Analytical Study of Structural Member Vibration Characteristics of Reinforced Concrete Rigid Frame Viaduct

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In order to analyze effectively the response of the members of a railway reinforced concrete rigid frame viaduct to the vibration, a new analysis method was developed. It divides the whole railway system into a vehicle / track model, and a track / structure model. These models were used to examine the influence of various vehicle, track and structure parameters on structural member vibration. The dominant factors were quantified for each frequency concerning the response of the center slab at a train speed of 270 km/h.

Keywords: structural member vibration, structure borne sound, high speed train, interaction

1. Introduction

Structure borne sound generated from concrete rigid frame viaducts or other types of concrete bridges is comprise mainly of frequency components in the range between several tens of Hz to 1000 Hz, and in the case of a high-speed railways the dominant frequency zone lies below 200 Hz [1].

A significant body of research has already been carried out on structure borne sound. However, regarding rigid frame viaducts made of steel reinforced concrete (hereafter referred to as "RC") and their structural members, there have been few cases where structure borne sound has caused an environmental problem, accordingly, no systematic studies have been carried out to date. However, in recent years, the increase in the speed of trains has brought about concerns that RC bridges, which hitherto were not considered a major source of noise, are also likely to generate structure borne sound to be reckoned with due to the increase in the dynamic response of the members.

In order to develop a method of predicting structure borne sound, the authors intend to use a coupling method consisting of structural analysis employing a finite element method and also acoustic analysis employing a boundary element method. In this method, the vibration velocity of the structure is obtained using a finite element method that takes into account the dynamic interaction between the vehicle, track and structure; then the results are used as the input conditions for acoustic analysis, by using the boundary element method to analyze the propagation of sound. The advantage of this method is to permit numerical experiments in which various parameters that exist in the propagation system for vibration and sound are incorporated.

In this research, in order to express the motion of vehicles and the issue of non-stationary and non-linear coupled vibration of vehicles and structures, a numerical analysis method is used in which a mode-converted equation of motion is solved using the Newmark method. In this case, however, the study must deal with not only a frequency region of the design of structure but also higher frequency regions. In other words, it must be extended to the high order vibration modes of the member. In addition, it is necessary to appropriately select the vibration modes to be taken into account, the element division and the time step.

Up to now the authors have developed a numerical analysis model for vibration up to about 200 Hz generated from the RC rigid frame viaduct which is the cause of structure borne sound, and have carried out studies focusing on the various parameters of the structure in the overall system composed of vehicle/track/structure [2]. In this research, in the coupling method consisting of the finite element method and the boundary element method, which is the final goal of the research, structural analysis using the finite element method was selected as the target, with special attention paid to the following points.

(1) In order to efficiently evaluate the effect of the various parameters in the vibration propagation system from the excitation source to the structure borne sound radiation surface, the excitation forces are obtained using the vehicle/track system model. The result is then input to the track/structure system model, and an analysis method of the vibration of the structural members is newly created, aiming at more efficient analysis.

(2) The phenomenon of vibration up to 200 Hz concerning the response to each member when a high-speed train is passing is elucidated by using the above model.

(3) In addition, the effect of the various vehicle and track parameters on the vibration of the structural members is quantitatively evaluated.

2. Research method

2.1 Analysis method

Figure 1 shows the outline of the structure, and Table 1 shows the material properties for each element. The structure consists of a standard 3-span RC rigid frame viaduct with a block length of 25 m and adjustable girders of 10 m span for both ends.
Table 1 Material properties

| Material      | Property                        | Value |
|---------------|---------------------------------|-------|
| Rail          | Material constant               | 60kg  |
| Track slab    | Dimension (mm)                  | 4930 × 2340 × 190 |
|               | Young modulus (kN/mm²)          | 31    |
| CA mortar     | Young modulus (kN/mm²)          | 3.5   |
|               | Thickness (mm)                  | 25    |
| RC viaduct    | Young modulus (kN/mm²)          | 26.5  |
| Girder concrete| Young modulus (kN/mm²)          | 25    |
| Damping constant (all modes) |                                | 2%    |

2.2 Dynamic model of vehicle

Figure 3 shows a dynamic model of the vehicle, and Figure 4 shows an outline of the axle arrangement of the vehicle. It is assumed that the body, bogies and wheelsets
Wheelset is 1158 and the total number of elements is 1446, while in vehicle/track system model, the total number of nodal points \( m \), which is 1/4 of the rail fastening interval (0.625 m). In the model, as shown in Fig. 3, the rail and track slabs were modeled using beam elements, and the track pads and the concrete asphalt (hereafter referred to as "CA") mortar beneath the track slabs were modeled using spring elements. The spring constant of the rail pad used in this analysis was set to 4096 points (observation time, approx. 0.4 s) in the analysis was set to a value that enabled the equation of motion to be analyzed. However, it is considered that the excitation force generated by a track irregularity with a wavelength in the order of several meters will not be reproduced sufficiently. Consequently, in the discrete models, a newly measured 10 m wavelength irregularity was added to the short wavelength irregularity which was measured, not on the structure to be analyzed, but on the slab track of a meter-gauge railway line. In this way a long wavelength component was added (hereafter referred to as "medium wavelength irregularity") to the short wavelength irregularity which was measured, not on the structure to be analyzed, but on the slab track of a meter-gauge railway line. In this way a long wavelength component was added (hereafter referred to as "medium wavelength irregularity"). The pseudo peaks due to the measurement wavelength were removed using a filter.

2.4 Dynamic model of interaction force between the wheel and rails

The dynamic interaction forces between the wheels and the rails are calculated after obtaining the contact points and the contact angles from the geometrical shape of both the wheels and the rails and also their relative displacements. Concretely, the contact force in the perpendicular direction is represented by a Hertz spring. The contact force in the horizontal direction is expressed as a creep force until the wheel flange touches the rail. After the contact, the wheel load and horizontal pressure act on the rail, causing the rail crown to move in the horizontal direction. As a result, torsion of the rail occurs. The torsion resistance which is generated by the rail and the rail fastener is expressed by a spring element.

Figure 5 shows the track irregularity which was used for analysis. In the integrated model, a 2 m length of irregularity (hereafter referred to as "short wavelength irregularity") measured using a measuring device with a length of 1 m was set consecutively to the slab track on the structure to be analyzed. However, it is considered that the excitation force generated by a track irregularity with a wavelength in the order of several meters will not be reproduced sufficiently. Consequently, in the discrete models, a newly measured 10 m wavelength irregularity was added to the short wavelength irregularity which was measured, not on the structure to be analyzed, but on the slab track of a meter-gauge railway line. In this way a long wavelength component was added (hereafter referred to as "medium wavelength irregularity"). The pseudo peaks due to the measurement wavelength were removed using a filter.

2.5 Numerical analysis method

In order to carry out efficient numerical analysis, the equation of motion concerning the vehicle, track and structure was modal-converted. The resulting equation of motion on the modal coordinate system of the vehicle and structure was progressively solved in time increment \( \Delta t \) units by using the Newmark mean acceleration method. However iterative calculation has to be carried out within \( \Delta t \) as long as the disproportional part becomes sufficiently small, because the equation of motion is non-linear. The mode order in the analysis was set to a value that enabled vibration up to about 400 Hz to be reproduced, and the analysis time step was set to 0.0005 sec. The frequency analysis was set to 4096 points (observation time, approx. 2 seconds, \( \Delta t = 0.49 \) Hz).
2.6 Analysis cases

Table 2 shows analysis cases. Special attention was paid to the various vehicle and track parameters. CASE 1 is the basic case. Because each parameter is affected by the train speed, calculations for each increment of 10 km/h were made in the range between 160 and 370 km/h.

In CASE 2-1, the vehicle was represented not by the dynamic model with 31 degrees of freedom shown in Fig. 4, but rather by a series of loads of constant force equivalent to the static wheel load, in order to study the effect of the vibration system of the vehicle. In CASE 2-2, the mass of each of the bodies, bogies, and wheelsets was uniformly reduced by 30%, and the spring constant was left unchanged, in order to study the effect of the mass of the vehicle. In CASE 2-3, the vehicle length was changed, in order to study the effect of the wheel layout. The vehicle length was changed so that the excitation pitch of the two axles of the front bogie of the carriage concerned and the excitation pitch of the two axles of the rear bogie of the same carriage were in opposite phases.

In CASE 3-1, the vehicle runs on smooth rails without irregularity, in order to study the effect of rail irregularity. In CASE 3-2, the spring constant of the track pad was reduced to half of the basic case, in order to study the effect of applying low spring constant. In CASE 3-3, the normal rail fastening interval of 0.625 m was changed to a continuous support, in order to study the effect of rail fastening intervals.

Table 2 Analysis cases

| CASE | Parameter     | Notes                        |
|------|--------------|------------------------------|
| 1    | -            | Basic case                   |
| 2-1  | Vehicle model| 31 DOF model → Constant load |
| 2-2  | Vehicle weight| 30% reduction               |
| 2-3  | Vehicle length| 25m → 23.75m                |
| 3-1  | Rail irregularity| Existent → Not existent |
| 3-2  | Rail pad stiffness| 60MN/m → 30MN/m         |
| 3-3  | Rail fastener interval| 0.625m → Continuous fastener |

2.7 Method for validating the analysis method

The appropriateness of the analysis model was validated by comparison of analysis results with actual measurements. The response acceleration during the passage of a train was measured using a piezo-electric type accelerometer (sensitivity: 6.42 pC/(m/s²); measurement frequency range: from 1Hz to 7kHz), and the data was recorded in a laptop computer via an AD converter board, with a sampling frequency of 2kHz. Frequency analysis was performed by FFT for a period of two seconds during train transit.

3. Analysis results from the basic case

3.1 Vibration mode

Figure 6 shows the vibration modes and natural frequencies of the structure. The natural vibration modes in the longitudinal direction and also in the transverse direction of the viaduct appear at 2.6 Hz and 2.7 Hz, respectively. Note, however, that in this model, the bottom end of the column is fixed, therefore if the ground, footing beam and foundation structure are appropriately modeled, the natural frequency may change. The first vibration mode of a cantilever slab appears at 11.1 Hz. There are also many modes in which cantilever slabs are coupled with center slabs or handrails. Concerning the vibration modes of a center slab, the primary mode appears at 21.0 Hz, and as the order of the mode increases, the number of nodes and antinodes increases. Reference [5] provides a detailed report concerning validation of these vibration modes by measurement. It was considered that analysis results agreed with measured results to some extent.

3.2 Acceleration response

Figure 7 compares measured and analysis results for the frequency analysis results of the response acceleration. The train speed was 270 km/h, and the reference points were at the positions shown in Fig. 2 (b). It was decided to compare measurements from about 10 trains (the number of trains differed for each member) with the analysis results, in consideration of random variations from one measurement to another. As seen from the above figure, peaks
appear at an integral multiple of the basic excitation frequency (3 Hz = 270/3.6/25) determined by the train speed and the carriage length.

When comparing the measurement and analysis results, the analysis results are roughly within the range of random variation of the measurement results for each member. When comparing the discrete models and the integrated model, the result of newly adding irregularity of a long wavelength component causes the response of the discrete models to become higher than that of the integrated model over the frequency band between roughly 20 Hz and 70 Hz.

Measured results are compared for a train speed of 270 km/h alone, therefore it is necessary to validate reproducibility for other speeds by accumulating data in the future. However, from the above-mentioned comparison, it is confirmed that this analysis method is appropriate to a certain extent, and as such the effects of the various parameters can be discussed in the following paragraphs.

4. Analysis results for various parameters

4.1 Effect of train speed in the basic case (CASE1)

4.1.1 Effect on excitation force

Figure 8 (a) shows the effect of the train speed on excitation force in the basic case. In the case where the spring elements are discretely distributed in the longitudinal track direction, one excitation force acting on the spring element located at the center of the model was taken up as the target excitation force. From the figure, peaks generated by a 2.5 m wheelbase shifted with increase of the train speed. On the other hand, there are also peaks that do not shift with increase of the train speed. For example, the peak in the vicinity of 84 Hz at a speed of about 190 km/h is generated as a result of the matching of the excitation frequency (84.4 Hz = (190/3.6)/0.625) of the mass spring system (consisting of the unsprung wheelset mass, rail mass and rail pad) with the natural frequency due to the reaction received by the unsprung wheelset mass at each rail fastening interval of 0.625 m, and the effect of the excitation force is considered to be small.

There are also peaks in the vicinity of 64 Hz and 96 Hz at a speed of about 290 km/h and in the vicinity of 80 Hz and 120 Hz at a speed of about 360 km/h; however comparing these cases with the results of CASE 3-1 (no irregularity, Fig. 8 (b)), it is considered that these peaks were caused by track irregularity.

4.1.2 Effect on response of members

Figure 9 shows the effect of differences in train speed on the response of each members.

Regarding the center slab, there are clear peaks in the vicinity of 26 Hz at a speed of 240 km/h, 64 Hz at a speed of 290 km/h, and 40 Hz at a speed of 360 km/h. These peaks are thought to be occurred because integral multiples of the excitation frequency (26.7 Hz at 240 km/h, 32.2 Hz at 290 km/h, and 40.0 Hz at 360 km/h) determined by each
train speed and a wheelbase of 2.5 m have approached the natural frequencies of the center slab (Fig. 6(e), (g), and (h)). The peak that appears in the vicinity of 84 Hz at a speed of 190 km/h is also the peak generated by an excitation force. This peak is generated as a result of approximate matching of the natural frequency of the mass spring system with the excitation frequency due to reaction received by the unsprung mass at each rail fastening interval.

Regarding the cantilever slab, there are clear peaks in the vicinity of 105 Hz at a speed of 200 km/h, 21 Hz at a speed of 240 km/h, 32 Hz at a speed of 290 km/h, and 11 Hz at a speed of 330 km/h. These peaks occur in the frequency bands corresponding to the characteristic modes of either the individual cantilever slab or the cantilever slab that is coupled to the center slab (Fig. 6 (c), (d), (g), and (l)). In contrast to the case of the center slab, excitation frequencies may not be necessarily in proximity to those determined by the train speed and the 2.5 m-long wheelbase (22.2 Hz at 200 km/h, 26.7 Hz at 240 km/h, 32.2 Hz at 290 km/h, and 36.7 Hz at 330 km/h). The peak in the vicinity of 11 Hz at a speed of approximately 330 km/h has a frequency of 11.1 Hz which is three times as high as the excitation frequency of 3.7 Hz determined by the train speed and the carriage length of 25 m; therefore, there is a possibility that the peak is excited by this excitation frequency. The vibration mode of the cantilever slab is a complicated natural vibration mode in which the cantilever slab is coupled to the center slab and the handrail. An evaluation for the mode with due consideration on the effect of other parameters will be performed in the future.

4.2 Effect of various parameters

Figure 10 shows the influences of the differences of various parameters on the acceleration response at the center slab.

In CASE2-1, the response was obtained by representing the vehicle not as a dynamic model with 31 degrees of freedom as shown in Fig. 4, but rather as a series of loads of constant force equivalent to the static wheel load. In the frequency band up to about 20 Hz, no difference in the response according to differences between models is found, thus indicating that the response is determined only by the mass of the carriage. On the other hand, in the high frequency region above 20 Hz, the response according to a series of the loads is smaller than that in the basic case, thus indicating the effect of the vibration system of the carriage. Generally, the natural frequency of the body is the order of several Hz, thus the effect of the unsprung wheelset mass is large in the vibration system of the vehicle in the high frequency region above 20 Hz.

In CASE2-2, the mass of each of the bodies, bogies and wheelsets are all reduced equally by about 30% compared to the basic case. The spring constant is left unchanged. The effect of the reduction of mass of the carriage is observed in the zone where the frequency is roughly less than 100 Hz and also in the zone where the frequency is higher than 180 Hz.

In CASE2-3, the peaks determined by the carriage length has shifted from 3 Hz of the basic case to 3.2 Hz
Fig. 11  The dominant factors for each frequency for the acceleration response of the center slab at a train speed of 270 km/h

5. Conclusion

The knowledge obtained in this research is summarized below.

(1) The vibration propagation system of the vehicle/track/structure can be divided into two systems, namely a vehicle/track system model and also a track/structure system model, and a new method was developed for efficiently analyzing the vibration of members up to about 200 Hz, which is the main cause of structure borne sound. Comparing the results of analysis using this method with the measured results confirmed that the results obtained using this method are more or less within the range of random variations of measured values.

(2) There are cases in which large peaks occur in cantilever slab or center slab responses, at specific speeds and frequency bands, with changes in train speed. The main factors responsible for the peak generation include the fact that the excitation frequency determined by the train speed, wheelbase and carriage length, approaches the natural frequency of the members of the structure, the existence of track irregularities, and the fact that the natural frequency of the spring mass system consisting of the sprung mass, the rail and the track pad approaches the excitation frequency due to the reaction received by the unsprung mass at each rail fastening interval of 0.625 m.

(3) Regarding the vehicle parameters, it was found that the effect of the unsprung mass is large in the frequency region above 20 Hz and the effect of the vehicle mass is large in the frequency region below 100 Hz and above 180 Hz, and that the axle arrangement affects the frequency at which peaks occurred.

(4) Regarding the parameters of the track, it was found that the effect of track irregularity is larger for frequencies between 20 Hz and 100 Hz and also above 150 Hz, the effect of track pad stiffness increases for frequencies above 30 Hz (between 30 Hz and 70 Hz, roughly the same or a slight decrease; above 70 Hz,
slight increase), and the effect of the rail fastening interval is significant for frequencies between 100 and 140 Hz. Figure 11 indicates the dominant factors for each frequency concerning the acceleration response of the center slab at a train speed of 270 km/h.

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