A Potential of Rail Vehicle Having Bolster with Side Bearers for Improving Curving Performance on Sharp Curves Employing Link-Type Forced Steering Mechanism*

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Abstract
The air spring of bolsterless bogie trucks, which have been widely employed in railway vehicles in recent years, undergoes a large distortion when the vehicles negotiate sharp curves in lines such as subway lines, and this can deteriorate the durability of air springs. Furthermore, bolsterless trucks tend to suffer from increased wheel lateral force around sharp curves with a radius of 100 m or less. In this paper we discuss the application of a link-type forced steering mechanism to bolster trucks with a bolster as a countermeasure against the above-mentioned situation. A numerical simulation is carried out using a MBS software, SIMPACK.
As a result, under the condition of reduced longitudinal stiffness in the primary suspension, a bolster truck with the link-type steering mechanism exhibits the potential to suppress the wheel lateral force occurring around sharp curves. Also, the deterioration in running stability due to the application of the steering mechanism can be recovered by adding moderate lateral damping in the secondary suspension. In addition, the obtained wear index shows that the forced steering truck has decreased flange wear resulting from passing through sharp curves.

Key words: Transport, Railway, Simulation, Curve Negotiation, Forced Steering Truck

1. Introduction
There are sharp curves with a radius of 160 m or less in conventional narrow-gauge lines, particularly in subway lines. When a railway vehicle negotiates such a sharp curve, a large lateral force acts between the wheel and the rail, which accelerates the wear on the wheel flange and on the side of the rail. In addition, in the case of vehicles with bolsterless trucks, the durability of the air spring deteriorates owing to the large longitudinal distortion caused by the increased bogie angle in sharp curves.

As a measure to reduce the lateral force during the negotiation of a curve, a forced steering mechanism has been employed [1 - 6]. However, it has been reported that for a vehicle with bolsterless trucks the link-type forced steering mechanism increases the wheel lateral force in sharp curves with a radius of 100 m or less because the wheel is strongly pressed onto the rail by the steering mechanism [2, 4]. On the other hand, a vehicle with side-bearer-type bolster trucks causes the trucks to turn through the slippage of the side bearers as a result of the lateral force induced where the wheel flange comes in contact with the rail in entrance transition curves. Owing to the slippage of the side bearers, the large
distortion of air springs can be avoided because the bolster rotates little relative to the car body, although the friction occurring in the side bearers may increase the lateral force slightly. In addition, the wheel lateral force is reduced in circular curves since the restoring moment arising from the distorted air springs decreases.

As mentioned above, it is expected that the large distortion of the air springs can be avoided, thus reducing the wheel lateral force in circular curves, by adding a forced steering mechanism to a bolster-type vehicle. In this paper, we demonstrate the suitability of a bolster-type vehicle for negotiating sharp curves by a numerical simulation using the MBS software SIMPACK. We also examine the possibility that a link-type forced steering mechanism can prevent an increase in the wheel lateral force around sharp curves [7].

2. Comparison between Truck Structures

2.1 Analytical vehicle model

An analytical model of a bolster-type vehicle running on a conventional narrow-gauge line [8] is developed using SIMPACK, which has a specialized module for the analysis of railway systems, allowing us to readily perform a valid calculation on the contact force between the wheel and the rail [9]. The vehicle model has a total of 36 degrees of freedom.

The vehicle with bolsterless trucks, which is used for comparison, is modeled by uniting the bolster with the truck frame as shown in Fig. 1. Also, the bolster anchor of the side-bearer-type truck is replaced with a traction device having an equivalent longitudinal stiffness and damping coefficient. Thus, the vehicle with bolsterless trucks has 34 degrees of freedom. The curving performances for each truck structure are compared using the two vehicle models.

![Fig. 1 Modeling of bolsterless truck using a truck model with a bolster and side bearings](image)

<table>
<thead>
<tr>
<th>Curve radius $R$ [m]</th>
<th>Speed $V$ [km/h]</th>
<th>Cant elevation $C$ [mm]</th>
<th>Transition curve length $X_{tr}$ [m]</th>
<th>Slack $S$ [mm]</th>
</tr>
</thead>
<tbody>
<tr>
<td>400</td>
<td>75</td>
<td>85</td>
<td>85</td>
<td>5</td>
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<td>15</td>
</tr>
<tr>
<td>100</td>
<td>30</td>
<td>55</td>
<td>55</td>
<td>20</td>
</tr>
</tbody>
</table>

2.2 Comparison between curving performance of truck structures

A curving simulation is carried out using SIMPACK for a bolster-type vehicle and a vehicle with bolsterless trucks, and the behaviors in a circular curve are compared. It was expected that the wheel lateral force would become small for the bolster-type vehicle. Curves with radii from 100 m to 400 m were investigated and the track conditions are listed
in Table 1. Here, \( V \) is the running speed, \( C \) is the cant, \( X_{TC} \) is the length of the transition curve and \( S \) is the slack. The values in Table 1 are regarded as typical for conventional lines.

Figure 2(a) shows the simulated longitudinal distortion of the air spring in the front truck, \( x_{AS} \), as the truck negotiates curves, and Fig. 2(b) shows the wheel lateral force acting on the outer side of the leading wheelset, \( Q_1 \). The displacement, \( x_{AS} \), obtained for the bolster-type vehicle is the relative displacement between the bolster and the car body, whereas it is the relative displacement between the truck frame and the car body for the vehicle with the bolsterless trucks. Here, it is assumed that the displacements on the left and right sides are identical. Figure 2(a) shows that the distortion of the air spring in the vehicle with the bolsterless trucks increases when the curve becomes sharper, while that in the bolster-type vehicle remains small. Furthermore, Fig. 2(b) shows that the wheel lateral force, \( Q_1 \), of the vehicle with the bolsterless trucks is greater than that of the bolster-type vehicle on a sharp curve of 100 m radius, although it is smaller on curves with radii of 200 m or more. These results show the superior curving performance of the bolster-type vehicle around curves with a radius of 100 m or less.

![Comparison between curving performance of truck structures](image)

3. Addition of Steering Mechanism

3.1 Steering mechanism

In this paper, as shown in Fig. 3(a), a link-type forced steering mechanism based on the bogie angle is used [10]. Figure 3(b) shows that the bolster and the axle box are connected to the steering lever, which rotates around a pin on the truck frame, via spring elements. When the vehicle negotiates a curve, the bogie angle generated by the turning of the truck causes the steering lever to rotate, which forces the front and rear wheelsets to turn in the opposite direction to each radial position. This allows the wheelset to decrease its attack angle and reduces the wheel lateral force during curving. The front and rear trucks each have two steering levers, each with one degree of freedom rotation; thus, the total number of degrees of freedom in the bolster-type vehicle with the steering mechanism is 40 because of the 4 degrees of freedom of the steering levers.

3.2 Conditions set in simulation

Table 2 shows the four conditions on the steering mechanism added to the bolster-type vehicle. Here, in addition to evaluating the steering performance with or without the steering mechanism, the effect of lowering the longitudinal support stiffness of the axle box, \( k_{wv} \), is compared, where the stiffness is reduced from the baseline value by 70%.

The friction coefficient, \( \mu \), between the wheel and the rail is varied around the baseline value of \( \mu = 0.3 \) in the range from 0.1 to 0.55, considering that it changes greatly depending...
on the conditions of the rail surface and the wheel tread. The curves considered have radii $R = 100 \text{ m} – 400 \text{ m}$ with the parameter values given in Table 1. Figure 4 shows an example of simulation results for the vehicle condition B0. The figure shows the waveform of the wheel lateral force acting on the outer leading wheel as it runs around a curve of $R = 300 \text{ m}$ with $\mu = 0.3$. The comparison of the steering effect under various vehicle conditions is performed by considering the wheel lateral force, which becomes constant as the vehicle enters a circular curve as shown in the figure.

Table 2  Truck condition for simulation

<table>
<thead>
<tr>
<th>Condition</th>
<th>Steering link</th>
<th>Stiffness $k_{w\alpha}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>A0</td>
<td>with</td>
<td>baseline</td>
</tr>
<tr>
<td>A1</td>
<td>with</td>
<td>decreased to 30 %</td>
</tr>
<tr>
<td>B0</td>
<td>without</td>
<td>baseline</td>
</tr>
<tr>
<td>B1</td>
<td>without</td>
<td>decreased to 30 %</td>
</tr>
</tbody>
</table>

Fig. 4  Example waveform of wheel lateral force
4. Running Simulation

4.1 Curving performance

Figure 5 shows a comparison between wheel lateral forces, $Q_1$, obtained from the simulation by varying the curve radius $R$. In the case of $R = 300$ m, shown in Fig. 5(a), the wheel lateral force is decreased upon adding the steering mechanism when the friction coefficient $\mu$ is higher than the baseline value of 0.3, that is, by changing the vehicle condition from B0 to A0, and from B1 to A1. The effect on the reduction of $Q_1$ is particularly large in the case of vehicle condition A1, in which the longitudinal support stiffness of the axle box, $k_{wx}$, is lower. On the other hand, $Q_1$ remains about the same magnitude in the case of the low friction coefficient, $\mu = 0.1$, even though the steering mechanism is added and the support stiffness $k_{wx}$ is lower.

In the case of $R = 200$ m, shown in Fig. 5(b), the effect of the steering mechanism on the reduction of $Q_1$ is small for vehicle condition A0, in which the support stiffness $k_{wx}$ remains unchanged from the baseline value. In this case, $Q_1$ increases to more than that for vehicle condition B0, which has no steering mechanism, when the friction coefficient is decreased. However, vehicle condition A1, in which the steering mechanism is added and the support stiffness is lower, still has the effect of reducing $Q_1$.

When the curve radius decreases to $R = 100$ m, as shown in Fig. 5(c), $Q_1$ for the vehicle conditions with the steering mechanism (A0 and A1) becomes larger than that for the vehicle conditions without the steering mechanism (B0 and B1) when the friction coefficient is less or equal to 0.3. However, the wheel lateral force for vehicle condition A1, in which the support stiffness $k_{wx}$ is lower, is about the same magnitude as that for condition B1 when $\mu$ has the baseline value of 0.3, and still remains about 20% higher than that for vehicle condition B1 when $\mu$ is decreased to 0.1. In contrast, when $\mu$ is increased to 0.55 the wheel lateral force decreases the most for vehicle condition A1. In addition, there is little change in $Q_1$ for conditions B0 and B1, which have no steering mechanism, even though the support stiffness $k_{wx}$ is lower.
Consequently, it is suggested that for the bolster-type vehicle \( Q_1 \) does not increase even in a sharp curve with radius 100 m unless \( \mu \) decreases greatly when the longitudinal support stiffness, \( k_{\text{ax}} \), is lowered and the link-type forced steering mechanism is added.

### 4.2 Effect of leverage ratio

The steering leverage ratio \( L_2/L_1 \) is the ratio of the lower length \( L_1 \) to the upper length \( L_2 \) of the steering lever as shown in Fig. 6. \( L_{10} \), the baseline value of which is \( L_{10} = 0.046 \) m, is the length from the centers of rotation of the steering lever to the position where it is connected with the wheelset. \( L_2 \), the baseline value of which is \( L_{20} = 0.3 \) m, is the length from the center of rotation to the position where it is connected with the bolster. Considering the spatial limitation, it is assumed that the steering leverage ratio varies in the range from 0.6 to 1.4 times the baseline value \( L_{20}/L_{10} \) by the scaling factor \( \alpha \). In the examination of the leverage ratio, only vehicle condition A1 is considered, because under this condition the leverage ratio has a large effect on the steering performance.

![Fig. 6 Leverage length of steering mechanism](image1)

![Fig. 7 Effect of leverage ratio (truck A1)](image2)

Figure 7 shows a comparison of \( Q_1 \) in a circular curve of 100 m radius, that is, in the steady state, with the friction coefficient \( \mu \) between the wheel and the rail varied as a parameter. When \( \mu \) is less than the baseline value of 0.3, \( Q_1 \) varies slightly with changing leverage ratio. Also \( Q_1 \) tends to increase with increasing \( \mu \). This is because the spring force required to steer the wheelset increases with the leverage ratio; thus its reaction force acts in the opposite direction, inhibiting the rotation of the truck. On the other hand, when \( \mu \) is 0.55, \( Q_1 \) is decreased and a steering effect is obtained with increasing leverage ratio. When \( \alpha \) is 1.4, however, \( Q_1 \) starts to increase.

Figure 8 shows the behavior of the front wheelset (the first axle) and the rear wheelset (the second axle) in the case of \( \mu = 0.55 \) shown in Fig. 7. The wheel lateral forces on the outer and inner sides of the rail in each axle are shown in Fig. 8(a), the lateral displacements of the front and rear axles are shown in Fig. 8(b), and the attack angles of the truck frame and both axles are shown in Fig. 8(c). For the wheel lateral forces shown in Fig. 8(a), positive values mean that the force acts from the outer side of the rail to the inner side, and vice versa for negative values. The subscripts of \( Q \) denote the axle number (1 or 2) and the wheel position (I for the inner rail side or O for the outer side). The lateral force \( Q_{1I} \) of the outer wheel on the first axle is always positive for all values of \( \alpha \), which means that the wheel is subjected to a reaction force acting from the outer rail at the flange contact. This corresponds to wheelset behavior in which the lateral displacement \( y_{w1} \) of the first axle reaches the flange clearance \( \delta \) of about 10 mm.

In contrast, the lateral displacement \( y_{w2} \) of the second axle remains within the flange clearance, and the wheel lateral forces of both the inner and outer sides of the rail are negative. Here, a lateral creep force acts toward the outer side of the rail. The difference between the wheel lateral force acting on the inner and outer wheels depends on the
difference between the gravitational forces acting on the contact points of each wheel tread. Observing the attack angle $\psi_{w1}$ of the front wheelset in Fig. 8(c), it can be seen that the marked reduction of $\psi_{w1}$ resulting from the increase in $\alpha$ to 1.2 leads to a decrease in the wheel lateral force $Q_{10}$ in Fig. 8(a). When $\alpha$ reaches 1.4, $\psi_{w1}$ becomes positive and the wheelset is in the state of oversteering, then the wheel lateral force $Q_{10}$ starts to increase since the lateral creep force acts toward the inner side of the rail. For the second axle, it is confirmed that the lateral creep force, which acts toward the outer side of the rail, increases with an excess leverage ratio. As a result, when the attack angle $\psi_{w2}$ increases the wheel lateral force of the inner wheel $Q_{2I}$ increases markedly.

Consequently, it can be concluded that an increase in the leverage ratio does not cause a reduction of the wheel lateral force when the friction coefficient $\mu$ between the wheel and the rail is 0.3 or less, although a reduction in the wheel lateral force due to an increased leverage ratio can be anticipated when $\mu$ is large.

4.3 Wear index

There is considerable wear of the wheel at the flange when a vehicle negotiates a sharp curve. Figure 9 shows a simulated waveform for the lateral displacement $y_{w1}$ of the first wheelset for vehicle condition A1 with $\mu = 0.3$. It is shown that the wheelset moves by about 10 mm toward the outer rail side (i.e. a value of -10 mm was obtained) and that the wheelset is in balance condition with the rail reaction force when it enters the transition curve. In this situation, the flange maintains contact with the rail, and wear on the wheel flange and the side of the rail occurs.

The curving performance is compared for a sharp curve of radius $R = 100$ m using the wear index $W_{IF}$ proposed by Heumann [11] given by

$$W_{IF} = \mu F_y \psi$$

(1)
Here, $\mu_F$ is the friction coefficient where the flange is in contact with the rail, $F_F$ is the flange reaction force, and $\psi$ is the attack angle of the wheelset. Vehicle steering performance is compared for condition A1, which has the steering mechanism and lowered longitudinal support stiffness $k_{wx}$ of the axle box, and condition B0, where the support stiffness $k_{wx}$ has the baseline value and there is no steering mechanism. Here, we set the baseline friction coefficient of $\mu = 0.3$ for vehicle condition A1, and $\mu = 0.1$ for condition B0 assuming the presence of a friction modifier which has been employed to reduce the wheel lateral force around sharp curves [12].

Simulated waveforms of the flange reaction force $F_{F1}$, attack angle $\psi_{w1}$ and wear index $W_{IF}$ of the first wheelset are shown in Fig. 10. It is shown in Figs. 10(a) and 10(b) that $\psi_{w1}$ of vehicle condition A1 is less than that for vehicle condition B0 because of the steering mechanism, although the flange reaction forces, $F_{F1}$, for conditions A1 and B0 have almost the same magnitude. As a result, the wear index $W_{IF}$ for condition A1 is smaller than that for condition B0 in the sharp curve and the exit transition curve as shown in Fig. 10(c). Thus, less flange wear is predicted for condition A1.
Next, the wear index integrated over the whole curved section including the transition curves is compared, as shown in Fig. 11. The integrated wear index ($W_1$)$_{SUM}$ for condition A1 with the steering mechanism remains at a low level, while that for condition B0 increases with $\mu$. Therefore, from the viewpoint of the wear index, the steering mechanism is effective for reducing the wear of the wheel flange; although it was shown above that the steering mechanism has a limited effect on reducing the wheel lateral force in a sharp curve.

4.4 Running stability

For a vehicle operating under condition A1 with the lowered longitudinal support stiffness of the axle box, $k_w$, it is concerned that the running stability may deteriorate because of the reduced restriction of the wheelset motion and also because of the effect of the steering mechanism itself. Thus, the convergence property of the car-body lateral vibration is examined by assuming a step-shaped alignment irregularity of 5 mm on a tangent track. The simulation results are compared in Fig. 12. $\mu$ is assumed to be 0.3, and the target speed is set to $V = 180$ km/h, considering the maximum design speed for a commuter train with some leeway.

![Fig. 11](image1.png)  Comparison of integrated wear index

![Fig. 12](image2.png)  Running stability in each condition
Figure 12(a) shows the result for condition B0 at a speed of 180 km/h, Fig. 12(b) shows the result for condition A1 at a speed of 135 km/h and Fig. 12(c) shows the result for condition A1 at a speed of 180 km/h. The waveforms shown are for car-body lateral displacements $y_{BF}$ and $y_{BR}$ on the floor above the front and rear truck center positions, respectively. The lateral vibration for condition B0 is stable and converges at the target speed of 180 km/h as shown in Fig. 12(a). On the other hand, the lateral vibration for condition A1 does not converge, even at 135 km/h, and becomes a lasting vibration as shown in Fig. 12(b). Also, the vibration grows with increasing running speed and diverges at 180 km/h as shown in Fig. 12(c).

Figure 13 shows the car-body lateral vibration of the vehicle with added lateral dampers running at the target speed of 180 km/h. The lateral vibration converges rapidly at the target speed, and it is confirmed that the running stability can be improved by adding lateral dampers with a relatively small damping coefficient. Here, the effect of adding the lateral dampers on the curving performance is small.

Accordingly, it is shown that the addition of lateral dampers is effective for improving the running stability, which may deteriorate in a vehicle with a steering mechanism and lowered longitudinal support stiffness of the axle box.

5. Conclusions

We examined the curving performance of a bolster-type vehicle with side bearers having a link-type forced steering mechanism around sharp curves with a radius of 100 m, similar to those found in subway lines.

(1) The bolster-type vehicle with an added steering mechanism can suppress the increase in the wheel lateral force occurring around a sharp curve when the longitudinal support stiffness of the axle box is reduced. When the friction coefficient between the wheel and the rail increases, it is even possible to reduce the wheel lateral force.

(2) When the friction coefficient between the wheel and the rail is high, the wheel lateral force occurring around sharp curves can be reduced by increasing the leverage ratio. However, it is possible that the large leverage ratio may increase the wheel lateral force when the friction coefficient is $\mu = 0.3$, which is the baseline value assumed in this study.

(3) From the viewpoint of the wear index on the wheel flange, it is expected that the flange wear can be suppressed by the addition of the steering mechanism through the reduction of the attack angle as the curve is negotiated.

(4) For a vehicle with the link-type steering mechanism and a low longitudinal support stiffness of the axle box, it is possible to ensure running stability up to the target speed by adding lateral dampers with a relatively small damping coefficient.

As mentioned above, it is expected that the large distortion of air springs generated in the vehicle with bolsterless trucks can be eliminated by employing a bolster-type vehicle, and that the flange wear can be decreased for bolster-type vehicles.
Appendix

The components of a vehicle and their respective degrees of freedom are shown in Fig. 14, along with an outline drawing of the analytical model.

The parameter values used in the calculation are those assumed for a subway vehicle used on conventional narrow-gauge lines.

Mass

Car-body: 14.62 t, Truck frame: 1.35 t, Wheelset: 0.85 t

Moment of inertia

Car-body roll: 2.04 m, Car-body pitch: 7.27 m, Car-body yaw: 5.60 m
Truck frame roll: 0.50 m, Truck frame pitch: 0.24 m, Truck frame yaw: 0.50 m
Wheelset roll: 0.50 m, Wheelset pitch: 0.24 m, Wheelset yaw: 0.50 m

Longitudinal distance between

Axles in truck: 2.1 m, Truck centers: 13.8 m

Lateral distance between

Axle springs: 1.6 m, Air springs: 2.0 m, Side bearers: 2.0 m, Bolster anchors: 2.45 m

Height from rail top

Center of gravity of truck frame: 0.72 m, Center of gravity of car-body: 1.97 m
Air spring center: 0.85 m, Bolster anchor: 0.55 m, Car-body floor: 1.175 m

Primary suspension

Lateral stiffness per axle: 9 800 kN/m, Vertical stiffness per axle: 2 440 kN/m
Longitudinal stiffness per axle box $k_{xx}$: 7 350 kN/m (baseline value) 2 000 kN/m (lowered value)

Lateral damping per axle: 9.8 kNs/m, Longitudinal damping per axle: 9.8 kNs/m

Secondary suspension

Lateral stiffness of air spring: 200 kN/m, Vertical stiffness of air spring: 185 kN/m
Longitudinal stiffness of air spring: 200 kN/m, Vertical damping of air spring: 20 kN s/m

Bolster anchor
Longitudinal stiffness: 2 250 kN/m, Longitudinal damping: 5 kN s/m
Friction coefficient of side bearer: 0.20

Wheel
Rolling radius: 0.43 m, Tread profile: modified arc
Rail profile: 50 kg-N

Steering lever
Height of rotation center from rail top: 0.43 m, Lateral distance: 2.0 m
Length of upper lever: 0.046 m (baseline), Length of lower lever: 0.3 m (baseline)
Upper lever stiffness: 50 000 kN/m, Upper lever damping: 5 kN s/m
Lower lever stiffness: 50 000 kN/m, Lower lever damping: 5 kN s/m

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