Safety Evaluation of Railway Vehicle against Crosswind Applying a Full-vehicle Model

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In order to analyze the behavior of railway vehicles subjected to crosswinds in more detail, we constructed a simulation program applying a full-vehicle model developed from a half-vehicle model. Using the full-vehicle simulation program, investigations were made into the influence of factors not considered in the half-vehicle model on overturning or wheel unloading ratio. Results revealed that the wheel unloading ratio reaches the maximum value when the yawing moment is zero. It also became apparent that when considering the relationship between the wind speed and the wheel unloading ratio or overturning, the relationship could be appropriately evaluated by using the wheel unloading ratio averaged through the vehicle as an indicator, regardless of the static wheel load imbalance of each axle.

Keywords: crosswind, critical wind speed of overturning, wheel unloading ratio, full-vehicle simulation, running safety

1. Introduction

The tendency toward lighter and faster rolling stock in recent years has become the main reason for lower safety margins against crosswinds. It is thus important to be able to evaluate the running safety of rolling stock against crosswinds, accurately.

One of the indices used for this evaluation is the “wheel unloading ratio” or the “critical wind speed of overturning.” The critical wind speed of overturning is defined as the wind speed which corresponds to the aerodynamic force required to make the windward wheel load fall to zero, i.e., the wheel unloading ratio of 100%. In Japan, the Kunieda static analysis formula [1], or the Detailed Equation based on this formula and reflecting recent research results [2], are often employed to estimate the wheel unloading ratio or critical wind speed of overturning. The latest series of studies by the Railway Technical Research Institute, using a dynamic analytical model expanded from the Detailed Equation, have clarified the influence of the fluctuating crosswind frequencies on dynamic vehicle behavior which had not previously been taken into account, and have verified the validity of the analytical model through a full scale experiment [3][4].

However, all of these researches were carried out using only a half-vehicle model, which can not take into account the dependence of the wheel unloading ratio (or critical wind speed of overturning) on the yawing moment induced by crosswind and the static wheel load imbalance of each wheelset. The yawing moment is especially critical in certain instances such as the exit of a tunnel where wind suddenly strikes the vehicle as it emerges from the tunnel, consequently, the magnitude of aerodynamic force differs between the front and the rear of the vehicle body, so it is necessary to quantitatively assess its impact beforehand.

In this research, a simulation program was constructed applying a full-vehicle model expanded from the above-mentioned dynamic analytical model (hereinafter referred to as the “full-vehicle simulation program”), and examined the dependence of the wheel unloading ratio (or critical wind speed of overturning) on those factors not previously taken into account by the half-vehicle model. This paper reports the investigation results.

2. Construction of the Full-Vehicle Simulation Program

2.1 Vehicle model

The full-vehicle model for this simulation was expanded from the half-vehicle dynamic analytical model constructed in Reference [3]. Specifically, it is a model in which yawing and pitching are added to the degrees of freedom (left/right, up/down and rolling) in the half-vehicle model. The y-z and x-z planes of the vehicle model are shown in Fig. 1. The characteristics of this vehicle model are that: its y-z plane is identical to that of the half-vehicle model; when the differences from the earlier static analysis and dynamic analysis with the half-vehicle model are examined, the influence of the difference of the analytical model upon the calculation results can be minimized; consistency with the past investigation results can be readily established; and a simulation program can be built with relative ease. Furthermore, this vehicle model was designed so that, in order to examine the dependence of the static wheel load imbalance on the wheel unloading ratio (or critical wind speed of overturning), the static wheel load imbalance can be set by giving the thickness of the spacer (adjustment plate) as the initial value for the vertical displacement of the axle and bolster springs. Using this vehicle model, the equations of motion relating to various mass factors (31 degrees of freedom in total) were derived and a program to solve the equations by the Runge-Kutta-Gill method for every 1/10,000 second was created.
2.2 Verification of the Simulation Program

2.2.1 Dynamic Lateral Force Load Test Using Actual Car

This section describes a test conducted using an actual car for verifying the validity of the full-vehicle simulation program. Lateral forces capable of generating a yawing moment were dynamically applied to the actual car and the measured values obtained were compared and collated with the calculated simulation data.

The test was performed on the rolling stock test plant in the Railway Technical Research Institute (Fig. 2). Lateral force loading columns were erected outside the testing shed. Motor-driven actuators attached to the columns were roped to the car body with wires. Then, the car was hauled in the lateral direction at two points: one point near the front door and the other point near the rear door of the car. The measuring targets included lateral force, wheel load, car body displacement, and internal pressure of air springs. Lateral force was measured using series-connected load cells between the wire rope and car body. Wheel load was measured using a measuring wheelset of wheel/rail contact force. As to car body displacement, lateral and vertical displacements from the ground level were measured by a potentiometric displacement gauge. The inclination of the car body was calculated from these measurements.

Similar verification tests for the half-vehicle model were reported previously in References [3] and [4]. While in those previous tests the lateral forces were in phase at the front position and at the rear position, the tests reported in this paper were conducted with the lateral forces out of phase between at the front position and at the rear position in order to estimate the influence of yawing moment. Specifically, assuming a wind speed with a sine-wave-like fluctuation and a lateral force proportional to the square of the wind speed, the following cases (1) through (4) were selected. The fluctuating frequency was varied from 0.05 to 2.5 Hz for each case. Also, as an additional case (5), verification was made using the aerodynamic force of natural wind, based on experimental results obtained by the measuring tests of aerodynamic forces on a full-scale train vehicle model subjected to natural winds conducted by the Railway Technical Research Institute in a windy area of Hokkaido from December 2001 through March 2004 [5].

In these tests, the lateral and lifting forces were measured at both the front and rear positions of the car body, and the results of dynamic lateral force load tests using these lateral forces were compared with the simulation results obtained using the same lateral force data.

- Case (1) Single wave (the front position only)
- Case (2) Single wave (1/4 periodic lag between the front and the rear position)
- Case (3) Continuous wave (anti-phase between the front and the rear position)
- Case (4) Continuous wave (1/4 periodic lag between the front and the rear position)
- Case (5) Aerodynamic force under natural wind

From the above cases, typical examples of the verification results in the cases (3) and (5) are shown in Figs. 3 and 4. The figures (a), (b) and (c) show the measured values of lateral force applied at each position of the front and the rear of the car in the dynamic lateral force load test, the measured and calculated results of the wheel unloading ratio at the axle No. 1 and the measured and calculated results of the car body yaw angle, respectively. From figures (b) and (c), general agreement was confirmed between the measured and calculated results in the wheel unloading ratio and car body yaw angle. Also, concerning test results other than those in the cases shown here, there was general agreement between the measured and calculated results. Thus, it has been confirmed that the full-vehicle simulation model conducted for the present research is generally an appropriate tool for estimating wheel load variations under the influence of yawing moments due to external force, which was impossible to do using the former half-vehicle model.
2.2.2 Wind Tunnel Test Using 1/10 Scale Train Model

This section describes a wind tunnel test conducted to verify the validity of the full-vehicle simulation program. A scale model in which static wheel load imbalance was occurring in each wheelset was used for the test. Measured values were compared and collated with the calculated simulation data.

The train model in this test is a one-tenth scale model of a commuter car for meter-gauge railway lines (103 series car). The characteristics of this train model are as follows: its shape is geometrically similar because the shape has great influence on aerodynamic force; the bogie spring system including stoppers which restrain excessive displacement of the car body is accurately reproduced; the spring constants are chosen considering the law of similarity between external forces and the resultant displacement of the car body. Concerning the law of dynamic similarity, with spring force (1), gravitational force (2), inertial force (3) and aerodynamic force (4) considered as physical quantities, multiples that would realize the law of similarity to the original in the relationship between aerodynamic force and car body displacement were investigated. Specifically, in view of the relationship between spring force and gravitational force, the spring constant and mass of the model were respectively set at 1/100 and 1/1000 that of the original. Particulars of the train model were designed according to meter-gauged commuter cars used in recent years. The external form of the car body was modeled on the 103 series car mentioned earlier. A bolsterless bogie was used as a model. For the car mass, a lightweight stainless steel car was assumed. Specifically, assuming a commuter car with a mass of approximately 25 tons, the mass of the model was set to approximately 25 kg (approximately 15 kg of car body mass and approximately 5 kg of bogie mass), which is 1/1000 that of the assumed commuter car. Also, presuming axle and bolster spring constants for the typical bolsterless bogie, coil springs close to 1/100 the presumed specification were selected.

Using the large-scale low-noise wind tunnel (Maibara Wind Tunnel) of the Railway Technical Research Institute, wind tunnel tests were carried out (Fig. 5). Wind speed was gradually increased, and the change in the relationship between wind speed and wheel load was measured. Each wheel load of the train model was measured by load cell inserted into the rail right under each wheel. For the simulation, the specified wheel load imbalance was selected and set by taking into consideration the thickness of the spacers (adjusting plates) in order to set the height of the axle and bolster springs. In an actual train vehicle or the above-mentioned train model, it is conceivable that static wheel load imbalance would occur due to torsion or asymmetry of the car body, nonuniformity of springs, etc. However, in the research reported in this paper, the influence of each of these components was considered to be equivalent by adjusting the thickness of the respective spacers. Table 1 shows the measured values of static wheel load and static wheel load imbalance in the model train, in which the axle numbers are counted starting from the front axle, and the front, rear, left and right are termed based on the forward direction of the car (the end panel on one side of the train set by taking into consideration the thickness of the spacers). In the wind tunnel test, the wind blows onto the “right” side indicated in Table 1, and in the case where the wheel load on the “right” side is light, the wheel load imbalance is indicated by a plus value. Incidentally, for actual railway vehicles in operation, the target service value for static wheel load imbalance is often set at 10% to 15%, so the large wheel load imbalances seen in the axle No. 3 and the axle No. 4 of the train model shown in Table 1 were not selected specifically.
in view of actual circumstances. In the simulation, a static wheel load imbalance almost equal to that of the train model was set by adjusting the thickness of spacers. Table 1 also shows the set values of static wheel load for the simulation.

Figure 6 (a) shows the dependence of the wheel unloading ratio on the square of wind speed in the wind tunnel tests. In the tests, it was confirmed that the vehicle model would not overturn until the wheel unloading ratios of all wheelsets reached 100%. It was further confirmed that the increase (gradient of graph) in the wheel unloading ratio averaged through all the wheelsets of the vehicle against the square of the wind speed remained almost constant until overturning occurred. The averaged wheel unloading ratio referred to herein is not the arithmetic mean value for the wheel unloading ratio of each wheelset but the wheel unloading ratio obtained from the arithmetic mean values for the right-side wheel load and those for the left-side of a whole vehicle. In the simulation as well, as shown in Fig. 6 (b), trends similar to those seen in the wind tunnel test were noted with respect to the averaged wheel unloading ratio. Thus, the simulation in the research work reported in this paper is generally considered an appropriate approach to evaluating running safety against overturning at the time when static wheel load imbalance occurs in the wheelsets.

3. Influence of Factors Not Considered in the Half-Vehicle Model on Overturning

3.1 Yawing Moment as Influential Factor

To investigate the dependence of wheel unloading ratio on yawing moment, a simulation was carried out presuming a continuous wave of lateral force as shown in Fig. 3 (a), with the front and rear lateral forces shifted in phases of a 1/16 cycle from 0 up to a 1/2 cycle. The fluctuating continuous wave frequencies used were 0.1, 0.3, 0.5 and 1.0 Hz. An example of input waveforms used in the simulation and the relationship between phase difference and maximum wheel unloading ratio for each frequency are shown in Fig. 7. From Fig. 7 (b), it was noted that the wheel unloading ratio reached the maximum value when the front and rear lateral forces were in phase for all frequencies. It was also found out that the wheel unloading ratio was easily affected by a phase difference when fluctuating lateral force frequencies were close to the natural frequency of lower-center rolling of the car (approximately 0.6 Hz for the car used in this test), but less affected for other frequencies. It was also noted that when the phase difference was π (opposite phase), the wheel unloading ratios were almost the same regardless of the lateral force frequency.

Next, a simplified model shown in Fig. 8 (a) is considered as a model simulating sudden exposure to wind. The aerodynamic force applied to the front of the vehicle rises sinusoidally for the duration of \( t_1 \), and its magnitude is maintained as constant once it reaches its peak. On the other hand, aerodynamic force to the rear rises after time \( t_2 \) from when it increases at the front and its waveform is the same as that of the aerodynamic force in the front position. The time \( t_1 \) is set at 2, 1, 0.5 and 0.3 seconds. The time lag \( t_2 \) is set at intervals of 0.2 \( t_1 \) from 0 up to \( t_1 \). In other words, \( t_2 = 0 \) corresponds to the case where lateral forces at the front and the rear increase in phase with each other, whereas \( t_2 = t_1 \) corresponds to aerodynamic force...
starting to increase at the rear after aerodynamic force at the front is saturated. The latter case is a simplified calculation model imitating the exit of a tunnel. It implies that the shorter the time $t_1$, the faster the vehicle runs into a strong wind section from the tunnel exit. To take a vehicle of 20 m in length as an example, if the acting points of the aerodynamic force in the front and rear positions are presumed to be the centers of the halves of the car body, the above-mentioned times correspond to a train velocity of approximately 18, 36, 72 and 120 km/h, respectively. The relationship between the time lag $t_2$ and the maximum wheel unloading ratio for each time period is illustrated in Fig. 8 (b), which shows that although there are some variations, the wheel unloading ratio nearly reaches its peak when the front and rear lateral forces rise in phase with each other.

Consequently, with regards to lateral force yawing moment influence on the wheel unloading ratio, it has been confirmed that when the front and rear lateral forces act in phase with each other, that is, when the yawing moment is zero, the wheel unloading ratio reaches the maximum value.

### 3.2 Influence of Static Wheel Load Imbalance

This section examines the influence of static wheel load imbalance on the estimation of the wheel unloading ratio (or critical wind speed of overturning). The half-vehicle model simulations described so far have not taken into account the influence of static load imbalance in each wheelset; therefore the wheel unloading ratio of the vehicle was uniquely determined against wind speed. In practice, however, all wheelsets of a vehicle have some static wheel load imbalance variability, and so as shown in Fig. 6, the wheel unloading ratio against wind speed varies for each axle. Accordingly, a simulation to examine the dependence of the wheel unloading ratio (or critical wind speed) on static wheel load imbalance was conducted, with the following results.

Using the same vehicle data corresponding to the full scale simulation given in Table 1, critical wind speeds of overturning were calculated varying the static wheel load imbalance of each wheelset. For the calculation, variations in static wheel load imbalance were predetermined by changing the position of the spacers inset at one or two places in the axle or bolster spring, or both. Also, since the overturning phenomenon was the subject of this research, based on relevant conclusions from the earlier mentioned wind tunnel test, the wind speed at which the wheel unloading ratio in all wheelsets became 100% was taken as the critical wind speed of overturning. Table 2 shows the inset position of the spacer, the variations in static wheel load imbalance and the calculated results of critical wind speed of overturning. In the simulation, only the lateral force was taken into consideration, with the lateral force coefficient of 1.2. Also, the selected spacer thicknesses for...
the axle and bolster springs were 10 and 20 mm, respectively. Furthermore, since the front and rear of the vehicle were symmetrical, inset axle spring spacers were only fitted to axles No. 1 and 2.

Table 2 reveals that where the values of the static wheel load imbalance averaged through the vehicle were almost equal, the critical wind speed of overturning were also almost the same despite differences in the static wheel load imbalance of each wheelset. Following on, in such a case, the relationship between the square of wind speed and wheel unloading ratio averaged through the vehicle were nearly identical. Figure 9 shows typical examples of the relationship between the square of the wind speed and averaged wheel unloading ratio in cases where the averaged static wheel load imbalances were nearly equal (case 2, 4, 8, 10, in all of which the averaged static wheel load imbalance was -0.2 or -0.3%). As Fig. 9 shows, the relationship between the square of wind speed and wheel unloading ratio averaged through the vehicle is nearly identical regardless of the difference of the static wheel load imbalance of each wheelset. This means that, by using the wheel unloading ratio averaged through the vehicle as an indicator, it is possible to determine the relationship between the wind speed and wheel unloading ratio up to the point of overturning independently.

As discussed above, concerning the influence of static wheel load imbalance, it was found that, regardless of wheelsets having different static wheel load imbalances, the relationship between wind speed and wheel unloading ratio or vehicle overturning can be appropriately evaluated using the wheel unloading ratio averaged through the vehicle as an indicator.

4. Conclusions

In order to increase the accuracy of estimating the critical wind speed of overturning, the full-vehicle model simulation program was constructed and its validity was verified. In addition, investigations were made into the wind speeds and for analyzing the behavior of a railway vehicle under particular wind conditions.

Table 2 Examples of static wheel load imbalance

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<th>Bogie 1</th>
<th>Bogie 2</th>
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Fig. 9  Relationship between square of wind speed and averaged wheel unloading ratio

As a result of verification tests using an actual car and a reduced model, the validity of the above-mentioned approach was confirmed. This enables further detailed analysis of the behavior of a vehicle subjected to crosswind, paving the way to quantitatively evaluating the difference between the verified approach and the former analysis based on the half-vehicle model. Also, investigations by the verified approach revealed that even the half-vehicle model is appropriate for a reasonable evaluation of safety against ordinary crosswind because the half-vehicle model signifies the severest condition concerning the influence of yawing moment, that is, it is on the safe side, against vehicle overturning. In addition, regarding the influence of static wheel load imbalance, the full-vehicle model is equivalent to the half-vehicle one, if the static wheel load imbalance averaged though the vehicle is considered as an indicator. Thus, the significant point of the approach discussed in this paper is that it quantitatively shows the reasonableness of the former half-vehicle model. Also, this approach is expected to be a useful tool for detailed investigations on the cause of overturning accidents induced by strong crosswinds and for analyzing the behavior of a railway vehicle under particular wind conditions.

References