than that in the proper working mode.

The results of the running tests showed that, in the deflated mode, the force acting on the air spring fluctuated to a greater degree, causing the wheel load to decrease significantly and therefore more likely to generate higher derailment quotient and greater wheel climb than in the proper working mode. On the other hand, no significant increase was observed in moments producing rotational constraint of the bogie and lateral force in the deflated mode over the proper working mode.

3. Stationary tests

3.1 Equipment measuring rotational constraints of bogies

Stationary tests were conducted to clarify the forces acting on deflated air springs and to develop a simulation model. Figure 4 shows a device, owned by RTRI, designed to measure the rotational constraint of a bogie [2]. This equipment measures the rotational constraints on the bogie by loading the car body with the air spring, damper and towing devices installed. In the stationary test, the following parameters were measured: force acting on the air spring, air spring displacement (relative displacement between the car body and bogie measured at the air spring) and rotation angle of the rotation table.

3.2 Test results

Figure 5 shows time-history waveforms of the force acting longitudinally on the air spring when the bogie was rotated with an amplitude of 1 degree and those when the bogie was rotated with an amplitude of 4.5 degrees, both with a frequency of 0.1 Hz. The relationship between rotation angle (bogie angle) and curve radius, which can be geometrically obtained, for a vehicle with a bogie center to center distance of 13.8 m is as follows: a rotation angle of 1 degree corresponding to a curve radius of 400 m, and likewise 4.5 degrees to 90 m.

Figure 5 shows that in the deflated mode the force acting longitudinally on the air spring increases much more rapidly than in the proper working mode and stays relatively flat at certain levels. The steep increase in longitudinal force is due to the car body and bogie being supported by the laminated rubber in deflated mode, whose longitudinal rigidity is 4 to 6 times that of the air spring in proper working mode.

The longitudinal force stayed relatively flat, which indicates that the shear force of the laminated rubber exceeded the maximum static friction force of the sliding part of the air spring, causing the part to slide. This means that in deflated mode the longitudinal force is greater than in the proper working mode when the rotation angle is small, and that the difference in the longitudinal force between these modes diminishes when the sliding part slides.

To simulate the characteristics described above, it was necessary to model a deflated air spring factoring in non-linearity.
4. Creation of deflated air spring model

4.1 Alphanumeric codes used in this section

The key alphanumeric codes used in this section are listed below.

- **F**: Force acting on the air spring
- **k**: Vertical or horizontal rigidity of the laminated rubber
- **μ**: Coefficient of static or dynamic friction of the sliding part
- **δ**: Vertical or horizontal flexure of the laminated rubber
- **x**: X or Y coordinate of the top face of the laminated rubber
- **y**: X or Y coordinate of the top plate of the air spring
- **x₀, y₀**: Differences in X or Y coordinate between the air spring top plate and laminated rubber top face
- **x₁₀, y₁₀**: Coefficient of static or dynamic friction of the sliding part
- **δ₀**: Vertical or horizontal flexure of the laminated rubber prior to the last calculation

4.2 Deflated air spring model

Figure 6 shows the deflated air spring model. With this model, the car body is supported vertically just by the vertical rigidity of the laminated rubber. Horizontally, the model consists of the horizontal rigidity of the laminated rubber and the sliding part inside the air spring. With the origin set at the center of the bottom face of the laminated rubber, the force acting vertically and that horizontally on the air spring can be calculated as follows.

(1) Vertical force

The force \( F_{g2} \) acting vertically on the air spring can be obtained with (1) using the vertical flexure of the laminated rubber.

\[
F_{g2} = k_{g2} \delta_2
\]  

(2) Horizontal force

The force acting horizontally on the air spring can be calculated using the flow in Fig. 7. The calculation starts with the sliding part in a no-sliding state and the initial values of \( x_0 \) and \( y_0 \) both set to zero.

To start with, maximum static friction and the laminated rubber’s shear force are calculated in Step (a) of Fig. 7 using (2) and (3) respectively.

\[
F_1 = \mu_1 F_{g2}
\]

\[
F_{g1} = k_{g1} \delta_1
\]

Where, \( \delta_1 \) can be obtained based on the X and Y coordinates on the top face of the laminated rubber. Using the known \( x_0, y_0, x_1, y_1 \), (4) can be used.

\[
\delta_1 = \sqrt{x_{10}^2 + y_{10}^2} = \sqrt{(x_0 - x_{10})^2 + (y_0 - y_{10})^2}
\]

When the shear force of the laminated rubber is smaller than the maximum static friction, no sliding occurs and the laminated rubber’s shear force represents the force \( F_{g1} \) acting on the air spring ((c)). In addition, the air spring top plate and the laminated rubber top face move together as a unit, \( x_0 \) and \( y_0 \) stay unchanged and the calculation advances to the next step.

On the other hand, when the laminated rubber shear force exceeds the maximum static friction, the sliding part slides and the force \( F_{g1} \) acting on the air spring is equal to the dynamic friction, which can be calculated using (5) ((e)).

\[
F_2 = \mu_2 F_{g2}
\]

As the air spring top plate and the laminated rubber top face slide, \( x_0 \) and \( y_0 \) can be calculated using (6).

\[
x_0 = x_0 - F_{g2} / k_{g1}
\]

\[
y_0 = y_0 - F_{g2} / k_{g1}
\]
following judgment: sliding when the horizontal flexure $\delta_{H}$ of the laminated rubber increases and not sliding when it decreases.

4.3 Verification of the simulation model

The specifications of a deflated air spring identified from the results of the stationary test are shown in Table 2. The vertical rigidity of the laminated rubber was taken directly from the relevant drawing. As an example of the comparison of results from a simulation conducted with those values and actual test results, Fig. 8 shows the waveform of the longitudinal force on the air spring.

This comparison indicates that simulation results represent appropriately the rapid change in longitudinal force on the air spring that occurred as the bogie started to rotate and change the direction of rotation, as well as the characteristic of the longitudinal force to gradually equal the dynamic friction. Hence, the model can be considered valid.

| Table 2 Specifications of a deflated air spring |
|------------------------------------------------|
| Specified parameter                          | Value  |
| Horizontal rigidity of laminated rubber      | 0.6 MN/m |
| Vertical rigidity of laminated rubber        | 8.9 MN/m |

| Table 8 Comparison of test and numerical simulation results |
|----------------------------------------------------------|
| Specification                                           | Test    | Simulation |
| Wheel load                                             | 40 kN   | 35 kN      |
| Lateral force                                           | 10 kN   | 5 kN       |
| Vertical force on the wheel                             | 20 kN   | 15 kN      |

5. Curving simulation

Using the deflated air spring model, discussed in the previous section, a curving simulation was conducted under the running test conditions described in Section 2.

5.1 Numerical model

The analytical model used in the simulation is shown in Fig. 9. It was a single-car model with 38 degrees of freedom. Inputs from the rails, longitudinal level irregularity, alignment and irregularity of cross level were factored in. Levi-Chartet nonlinear characteristics were applied to the creep force acting on the wheel/rail.

5.2 Simulation results

Figure 10 shows the results of the simulation together with the test results.

(1) Wheel climb
The simulated wheel climb agreed with the test results fairly accurately both in terms of timing and amount at around the 255-meter point and in the transition curve section near the exit.

(2) Wheel load, lateral force and derailment quotient
The simulated wheel load agreed with the test results fairly accurately including fluctuation caused by twists and cant transitions.

The simulated lateral force was a difference of nearly 5
kN with the test results from about the 150-meter point to about the 250-meter point. The difference shrank after the 250-meter point, and the simulated lateral force was considered acceptable overall.

The simulated coefficient of derailment failed to agree with the test results where the parameter in the test exceeded 2. This was probably due to the fact that even a slight difference can have a significant impact when wheel load, the denominator in the calculations, is extremely small. Other than that, the analytical model simulated the test results satisfactorily, including the timing where the derailment quotient started to increase.

(3) Forces acting on the air spring

As with the wheel load, the simulated vertical force on the air spring agreed with the test results fairly accurately including fluctuation caused by twists and cant transitions.

As for the moments producing rotational constraints on the bogie, the simulated moments steadied at levels about 3 kNm away from the levels observed in the test. Other than that, the simulated results reproduced satisfactorily the characteristics of the deflated mode including large fluctuation.

The simulated lateral force on the air spring was about 1 kN off from the test results, which can be considered mostly consistent with the test results.

The above shows that the air spring model was largely capable of simulating the characteristics of the deflated air spring mode and can therefore be considered a valid simulation tool to evaluate vehicle safety when running through curves.

6. Conclusion

Running tests were conducted with air springs in deflated mode to evaluate operational safety. A simulation program for running in curves, including a deflated air spring model was developed, providing the following findings:

(1) The running test conducted with deflated air springs on an actual vehicle reduced wheel load more than when the air springs were operating properly, and showed that the greater the reduction in wheel load, the greater the likelihood that the derailment coefficient and wheel climb would increase.

(2) A comparison of simulation results with test results validated the curve simulation method, using a deflated air spring model. The air spring model was largely capable of simulating the characteristics of the deflated air spring mode and can therefore be considered a valid simulation tool to evaluate vehicle safety when running through curves.

References

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