Braking Force Estimation of Each Car

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Generally braking performance of train sets is evaluated by stopping distance or deceleration. However, stopping distance and deceleration are performance indices established for a whole train set, and do not represent performance of each car individually. For more detailed analysis of detail braking performance, it is useful to have the braking force in each car. This paper describes a method designed to estimate the braking force in each car based on the acceleration and coupler force, by using coupling devices as force sensors. Running tests results on an 18-car freight train using the devised method showed that it was possible to estimate the braking force of each car and that the estimated value was close to the theoretical value.

Keywords: braking performance, stopping distance, deceleration, braking force, coupler force

1. Introduction

Brakes are extremely important devices used for decelerating and halting trains, and their performance must therefore be fully understood. Braking performance, frequently evaluated through stopping distance and deceleration, is typically judged in the case of conventional railway lines from the distance required for trains to brake for an emergency, from the maximum commercial running speed. The procedure used to test braking performance also reflects braking performance during operation. The method also saves time and effort, because it measures the single-axis velocity of trains. Given these advantages, this method could be considered to be a fundamental method for measuring braking performance.

Stopping distance and deceleration however only give an indication of braking performance for a whole train set, and do not give any insight into the braking of individual vehicles. If the final braking distance is the same for two cases being compared, it is impossible to tell if deceleration was gradual and smooth throughout the train set in one case or if there was a sudden drop in one vehicle in the other. Whereas in reality, train sets have varying characteristics and contain different vehicle types. Therefore, in order to further improve braking performance and reliability, it is critical to be able to get a more detailed picture of actual braking performance, based on individual vehicles.

As part of efforts for this problem, the following methodologies are tried out: determining one-wheel or one-axle braking force by measuring the forces acting on the brake shoe hanger in the foundation brake [1,2], and calculating one-bogie braking forces by focusing attention on the tractive force acting on a single link connecting the vehicle and the bogie [3].

This study offers a procedure to estimate the braking force on an arbitrary vehicle in a train set by applying a vehicle coupler as a sensor using the force acting on it (coupler force, hereinafter) and the vehicle acceleration [4]. This paper first proposes a dynamic model to estimate the braking force from the coupler force and acceleration. Then, test results from applying this procedure to actual freight trains in operation are addressed.

2. Braking force acting on each vehicle in a train set

In a train comprising \( n \) vehicles, a model diagram of the forces acting on each decelerating vehicle in consideration of only the longitudinal movement is sketched in Fig. 1 (a) with symbol definitions listed in Table 1. The kinetic equation with multiple-degree of freedom systems connected by springs and dampers is defined as (1).

\[
M \ddot{x} + C \dot{x} + K x = f
\]  

(1)

Where, \( M \) is a mass matrix in response to each vehicle, \( C \) attenuation matrix of dampers, \( K \) rigidity matrix of springs, \( \dot{x}, x, \) and \( x \) are vectors respectively of deceleration, velocity, and displacement. \( f \) is an external-force vector, showing here a braking force acting on each vehicle.

Each vehicle in a train set individually exerts a braking force to decelerate itself and transmits the force via the buffer and coupler to the neighboring vehicles. While springs and dampers act on buffer rubbers popularly used, springs particularly exhibit non-linear rigidity that gradu-

<table>
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<th>Table 1 Symbol definition</th>
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<td>Symbol</td>
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<tr>
<td>( m, M )</td>
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<tr>
<td>( x )</td>
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<tr>
<td>( k )</td>
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<tr>
<td>( c )</td>
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<td>( f, F )</td>
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<td>( R )</td>
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<td>( \alpha )</td>
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ally increases with displacements. In order to handle the braking force for each vehicle as a dynamic model with multiple degree of freedom, a precise model capable of describing it and numerical simulation are required.

References [5-8] have been reported as the research handling these issues which employ a model addressing translation and rotation of wheelsets, bogies, and vehicles after precise consideration of the non-linearity of springs and accompanying components. These studies, aimed at buckling analysis and reduction of maximum coupler forces, have collected many findings on the coupler-force analyses that simulate the system motion from an external force (coupler force) as an input, by comparing the coupler-force simulations with actual measurements. Against formal-order analyses that simulate the system motion from an external force in a kinetic equation, however, the reverse analysis estimating the input from actual measurements like this study needs pre-known parameters and movements to solve it. The output however may be misleading, depending on the measurement results (3 requirements must be satisfied: existence, uniqueness, and continuity of solution). As it is not easy through such an approach to estimate the braking force on each vehicle, a procedure compatible for both practical use and precision is preferable.

Model simplification as shown in Fig. 1 (b) is then considered on the following premises. If the braking force acting on each vehicle has steady values (the values themselves for each vehicle can differ), the relative movement of each vehicle should converge once a certain period passes, physically resulting in its equal acceleration. Defining the acceleration of the whole train set as $\alpha$ and further assigning the sum of matrix terms of attenuation and rigidity as $R$ (coupler force), the braking force at each vehicle is expressed as (2).

$$f_i = \begin{cases} -m_i \alpha - R_k & (k = 1) \\ -m_i \alpha + R_{k-1} - R_k & (1 < k < n) \\ -m_i \alpha + R_{n-1} & (k = n) \end{cases} \quad (2)$$

Where, Suff. $k$ and $n$ are natural numbers satisfying $1 \leq k \leq n$, and (2) implies head, intermediate, and tail vehicles in top-to-bottom order. The acceleration $\alpha$ takes a negative value as a deceleration. When the train runs through an inclined section, a slope-converted value should be assigned as $\alpha$ for the section corresponding to horizontal running. When running resistance is in consideration, the running resistance corresponding to the vehicle running should be deducted from the right member of (2).

A freight train comprising a locomotive positioned at the head and other $j$ freight cars as shown in Fig. 1 (c) is considered here. Renaming the locomotive mass and braking force as $M$ and $F$ to distinguish the symbol of the locomotive from those for freight cars, it is expressed as (3), (4).

$$F = -M \alpha - R_i \quad (3)$$

$$f_i = \begin{cases} -m_i \alpha + R_{i-1} - R_i & (1 \leq i < j) \\ -m_i \alpha + R_i & (i = j) \end{cases} \quad (4)$$

Meanwhile, taking the grand sum of the equations for all freight cars, (5) is derived from $\Sigma f$ (total braking force)
and $\Sigma m$ (total mass of freight cars).

$$\sum_{i=1}^{n} f_i = - (\sum_{i=1}^{n} m_i \alpha + R_i)$$

(5)

The above tells that the braking force for the locomotive is determined from its mass, acceleration, and the neighboring coupler force, similar for the total braking forces of freight cars. It is also found that the braking force on an intermediate vehicle is calculated from its mass, acceleration, and its neighboring two coupler forces.

3. Verification through running tests

3.1 Test process

Using the test trains (Fig. 2) consisting of a locomotive (DF200 type) and container-freight cars (104 and 106 types), running tests were conducted. Figure 3 gives an outline. A 6-vehicle train consisting of 1 locomotive and 5 freight cars was first tested to assess the applicability of this procedure. Then, using an 18-vehicle train (1 locomotive + 17 freight cars), estimation of breaking forces with this procedure was studied in comparison with theoretical data and an estimation was made of the braking force of an arbitrary freight car. The mass of each car was adjusted to simulate full-loaded conditions, by using dead weights. Coupler forces (+ for tensile and – for compressive) were measured by equipping the couplers with strain gauges. Velocity was calculated from the wheel revolutions of the locomotive. Acceleration was measured with sensors which mounted on the locomotive and the first freight car (right behind the locomotive). Braking force was estimated for the locomotive using its own acceleration and for freight cars by that of the first freight car.

3.2 Test with 1 locomotive + 5 freight cars

3.2.1 When only locomotive brakes are applied

Figure 4 shows results for velocity, acceleration, coupler force, and estimated braking force when applying 2-notch braking only on the locomotive (independent brake, hereinafter) at time 0 s from the initial velocity of 25 km/h. This operation, causing only the locomotive to decelerate, makes the following freight cars push the locomotive. A compressive force is therefore exerted on the coupler between the locomotive and freight cars. The coupler force actually measured exhibits this trend, showing that a com-

![Fig. 2 Test train](image)

![Fig. 3 Outline of test train](image)
pressive force impacts right after the brake is applied and vibrations gradually fall to a stable state over a certain period. In this status with stable coupler forces, the accelerations of the locomotive and freight cars almost coincide with each other, showing the convergence of relative movement of the vehicles.

Estimated braking force, indicating steady values at the locomotive from startup to the end, stays roughly level at 0 kN after a certain amount of time despite vibrational movement occurring immediately after the brake is applied. This result coincides with braking only being applied to the locomotive. While the estimated braking force in the locomotive shows stable values even if the coupler force stays in a transient response region because it is canceled out by its own transient accelerative response. However, the estimated braking force totaled over the freight cars, which represents the relative motion of the 5 cars only by 1-car acceleration, exhibits the errors reflecting relative motion between cars before they become stationary and also indicates reasonable values after full convergence.

### 3.2.2 When both locomotive and freight cars are braked

Figure 5 shows a result of velocity, acceleration, coupler force, and estimated braking force when 3-notch braking is applied to both the locomotive and freight cars (continuous brake, hereinafter) at time 0 s from the initial velocity of 25 km/h. Continuous brake is regularly used while running between stations. This operation is programmed so as to roughly equalize the decelerations between the locomotive and freight cars. Deleting the acceleration $\alpha$ from (3) and (5) to sort them by $R_1$, (6) is given.

$$R_i = \frac{M \sum f_i (\sum m_i) F}{M + \sum m_i} \quad (6)$$
If the masses of locomotive and freight cars and braking forces are controlled in adequate proportion in (6), the numerator of the right member approaches 0, indicating that there is no large force acting on the coupler. The coupler forces actually measured, exhibiting this tendency, transit here a moderate path almost without forces except a few instantaneous compressive forces. These spiky responses, appearing when there is a constant difference of accelerations between locomotive and freight cars (particularly appearing frequently for large opposite peaks at the same time), is deemed to be indicative of impacts due to relative movements.

Steady-state values are observed here for the estimated braking force of the locomotive similar to when independent braking is applied. For independent 2-notch and continuous 3-notch braking, the brake-cylinder pressure (BC pressure, hereinafter) is almost evenly set. Judging from the values of the both estimated braking forces almost equivalent at about 60 kN, it is found here that the braking forces of the locomotive can be detected separately from those of freight cars.

Meanwhile, the freight-totaled braking force also indicates comparatively large fluctuations although it stays almost at a constant value except for the region right after braking begins. These fluctuations, coinciding with the timing of the spiky responses of the coupler force, are considered to be errors caused by the relative movement of vehicles, similar to what occurred with independent braking. While the freight cars in independent braking are relatively stable with only coupler forces being exerted, braking forces individually act on each freight car for continuous braking. The braking force exerted on each vehicle further exhibits a transmission delay of the braking commands and unevenness in friction between wheels and brake shoes. From these factors, continuous braking is prone to relative movements among vehicles, probably affecting the estimated braking force.

### 3.2.3 When freight-car braking condition is intentionally varied

Transitions in estimated braking forces were observed for continuous 3-notch braking when braking of freight cars was intentionally varied. The braking condition of freight cars is altered here by turning the manually operated valves for each bogie on and off. Table 2 shows the test conditions and Fig. 6 the test results. As stated in Section 3.2.1, estimated braking forces for freight cars during continuous braking are prone to fluctuations in terms of vibrational values. Therefore, the dead time and transient-response characteristics of BC pressure have been considered here. In other words, the estimated braking force in each condition is calculated by taking a time-averaged value over a running section required for stopping after the BC pressure reaches 63.2% of a steady value and further by taking a frequency-averaged value over the test with same conditions.

The calculated values exhibited a trend in proportion to the number of vehicles with brakes enabled. Each of the plots shows a high linearity, thereby demonstrating that the braking force estimated using this procedure possesses a resolution at least equivalent to 1 bogie for continuous 3-notch braking.

### 3.3 Test with 1 locomotive + 17 freight cars

#### 3.3.1 Comparison between estimated and theoretical values

Figure 7 (a) shows total braking forces estimated and those with theoretical values for 17 freight cars when the train emergency brakes at time 0 s from the initial velocity of 75 km/h. The theoretical value here is a braking force that is expected value at the design phase, defined as (7) for 1 freight car and as (8) for all freight cars (totaled).

\[
\bar{f}_i = \tau \cdot k \cdot \eta \cdot S \cdot P \cdot \mu
\]

Table 2 Braking condition for freight cars

<table>
<thead>
<tr>
<th>Number of vehicles with brakes enabled</th>
<th>Braking condition for freight cars</th>
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<tbody>
<tr>
<td>5</td>
<td>Brake on for every freight car</td>
</tr>
<tr>
<td>4</td>
<td>Brake off for 1 freight car</td>
</tr>
<tr>
<td>3</td>
<td>Brake off for 2 freight cars</td>
</tr>
<tr>
<td>2.5</td>
<td>Brake on for only one side of bogies for every freight car</td>
</tr>
</tbody>
</table>
\[ f_j = \sum_{\tau} j \cdot \kappa \cdot \eta \cdot S \cdot P \cdot \mu \]  

(8)

Where, \( \tau \): Number of brake cylinders per 1 bogie, \( \kappa \): Lever ratio, \( \eta \): Mechanical efficiency, \( S \): Brake-cylinder area, \( P \): BC pressure, \( \mu \): Friction coefficient between wheels and brake shoes. \( \tau, \kappa, \) and \( S \) are constants inherent in vehicle parameters, \( \eta \) is an approximate value obtained from preliminary study, \( P \) is a value actually measured at No. 1 freight-car which is applied to all cars.

Friction coefficient between wheels and brake shoes, generally depending on velocity, pressing force, temperature, dry-wet condition and so forth, is approximated as a function of velocity.

The results show that the theoretical values grow with the passage of time. The estimated braking forces totaled over the all freight cars exhibit a fairly good result following the theoretical data curve on average although indicating some vibrational behavior. It is thought that amplification is large compared to that in Section 3.2.2 because of the increase in braking force due to added vehicles and emergency braking.

3.3.2 Braking force of an arbitrary vehicle

The car subject to this procedure is can be selected randomly. For example, when the first freight car is selected:

\[ f_1 = -m_1 \alpha + R_0 - R_2 \]  

(9)

is derived from (4), which can be used to estimate the braking force of this vehicle if its acceleration and neighboring 2 coupler forces are obtained. In the same test as Fig. 7 (a), the estimated braking force and theoretical value for the first freight car are shown in Fig. 7 (b).

Observations from this test produce the same outcome as Fig. 7 (a), i.e. that the estimated braking force follows the theoretical value while showing some vibrational behavior. Therefore, this procedure can be applied to any car in the train set. Verifications were then made to see if this method could be applied to estimate the braking force for each vehicle in a train set.

4. Conclusions

In this study, a procedure was developed to estimate the braking force acting on each vehicle in a train set using coupler forces and accelerations, based on the assumption that the relative movements among vehicles converge. By applying this process to running tests using a freight train, it was verified that this procedure can be used to estimate any braking force in a train set with to a reasonable degree of accuracy and that the brake force estimation exhibits a trend following the theoretical values except fluctuating components. While a large relative movement among vehicles may cause some errors, the only alternatives are measuring individually the accelerations of vehicles or averaging the results.

Further work about the problems on simplification of measurements and improvement of estimation precision require studies to make this procedure more practical.

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References


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