Effect of Crosswind on Derailment of Railway Vehicles Running on Curved Track at Low Speed

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Abstract

Owing to the lightening of railway vehicles and increased operation speeds, the reduction of running safety in the presence of crosswind is becoming an important problem. In particular, the running safety tends to decrease when vehicles run on curved track. When a crosswind acts on a vehicle negotiating a curve from the outer side, flange climbing can occur. In this study, a full-vehicle model was constructed using the multi-body simulation software SIMPACK, and a simulation of a bogie vehicle with two-axle trucks negotiating a curve was carried out to examine the running safety under the condition where a crosswind acts on the vehicle from the outer side of the curve. As a result, it was verified that the derailment coefficient of the first wheelset becomes large in the exit transition curve and the coefficient of the third wheelset does in the entrance transition curve, and this trend becomes pronounced at low operation speeds in the presence of a stronger crosswind. It was also shown that the critical derailment coefficients obtained by modified Nadal’s formula considering the effect of attack angle become close to the actual derailment coefficients at the timing that flange climbing occurs.

Keywords: Railway, Derailment, Crosswind, Modified nadal’s formula, Multi-body simulation software

1. Introduction

Owing to the lightening of railway vehicles and increased operation speeds, the reduction of running safety in the presence of crosswind must be considered. In particular, the running safety is strongly affected by crosswind through the interaction of the centrifugal force and the gravitational restoring force by cant when vehicles run on curved track. Regarding the analysis on overturn caused by a crosswind, several studies [1-7] have been published, whereas there have been few studies on derailment by a crosswind.

A train overturning accident, in which 5 passenger were killed, occurred on the JR Uetsu line in 2005. The cause of this accident is considered to be a strong crosswind [8]. To prevent such accidents amid strong winds, the operation speed is lessened by regulations depending on the wind speed. On the other hand, a crosswind may contribute to derailment when a train negotiates curved track under the lessened operation speed [9], since the running speed becomes lower than the balanced speed. The derailment is caused by a crosswind blowing from the outer side of the curve, and the wind speed is lower than that expected to cause overturn. In Ref. [9], however, effects of the factors which cause derailment are not examined sufficiently. Thus it is desirable to clarify the effects of crosswind on the derailment on curved track.

In this study, a full vehicle model is developed using the multi-body simulation software SIMPACK [10], and a simulation is carried out on a vehicle running on a curved section, subjected to a crosswind from the outer side of the curve. Here, the vehicle behavior is confirmed in the output animation, and the flange climbing behavior is analyzed using the output data such as wheel lateral force \( Q \) and wheel load \( P \) and so on. The derailment coefficients, \( Q/P \), are calculated and compared with Nadal’s critical derailment coefficient [11] employing an equivalent frictional feature, which is expressed as a non-linear function of the attack angle [12]. The result shows that the critical...
derailment coefficients employing the equivalent frictional feature, \((Q/P)_{\text{cri}}\), become close to the actual derailment coefficients, \(Q/P\), at the timing that flange climbing occurs.

**Notation**

The main symbols used in this paper are denoted as follows.

- \(v\): Operation speed
- \(w\): Natural wind speed
- \(u\): Relative wind speed
- \(\alpha\): Direction angle of natural wind
- \(\beta\): Direction angle of relative wind
- \(F_S\): Side force
- \(F_L\): Lift force
- \(C_S\): Side force coefficient
- \(C_L\): Lift force coefficient
- \(\rho\): Air density
- \(S\): Side area of car-body
- \(F_M\): Force acting on side of car-body
- \(F_F\): Force acting on front of car-body
- \(P\): Wheel load
- \(Q\): Lateral force
- \(Q/P\): Derailment coefficient
- \((Q/P)_{\text{Nadal}}\): Critical derailment coefficient obtained by Nadal’s formula
- \(\gamma\): Flange angle
- \(\mu\): Friction coefficient
- \(N\): Normal Force acting on contact patch
- \((Q/P)_{\text{cri}}\): Critical derailment coefficient obtained by using frictional feature \((f_y/N)\)
- \(f_y\): Lateral creep force
- \(\psi\): Attack angle
- \(\kappa\): Lateral creep coefficient
- \(\varepsilon\): Index indicating saturation property of creep force
- \(\phi\): Roll angle of wheelset

2. Full-Vehicle Model

In this study, the running behavior of a train is examined by numerical simulation. Here, the multi-body simulation (MBS) software SIMPACK [10], which has a module for railway system dynamics and allows realistic simulations. A full-vehicle model is developed as shown in Figs. 1 and 2.

It is assumed that this vehicle model has two bolsterless trucks and each truck has two wheelsets. The full-vehicle model has 42 degrees of freedom in total, since each body such as the car-body, two trucks and four wheelsets has six degrees of freedom, respectively; they are longitudinal \(x\), lateral \(y\), vertical \(z\), roll \(\phi\), pitch \(\theta\) and yaw \(\psi\). To prevent the excess relative displacements between the car-body and the trucks, stoppers are set for the lateral and vertical directions in preparation for motions when overturn occurs. Fig. 3 shows a schematic diagram illustrating the structure of the stoppers. When the relative displacements between the car-body and the trucks exceed a movable space, the reaction force of stopper acts between them and the excess motion is restrained (see section 4.1 for details). The stopper is set on the center of each truck for lateral motion, and on both sides of each truck for vertical motion.
3. Modeling of Wind Force

3.1 Aerodynamic force acting on train [3]

When a natural wind blows at speed $w$ in a direction of angle $\alpha$ and a train runs at operation speed $v$, a relative wind acts on the vehicle at speed $u$ in a direction of angle $\beta$ as shown in Fig. 4. From the relation of the speed vectors, speed $u$ and angle $\beta$ are given by

\[ u = \sqrt{w^2 + v^2 + 2wv\cos \alpha} \]  
\[ \beta = \cos^{-1}\left(\frac{w\cos \alpha + v}{u}\right) \]

In general, aerodynamic force is represented as a side force $F_S$ and a lift force $F_L$ by

\[ F_S = \frac{1}{2} \rho u^2 S C_S \]  
\[ F_L = \frac{1}{2} \rho u^2 S C_L \]

where, $\rho$ is the air density, $S$ is the side area of the car-body. $C_S$ and $C_L$ represent the side force coefficient and the lift force coefficient, respectively, and they depend on $\beta$.

$F_S$ acts laterally on the car-body side at the height of side surface center, and $F_L$ acts upward on the center of the car-body bottom.

Fig. 5 shows the aerodynamic force coefficients, $C_S$ and $C_L$, quoted from Ref. [13]. These coefficients were obtained in a wind tunnel experiment with a uniform flow. Here, suffix 1 attached to $C_S$ and $C_L$ indicates the leading car and suffix 2 does the trailing cars in a train.

3.2 Assumption for side force on leading car

The side force $F_S$ given by Eq. (3) can be divided into two forces for the leading car as shown in Fig. 6, because a crosswind acts on the front surface of a car-body as well as on the side surface. One of them is $F_M$, which acts on the side surface of the car-body, and the other is $F_F$, which acts on the front surface. Assuming a two-vehicle train, $F_M$ acts on the trailing car as well as on the leading car, while $F_F$ acts only on the leading car. Thus, assuming that the difference between the side forces acting on both cars corresponds to $F_F$, the relation between $F_F$ and $F_M$ is represented by Eq. (5), referring to the values in Fig. 5.

\[ F_F = \frac{C_{S1}}{C_{S2}} F_M = 0.23 F_M \]

where, $C_{S1}$ becomes maximum when the relative wind angle $\beta$ is 60°, where $C_{S1}$ and $C_{S2}$ are 1.07 and 0.87.
respectively. From Eqs. (3) to (5), the relation between $F_L$ and $F_M$ is obtained as follows:

$$F_L = C_{L1}F_M = 0.69F_M$$

(6)

where, the lift force coefficient $C_{L1}$ is 0.60 alike at the angle $\beta$ of 60°.

4. Simulation

4.1 Conditions

A curving simulation is carried out using a full-vehicle model, which is assumed to be a leading car. Fig. 7 shows an example of a SIMPACK window during the simulation. The parameter values of an express train on the conventional narrow-gauge lines of JR are used (see appendix). The wheel has the arc profiled tread and the rail profile is the 50-kgN type [14]. The reaction forces of the stoppers which restrain lateral and vertical relative motions between the car-body and trucks are set as shown in Fig. 8, where the movable spaces are modeled as dead zones.

Considering flange climb derailment caused by the reduction of wheel load on the outer rail, an external force corresponding to a crosswind acts on the vehicle from the outer side of a curved section as shown in Fig. 7. In addition to the effect of a crosswind, the running at lower speeds than the balanced speed contributes to the reduction of the outer wheel load under the over-canting condition and flange climbing is promoted on the outer rail. To avoid an impulsive effect in the simulation, each aerodynamic force is increased from zero at a constant rate for 20 s and then maintained at a predefined value as shown in Fig. 9, in which no values are indicated on the vertical axis since the magnitude of the aerodynamic force is varied in every simulation. Fig. 10 shows the track condition. Here, a tight curve with a radius of 160 m is assumed, which is the minimum curve radius prescribed in the regulations. The lengths of the transition curve and the circular curve are 65 m and 100 m, respectively. The cant and the curvature
change gradually in the transition curves, and the cant is 65 mm and the radius of curvature is 160 m in the circular curve. The friction coefficient $\mu$ between the wheel and rail is 0.4.

The train speed $V$ is restrained under strong winds depending on the natural wind speed $w$ as follows [15]:

- $25 \text{ m/s} \leq w < 30 \text{ m/s}$ : Slow down ($V=25 \text{ km/h}$)
- $w \geq 30 \text{ m/s}$ : Suspended

In the simulation, the running speed is set to the basic speed on the curve, $V=40 \text{ km/h}$, and the slow down speed, $V=25 \text{ km/h}$. It is premised that the vehicle proceeds into the entrance transition curve after the aerodynamic force has reached a constant predefined value, and it is assumed that the relative wind angle is fixed at $\beta = 60^\circ$ through running on the curved section; this is the condition under which derailment tends to occur most easily.

### 4.2 Derailment coefficient

The derailment coefficient $(Q/P)$, which is defined by the lateral force $Q$ divided by the wheel load $P$, is used as an index to indicate the likelihood of derailment. The derailment coefficients of the leading wheelset in each truck, the first wheelset and the third one, are shown in Fig. 11. Both wheelsets are focused on since the derailment coefficients of their outer wheels tend to become larger than those of trailing wheelsets. During the time until 20 s, the aerodynamic forces are increased toward predefined values. When the aerodynamic force does not act on the vehicle, namely in the case of $F_S = 0$, the values of $Q/P$ for both wheelsets reach a maximum on the circular curve. On the other hand, when the aerodynamic force of $F_S (=F_F+F_M) = 50 \text{ kN}$ acts on the vehicle, $Q/P$ increases greatly. Besides, the value of $Q/P$ on the first wheelset increases most in the exit transition curve and that on the third wheelset does in the entrance transition curve.

Nadal’s critical derailment coefficient, which is an index used to evaluate the potential for developing derailment, is expressed as follows [11]:

$$
\left( \frac{Q}{P} \right)_{Nadal} = \tan \gamma - \frac{\mu}{1 + \mu \tan \gamma}
$$

(7)

where, $\gamma$ is the flange angle and $\mu$ is the friction coefficient. In the case of $\gamma=65^\circ$ and $\mu=0.4$, which are parameter values considered in this study, the critical derailment coefficient becomes $(Q/P)_{Nadal} = 0.94$. Even when the derailment coefficient exceeds 0.94, however, the vehicle continues running, that is, derailment does not occur. This is because the attack angle $\psi$, which has a strong effect on derailment, is not contained in Eq. (7). To consider the effect of $\gamma$ on derailment, the friction coefficient $\mu$ is replaced with an equivalent expression $(f_f/N)$ [12].

$$
(f_f/N) = \mu \frac{\kappa/N \times \psi}{\left\{ \mu^2 + (\kappa/N \times \psi)^2 \right\}^{\frac{1}{2}}}
$$

(8)

where, $f_f$ is the lateral creep force, $N$ is the force acting on the contact patch in the normal direction, $\kappa$ is the lateral creep coefficient, and $\psi$ is an index indicating the saturation property of creep force. In this study, $\psi=1.5$ is used. $\kappa/N$ is obtained from the wheel profile and rail profile. Based on Hertz’s theory [16], semi-axes $a$ and $b$ of contact ellipse are determined, and using the approximation $\kappa = E \cdot a / N$, $\kappa/N$ can be represented as follows (see appendix):

$$
\frac{\kappa}{N} = \frac{E \cdot a}{N} = 1030 \times \frac{1}{N^{\frac{1}{3}}}
$$

(9)

where, $E$ is Young’s modulus. Finally, replacing $\mu$ in Eq. (7) with $(f_f/N)$ and considering the roll angle $\phi$ of the wheelset, the following equation is obtained as the critical derailment coefficient.

$$
\left( \frac{Q}{P} \right)_{crit} = \frac{\tan(\gamma + \phi) - (f_f/N)}{1 + (f_f/N)\tan(\gamma + \phi)}
$$

(10)

Hereinafter, “critical derailment coefficient” refers to that given by Eq. (10).

### 4.3 Comparison between $Q/P$ and $(Q/P)_{crit}$

Fig. 12 shows a case in which derailment occurs. The derailment coefficients $Q/P$ of the first and third wheelsets are compared with the critical derailment coefficients $(Q/P)_{crit}$ in the curved section, the critical derailment coefficients $(Q/P)_{crit}$ decrease while the derailment coefficients...
$Q/P$ increase. This is because the attack angle increases and then the expression $(f_y/N)$ becomes large in the curved section. In this example, derailment occurred after $Q/P$ of the first wheelset comes very close to its critical derailment coefficient $(Q/P)_{cri}$ in the exit transition curve. Thus, the values of $Q/P$ and $(Q/P)_{cri}$ at the timing that $(Q/P)_{cri} - Q/P$ becomes minimum are selected as representatives for the following examination.

Fig. 13 shows the representative values of $Q/P$ and $(Q/P)_{cri}$ at the timing that derailment occurs, varying the natural wind speeds $w$. Fig. 13 (a) is the case for the running speed $V = 25$ km/h and Fig. 13 (b) is for $V = 40$ km/h. In both running speeds, it is shown that the representative values vary with the wind speed and the derailment coefficients $Q/P$ are larger than the critical derailment coefficients $(Q/P)_{cri}$ in both wheelsets. In addition, the representative values of the third wheelset are larger than those of the first wheelset. Here, the symbol ■ indicates the derailed wheelset, and it is also found that the first wheelset derails in relatively low wind speeds, where the minimum value of $(Q/P)_{cri} - Q/P$ is about -0.05.

Fig. 14 shows positions at which derailment occurs under the different wind speeds. The position is expressed on the vertical axis as running distance. Here, it is necessary to note that derailment occurs at the wind speed $w$ over 35 m/s and, in a practical sense, train operation is suspended by regulations. If the train runs at $V = 25$ km/h, which corresponds to the slow down speed under strong winds, derailment occurs at a lower wind speed compared with the running at $V = 40$ km/h. As the wind speed increases, the position of derailment occurrence moves from the exit transition curve to the entrance transition curve through the circular curve. When the wind speed increases to 40 m/s-plus, overturn occurs on the tangent track before entering the curved section. This tendency is similar to the case of $V = 40$ km/h. Besides, at the relatively low wind speeds, the first wheelset derails in the exit transition curve.

Figs. 15 and 16 show enlarged waveforms of the derailment coefficient $Q/P$, the critical derailment coefficient $(Q/P)_{cri}$ and the vertical displacement of the outer wheel around the time of derailment occurrence. Fig. 15 corresponds to the case of $V = 40$ km/h and $w = 37.0$ m/s in Fig. 14, where the first wheelset derailed in the exit transition curve.
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Curve. After the time passed 42 s, \( Q/P \) of the first wheelset exceeds its \( (Q/P)_{cri} \) and then the vertical displacement of its outer wheel increases up to the flange height, 30 mm. Soon after that, the vertical displacement begins to decrease; this indicates that the first wheelset occurred flange climb and derailed.

Fig. 16 corresponds to the case of \( V = 40 \text{ km/h} \) and \( w = 40.1 \text{ m/s} \) in Fig. 14, where the third wheelset derailed in the entrance transition curve. Here, \( Q/P \) of the third wheelset exceeds its \( (Q/P)_{cri} \) at the time around 29.5 s, and then the vertical displacement of its outer wheel increases up to 30 mm, and then it begins to decrease; hence, the third wheelset is deemed to have derailed.

Fig. 17 is the case of \( V = 40 \text{ km/h} \) and \( w = 40.8 \text{ m/s} \) in Fig. 14; the enlarged waveforms in the entrance transition curve are shown. Here, \( Q/P \) of the third wheelset exceeds its \( (Q/P)_{cri} \) and begins flange climbing earlier than the first wheelset. Regarding to the vertical displacement of the outer wheels, the third wheelset continues to increase. 

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Fig. 15 Derailment of the first wheelset \( (V = 40 \text{ km/h}, w = 37.0 \text{ m/s}) \)

Fig. 16 Derailment of the third wheelset \( (V = 40 \text{ km/h}, w = 40.1 \text{ m/s}) \)

Fig. 17 Derailment of the first wheelset and lift of the third wheelset \( (V = 25 \text{ km/h}, w = 40.8 \text{ m/s}) \)
further above the flange height of 30 mm though the first wheelset decreases after it reaches 30 mm. In addition, the lateral displacements of both wheelsets increase monotonically. This means that the first wheelset derailed in the front truck and the third wheelset shifted laterally lifting over the rail. The lifting behavior of the third wheelset is caused by the decreased centrifugal force while running at the slow down speed and also by the wheel unloading of the rear truck due to the increasing cant on the entrance transition curve, where the rear suspension tends to expand.

4.4 Safety allowance under slow down operation

In the preceding section, the derailment behavior under crosswinds was examined at the wind speeds over the regulation speed for suspended operation. In this section, safety allowance against derailment is investigated under the slow down operation amid conceivable wind speed, using the critical derailment coefficient \( Q/P \).

A running simulation on the curved track, where the cant is 65 mm and the radius of curvature is 160 m, is carried out under a crosswind fixing the wind speed to \( w = 30 \text{ m/s} \), which is almost the maximum wind speed for the slow down operation. Therefore, the train speed is set to the slow down speed \( V = 25 \text{ km/h} \) and, for comparison, the normal operation speed \( V = 40 \text{ km/h} \) on the intended curve.

Under the above mentioned condition, derailment does not occur. Thus, safety allowance was evaluated by comparing \( Q/P \) with \( (Q/P)_{cri} \). The minimum values of \( (Q/P)_{min} - Q/P \) on each wheelset are shown in Table 1 (a) for \( V = 25 \text{ km/h} \) and Table 1 (b) for \( V = 40 \text{ km/h} \).

<table>
<thead>
<tr>
<th>Wheelset</th>
<th>( Q/P )</th>
<th>( (Q/P)_{cri} )</th>
<th>( (Q/P)_{min} - Q/P )</th>
</tr>
</thead>
<tbody>
<tr>
<td>First wheelset</td>
<td>1.18</td>
<td>1.35</td>
<td>0.17</td>
</tr>
<tr>
<td>Third wheelset</td>
<td>1.08</td>
<td>1.39</td>
<td>0.31</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Wheelset</th>
<th>( Q/P )</th>
<th>( (Q/P)_{cri} )</th>
<th>( (Q/P)_{min} - Q/P )</th>
</tr>
</thead>
<tbody>
<tr>
<td>First wheelset</td>
<td>1.16</td>
<td>1.37</td>
<td>0.21</td>
</tr>
<tr>
<td>Third wheelset</td>
<td>0.90</td>
<td>1.39</td>
<td>0.49</td>
</tr>
</tbody>
</table>

In Section 4.3, it was shown that derailment does not occur unless \( Q/P \) exceeds \( (Q/P)_{cri} \) in Fig. 13, and that the minimum values of \( (Q/P)_{min} - Q/P \) in which derailment occurs are less than about -0.05. Here, every value of \( (Q/P)_{min} - Q/P \) in Table 1 is positive. This means that the safety allowance for derailment is secured if the wind speed \( w \) is less than 30 m/s. However, it must be noticed that the safety allowance at the slow down speed \( V = 25 \text{ km/h} \) is lower than that at the basic speed \( V = 40 \text{ km/h} \).

5. Conclusions

By means of the multibody simulation software SIMPACK, a running simulation was carried out for a bogie vehicle negotiating a tight curve with the radius of curvature 160 m. The vehicle was assumed to be the one on the JR narrow gauge lines. In the simulation, an external force simulating a crosswind was acted from the outer side of the curve. By employing an equivalent frictional feature \( (f_r/N) \) in place of the friction coefficient \( m \) in the Nadal’s formula of critical derailment coefficient, the behavior of flange climb derailment was examined. The obtained results are as follows:

- The derailment coefficient \( Q/P \) at the timing, which flange climbing occurs, comes very close to the critical derailment coefficient \( (Q/P)_{cri} \) calculated from Nadal’s formula modified with the equivalent frictional feature which depends on the attack angle. This is especially true under the condition that derailment occurs in the exit transition curve.

When the wind speed is relatively low, the first wheelset derails in the exit transition curve. This is because the outer wheel load of the front truck decreases owing to the decreasing cant on the exit transition curve.

As the wind speed increases, the position of derailment moves to the entrance transition curve, where the third wheelset is derailed by the wheel unloading of the rear truck due to the increasing cant.

It is confirmed that the safety allowance against derailment is secured under the slow down operation at \( V = 25 \text{ km/h} \) even if the wind speed is \( w = 30 \text{ m/s} \). However, it must be noted that the safety allowance at the slow down speed is lower than that of the normal operation at the speed of \( V = 40 \text{ km/h} \).

Appendix

The derivation of Eq. (9) is given below. The following equations are defined in Hertz’s theory [16].

\[
A + B = \frac{1}{2} \left( \frac{1}{R_{nx}} + \frac{1}{R_{ny}} + \frac{1}{R_{rx}} + \frac{1}{R_{ry}} \right)
\]

\[
B - A = \frac{1}{2} \left( \frac{1}{R_{nx}} - \frac{1}{R_{ny}} + \frac{1}{R_{rx}} - \frac{1}{R_{ry}} \right)
\]

where, \( R_{nx} \) and \( R_{ny} \) are the longitudinal and lateral curva-
tural radii of the wheel, and $R_a$ and $R_c$ are those of the rail, respectively. The long and short semi-axes of the contact ellipse, $a$ and $b$, are given by the following equation.

$$\left(\frac{a}{m}\right)^3 = \left(\frac{b}{n}\right)^3 = \frac{3N(1-\nu^2)}{2E(A+B)}$$

where, $n$ is Poisson’s ratio and $E$ is Young’s modulus of wheel and rail. $m$ and $n$ are constants given in the table using the parameter $\eta$ defined as Eq. (15).

$$\cos \eta = \frac{B-A}{B+A}$$

In this study, considering the contact at wheel flange, $R_{w} = 0.449$ m, $R_{a} = \infty$, $R_{c} = \infty$ and $R_{b} = 0.013$ m. Here, $h$ is calculated as $19.3^\circ$ and the constants $m$ and $n$ are obtained from the table in Ref. (16) as 4.05 and 0.405, respectively. Thus, using $a$ and $b$ calculated by Eq. (13) with $\nu = 0.3$ and $E = 210$ GPa, and $\kappa/n = 1032 \times 10^{1/3}$ is obtained as shown in Eq. (9).

The parameters used in the calculation are indicated as below.

- Mass of car-body: 30.446 t
- Mass of bogie frame: 2.45 t
- Mass of wheelset: 1.6 t
- Mean radius of wheel: 0.425 m
- Bogie wheelbase: 2.3 m
- Distance between centers of bogies: 14.4 m
- Lateral distance between axle springs: 1.64 m
- Lateral distance between air springs: 1.93 m
- Height of bogie frame mass center from rail top: 0.50 m
- Central height of air spring from rail top: 0.777 m
- Height of car-body mass center from rail top: 1.67 m
- Roll radius of inertia of car-body: 1.5 m
- Pitch radius of inertia of car-body: 6.5 m
- Yaw radius of inertia of car-body: 6.0 m
- Roll radius of inertia of bogie frame: 0.53 m
- Pitch radius of inertia of bogie frame: 0.80 m
- Yaw radius of inertia of bogie frame: 0.78 m
- Roll radius of inertia of wheelset: 0.59 m
- Pitch radius of inertia of wheelset: 0.25 m
- Yaw radius of inertia of wheelset: 0.59 m
- Lateral stiffness of axle box (per axle): 11 180 kN/m
- Longitudinal stiffness of axle box (per axle): 11 200 kN/m
- Vertical stiffness of axle spring (per axle): 1 440 kN/m
- Vertical damping of axle damper: 39.2 kN/s/m
- Damping of lateral damper: 58.8 kN/s/m
- Longitudinal and lateral stiffness of air spring: 219 kN/m
- Vertical stiffness of air spring: 371 kN/m
- Vertical damping of air spring: 17.5 kN/m
- Central height of wind pressure from rail top: 2.24 m

References

13. p. 70 in Reference (8).
15. pp. 35-37 in Reference (8).

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