Safety Assessment for Flange Climb Derailment of Trains Running at Low Speeds on Sharp Curves

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Flange climb derailment of railway vehicles occurs during low-speed running on sharply curved and twisted tracks. We studied the mechanism behind flange climbing to clarify and quantify the factors that cause derailment. In this study, passenger vehicle running tests were carried out on an RTRI test line subsidized by the Ministry of Land, Infrastructure and Transport. Derailment actually occurred during the tests, which measured wheel/rail contact forces, the forces exerted on the wheel sets and the dynamic behavior of the vehicle. Bench tests were also performed using roller rigs to investigate the characteristics of creep force. Based on the test results, this paper proposes a method to assess levels of safety against vehicle derailment on curved tracks.

Keywords: railway vehicle, flange climb derailment, safety assessment, curve negotiation, creep force, derailment quotient, ratio of off-loading

1. Introduction

A train derailment occurred at Nakameguro Station on the Tokyo Metro Co. Hibiya Line in March 2000. The accident was a typical flange climb derailment at the initial section of a transition curve following a sharp left curve, where the load on the right wheels (i.e. those on the high rail) decreased due to over-canting and track twist. After this accident, anti-derailing guards were installed as an emergency measure on curve sections with a radius of less than 200 m, and safety assessment was enforced using a theoretical estimated Y/Q ratio calculation (where Y, Q and Y/Q represent lateral force, wheel load and derailment quotient respectively). More detailed research into the mechanism behind flange climb derailment has also been performed to clarify and quantify safety measures against the problem.

In this research, passenger vehicle running tests were carried out on an RTRI test line subsidized by the Ministry of Land, Infrastructure and Transport. Derailment actually occurred during the tests, which measured wheel/rail contact forces, the force exerted on the wheel sets and the dynamic behavior of the vehicle. Bench tests were also performed using roller rigs to investigate the characteristics of creep force between wheel and rail. Methods of safety evaluation were then verified based on the test results and on theoretical analysis.

Using analytic equations and field test data, this paper describes methods of safety assessment against flange climb derailment during low-speed running on sharp curves.

2. Conventional safety assessment methods

2.1 Safety assessment using analytic equations

When a railway vehicle exits a sharp curve at low speed, the outer load on the lead wheelset decreases, and large angles of attack and lateral force are exerted on the outer wheel flange. Flange climb derailment therefore tends to occur where the coefficient of friction between wheel and rail is high. The estimated Y/Q ratio shown below is used to evaluate safety in such low-speed operation.

\[\text{Estimated } Y/Q = \frac{\text{Critical } Y/Q}{\text{Estimated } Y/Q} \quad \cdots \cdots (1)\]

Critical Y/Q is calculated by Nadal’s equation using parameters involving the angles of attack and wheel/rail contact, and Estimated Y/Q is calculated by equations estimating wheel load and lateral force using parameters involving the geometry of the vehicle and track. The train speed is set at 10 km/h for this calculation.

Though the base value of the estimated Y/Q ratio is 1.0, the value must not be less than 1.2 to be within the safety margin.

2.1.1 Critical Y/Q

The critical Y/Q is calculated by the following equation.

\[\tan \alpha = \frac{(f_y/N)}{1 + (f_y/N) \tan \alpha} \quad \cdots \cdots (2)\]

Where \(\alpha\) = flange angle (rad), \(f_y\) = lateral creep force (N),
\[ N = \text{normal force (N)}, \quad \frac{(f_\beta / N)}{\mu} = \frac{\psi \times \kappa \times (N/N)}{\mu^\beta \times (\psi \times \kappa \times (N/N))^\beta} \]

Where \( \psi \) = angle of attack (rad), \( \mu \) = coefficient of friction at wheel flange ( = 0.3), \( \beta \) = index expressing saturation characteristic of creep force ( = 1.5), and \( \kappa \) = lateral creep coefficient (\( \kappa / N = 27.0 \)).

Equation (3) expresses the relationship between the angle of attack and the \( f_\beta / N \) ratio, and is known as the Levi-Chaltet Equation. The value of \( \kappa / N = 27.0 \) is defined based on past theoretical analysis and experiment results. The coefficient of friction between the wheel flange and rail gage corner \( \mu \) is set at 0.3. The angle of attack \( \psi \) is calculated using equation (4).

\[ \psi = \frac{a}{R} \times \psi_f - \psi_w \]

Where \( 2a \) = wheelbase(m), \( R \) = curve radius at the center of the leading bogie (m), and \( -\psi_f - \psi_w \) = yawing angle of truck-frame and wheelseet to the track center, calculated using estimation models according to the curvature, slack, wheelbase and profile of the wheel tread. The models are based on vehicle dynamics simulation for standard commuter trains, which have a distance of around 14 m between bogie centers and a wheelbase of approximately 2 m.

To ensure running safety, the maximum value of friction coefficient \( \mu \), the accuracy of the angle of attack's estimated value and the \( f_\beta / N \) ratio (equation (3)) must be verified.

### 2.1.2 Estimation equations of \( Q \) and \( Y \)

The values for wheel loads \( Q \) are calculated using factors including axle load, imbalance between the right and left static wheel loads, suspension stiffness, the height of the vehicle’s center of gravity, curve radius, cant, twist and track irregularities. The estimated wheel load values correspond closely to the field test results. However, the lateral forces \( Y \) exerted on the outer wheel flange are difficult to estimate accurately as they are affected by creep forces between wheel and rail on both the outer and inner sides. The lateral force \( Y \) values are therefore calculated by equation (5) using the inside \( Y/Q \) ratio model described later.

\[ Y = \kappa \times Q_i + Y_{AS} \]

Where \( \kappa \) = inside \( Y/Q \) ratio, \( Q_i \) = inside wheel load (kN), and \( Y_{AS} \) = axle load (kN) \times unbalanced centrifugal acceleration / acceleration of gravity ( = angle of cant excess)

The maximum value of \( \kappa \) is set at 0.55 based on the Hibiya Line field test results, and the inside \( Y/Q \) ratio \( \kappa \) is given by estimation models based on curvature and wheel tread profile obtained from computer simulation analysis.

To ensure running safety, the accuracy of the maximum value and the estimation models for the inside \( Y/Q \) ratio \( \kappa \) must be verified using experimental data under a range of curvature and attack angle conditions.

### 2.2 Safety assessment through running tests

When a new railway vehicle is introduced for commercial use, it must be tested on a commercial line under normal operating conditions. In field testing, wheel loads \( Q \), lateral forces \( Y \) and the \( Y/Q \) ratio are measured using load-measuring wheelsets. The \( Y/Q \) ratio must be less than the limiting value, which is set, for example, to 0.8 at a wheel flange angle of 60 degrees. When a vehicle exits a sharp curve at low speed, the measured \( Y/Q \) ratio sometimes exceeds the conventional limiting value because the tangent area of the wheel flange (whose angle is constant) touches the rail during curving. Especially high \( Y/Q \) values are observed at curves where lubricating oil is applied to the gage corner as a measure against wear to the rail and wheel.

To properly create a vehicle curving scenario, in addition to the \( Y/Q \) ratio it is necessary to add a new index and a limit for safety evaluation whose values do not depend on the coefficient of friction at the wheel flange. A procedure for safety assessment on sharp curves at low speeds must also be established taking into account the mechanism behind flange climb derailment and the high value of friction coefficient.

### 3. Outline of experiments

#### 3.1 Bench tests for creep force characteristics

Creep force tests were performed for three years from 2001. To investigate the characteristics of creep forces between wheel and rail when the flange starts to climb the rail, roller rigs with a diameter of 1,600 mm and a full-scale wheelseet featuring conventional standard wheels with a flange angle of 60 degrees were used. The outline of the test apparatus is shown in Fig.1. In these tests, the angle of attack and the load balance on the left and right wheels were changed. The creep forces, lateral and vertical displacement of wheels and angle of attack were then measured.

#### 3.2 Running tests on the RTRI test line

Tests were performed for two years from 2001 (an outline of the running tests is shown in Fig.2). The rail’s cross-section profile used a Japanese 50N rail, and for the wheel a standard cone profile for conventional lines with a flange angle of 60 degrees was used. Anti-derail-
ing guards were installed on curve sections for continuous observation of the dynamic behavior of the wheelset from the beginning of flange climbing to derailment (i.e. when the flange rides on the rail head). Flange climbing occurred at the point where the twist in a track length of 5 m exceeded 20 mm. The relative vertical displacement between wheel and rail (i.e. the flange climb height), the angle of attack, wheel/rail contact forces, external forces and moments exerted on the wheelset were measured in these tests.

4. Results of creep force characteristics tests

In the following sections, the term inside wheel represents the wheel whose tread is in contact with the head of the rail or roller rig, and outside wheel represents the wheel whose flange is in contact with the gage corner of the rail or roller rig.

4.1 Influence of attack angle on creep forces

The relationship between the angle of attack and the \( f/N \) ratio of the wheels on both sides is shown in Fig.3 (\( f \) and \( N \) represent the creep force and normal force respectively). In Fig.3, the small circles represent test data under dry conditions, and the solid lines denote calculation results using Kalker’s theory with a friction coefficient of \( \mu = 0.6 \). The test data shows that the attack angle when the \( f/N \) ratio of the inside wheel is greatest is approximately twice that of the theoretical value. The test results demonstrate the following points:

1. The maximum value of the \( f/N \) ratio on the outside wheel is about 0.6 (average = 0.516, standard deviation = 0.066) under dry conditions.
2. The maximum value of the \( f/N \) ratio on the inside wheel is about 0.55 under dry conditions.
3. The creep force on the outside wheel is always saturated and independent of the attack angle because of high creepage in the area in contact with the flange.
4. The creep force on the inside wheel saturates as the attack angle reaches approximately 1.0 - 1.5 degrees.

The outside wheel’s dispersive data is thought to be a result of the roughness and activation of flange surface changing frequently during the test due to high creepage. The same coefficient of friction values were measured on rail surfaces in commercial lines [3].

4.2 Lateral creep force on the outside wheel

Figure 4 shows the relationship between angle of attack and the \( f_z/N \) ratio or \( f_y/N \) ratio, based on the test results shown in Fig.3 (\( f_x \) and \( f_y \) denote the longitudinal and lateral creep forces respectively). Lateral creep force
has a strong influence on flange climb derailment as shown in equation (2). The solid lines in Fig.4 show results calculated the same way as those in section 4.1.

The test results demonstrate the following points:
1. As the attack angle becomes larger, longitudinal creep force decreases and lateral creep force increases because the creep force on the outside wheel is always saturated.
2. As the wheel flange climbs the rail, longitudinal creep force increases and lateral creep force decreases, even if the angle of attack does not change.

If the saturation characteristic of the outside wheel’s lateral creep force at a flange climbing height of 10 mm is applied to equation (3), calculation results substituting $\beta$ with 2.0 and $\kappa_{ce}/N$ with 35 correspond closely to the test data.

5. Results of running tests on the RTRI test line
5.1 Angle of attack according to curve radius

The attack angles of the leading wheelset were measured in these running tests. The coefficient of friction on the rail as measured with a tribometer was between 0.35 and 0.47. The relationship between the attack angle and the curve radius at the leading wheelset is shown in Fig.5. The solid line in Fig.5 shows the calculation results using equation (4) and estimation models. It is confirmed that the estimated values for the angle of attack correspond to the maximum values in the test data depending on the curve radius, and that the angle of attack increases sharply on curves with a radius of less than 200 m for standard commuter vehicles.

5.2 Inside Y/Q ratio

The data for the inside Y/Q ratio measured in the running tests and the curvature at the leading wheelset were adjusted. The coefficient of friction on the rail measured using a tribometer was about 0.4 on average, and ranged from 0.39 to 0.43. The relationship between the curvature and the inside Y/Q ratio is shown in Fig.6. The solid line in Fig.6 shows estimation models for $\mu = 0.4$ and the standard cone-profile wheel described in section 2.1.2. It is confirmed that the estimated values for the inside Y/Q ratio according to the curvature correspond to the average data from the test results.

6. Verification of the estimated Y/Q ratio

The accuracy of the models for estimating the attack angle and the inside Y/Q ratio was verified through running tests on the RTRI test line. Bench tests were also carried out to investigate the creep force characteristics of the wheels on both sides. If the maximum values for the inside Y/Q ratio $\kappa$ and the friction coefficient $\mu$ of the outside wheel are set at 0.55 and 0.6 respectively, anti-derailment safety can be sufficiently evaluated.

The critical Y/Q value according to the curve radius can be deducted from (1) equation (2), (2) the saturation characteristics of the $f_y/N$ ratio according to the attack angles, and (3) the estimation models for angles of attack according to the curve radius, i.e. equation (4). The results for the critical Y/Q ratio calculated in this way are shown in Fig.7. The maximum value of friction coefficient $\mu$ at the outside wheel is set at 0.6. In Fig.7, the values calculated using the conventional method are also plotted. Under conventional calculation, the critical Y/Q ratio is derived from equations (2), (3) and (4) with the outside friction coefficient $\mu = 0.3$ and divided by the factor of safety 1.2.
7. Safety assessment method on low-speed running tests

In the field test, it was not generally possible to obtain values for the coefficient of friction on rail and wheel. To ensure safety under actual curving conditions, it is therefore necessary to introduce a new index (in addition to the \( Y/Q \) ratio) whose values do not depend on the coefficient of friction at the wheel flange.

The running tests measured changes in wheel/rail contact forces and wheelset behavior from the beginning of flange climbing to derailment, and the relationship between running distance and the height of flange climbing and several indices based on \( Q \) and \( Y \) data was investigated.

7.1 Running distance necessary for derailment

A large imbalance between the right and left static wheel loads was set, and the test vehicle was run on a circular curve with the tangent area of the outside wheel flange in contact with the rail gage corner. The outside wheel began to climb at the point where track twist over a 5 m length exceeded 20 mm. The test data showing the running distance from the beginning of flange climbing is shown in Fig.8. The vehicle was found to run for about 1 m from the beginning of flange climbing at the height of flange climbing exceeded 20 mm. The test data showing the height of flange climbing and several indices based on \( Q \) and \( Y \) is shown in Fig.9. The results in Fig.8 and Fig.9 indicate that the components of indices with a frequency of under 5 Hz can be applied to the evaluation of running safety at speeds between 10 km/h and 20 km/h.

7.2 Indices for safety evaluation

Figure 10 shows a sample wave form for the height of flange climbing and the forces exerted on the leading wheelset. The results indicate that the following equations hold for the vehicle at low speed during flange climbing:

\[
\cdot Q_l + Q_b \approx 2 Q_0 (= \text{constant}) \]
\[
\cdot X_l + X_b \approx 0 (= \text{constant}) \]
\[
\cdot Y_b - Y_0 \leq 0 \]
\[
\cdot \Delta Q/Q_0 - M/(2bQ_0) \approx 0 (= \text{constant})
\]

From the equation of motion for the rolling of a wheelset and the relationship given by equation (6), approximated equation (7) can be deduced as the rolling acceleration becomes equal to zero.

\[
\frac{\Delta Q}{Q_0} = \frac{4\left(b - \frac{r^2}{2b}\right) - \kappa(2r + z) + \frac{Y}{Q}[(2r + z) - \frac{z}{2}]}{\kappa(2r + z) + \frac{Y}{Q}[(2r + z) - \frac{z}{2}]} \quad (7)
\]

Where \( \Delta Q/Q_0 \) is ratio of off-loading, \( r \) is radius of wheel, \( 2b \) is distance between right and left wheel/rail contact points, \( b \) is lateral displacement of wheel/rail contact point from the neutral position to the beginning of flange climbing (= 0.032 m), \( r \) is change of wheel radius from the neutral position to the beginning of flange climbing (= 0.0090 m), \( z \) is vertical displacement of wheel from the neutral position to the beginning of flange climbing (= 0.0012 m), \( Y/Q \) is derailment quotient (= equation (2)), and

\[
\kappa = \frac{\tan \alpha}{1 - \left(\frac{f_c}{N}\right) \tan \alpha} : \text{inside } Y/Q \text{ ratio.} \quad (8)
\]

Where \( \alpha = \text{contact angle (rad)} \), \( f_c = \text{lateral creep force (N)} \),

\[
\Delta Q/Q_0 = \text{Derailment quotient.}
\]

\[
\text{Static wheel load} = Q_k + Q_b + Q_0
\]

\[
\text{Derailment} = \frac{Z_w}{(Q_k + Q_b + Q_0) / 2} \]

\[
\text{Lateral force} = X_l + X_b
\]

\[
\text{Longitudinal force} = Y_l
\]

\[
\text{Rolling moment} = M/(2bQ_0)
\]

\[
\text{Distance between right and left wheel/rail contact points} = 2b
\]

\[
\text{Running distance (m) to Derailment}
\]

\[
\text{Running distance (m) to Derailment}
\]

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\text{Running distance (m) to Derailment}
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\text{Running distance (m) to Derailment}
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\[
\text{Running distance (m) to Derailment}
\]
The relationship between the \( Y/Q \) ratio and the off-loading ratio \( \Delta Q/Q_0 \) is shown in Fig. 11, where the data measured in the running tests and the calculation results from equation (7) are plotted. It is confirmed that the test data corresponds approximately to the line derived using equation (7).

It is found that \( (Y_R - Y_I)/2Q_0 \) is also closely related to the height of flange climbing, but the index is not suitable for safety assessment. It was therefore decided to use the relationship among \( Y/Q \), \( \Delta Q/Q_0 \) and \( \kappa \) (i.e. equation (7)), and to investigate the influence of the friction coefficient on these indices.

### 7.3 Influence of the coefficient of friction on indices

The \( Y/Q \) ratio varies according to the coefficient of friction, as shown by equations (2) and (3). However, the ratio of off-loading \( \Delta Q/Q_0 \) is not so affected by outside coefficients of friction \( \mu \) because the friction coefficient of the outside wheel flange influences not only the creep forces on the outside wheel but also those on the inside wheel. The influence of the friction coefficient on the indices is shown in Fig. 12.

### 7.4 Limiting values of \( \Delta Q/Q_0 \)

The points described above indicate that the ratio of off-loading \( \Delta Q/Q_0 \) is suitable for use as the safety evaluation index in low-speed running tests.

The critical \( \Delta Q/Q_0 \) value according to the curve radius can be deduced from equations (2), (4), (7) and (8) using the outside friction coefficient \( \mu = 0.6 \), the inside \( Y/Q \) ratio \( \kappa = 0.55 \) and the saturation characteristics of lateral creep forces obtained by the bench tests. The limiting values of \( \Delta Q/Q_0 \) are shown in Fig. 13. The limit for curves with a radius of over 245 m is set at 0.6, with the safety margin based on the conventional criteria.

### 7.5 Procedure for safety assessment in low-speed running tests

The following procedure is proposed for safety assessment in low-speed running tests:

1. When the \( Y/Q \) ratio at a frequency of less than 5 Hz does not exceed the conventional limiting \( Y/Q \), the vehicle is judged to be able to run safely.
2. If the measured \( Y/Q \) exceeds the conventional limiting \( Y/Q \), the off-loading ratio \( \Delta Q/Q_0 \) at the point should be measured whose frequency is less than 5 Hz. When the \( \Delta Q/Q_0 \) does not exceed the limiting value of \( \Delta Q/Q_0 \) shown in Fig. 13, the vehicle is judged to be able to run safely.

The suitability of this procedure is demonstrated by the test data (Fig. 14).

### 8. Conclusions

To clarify and quantify the mechanism behind flange climbing and the factors that cause derailment, the authors carried out passenger vehicle running tests on an RTRI test line and bench tests using roller rigs to investigate creep force characteristics. It was confirmed that rail vehicles can run safely at low speeds on curves where anti-derailing guards may not be installed, when safety is assured by the conventional evaluation method using the estimated \( Y/Q \) ratio, even if the coefficient of friction at the outside wheel increases up to 0.6. We have also proposed a safety assessment procedure using the off-
loading $\Delta \frac{Q}{Q_0}$ ratio in low-speed running tests on curved and twisted tracks.

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**References**