Analytical Study on Structure Member Vibration Characteristics of Reinforced Concrete Viaducts

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Abstract. In order to analyze effectively the vibration response of the structure members of railway reinforced concrete viaducts, we have developed a new analysis method. It divides the whole railway system into vehicles/track model, and the track/structure model. Using these models, we examined the influence of various parameters of the vehicle, track and structure on the structure member vibration. As a result, the following has become clear. For example, for 20Hz or more, unsprung wheelset mass has a great influence on response of structure member. For 20-100Hz and 150Hz or more, rail surface roughness, for 60Hz or more, the stiffness of the rail pad, for less than 60Hz, the stiffness of the slab of structure and so on, have great influence on structure member vibration respectively.

Keywords: Structure member vibration, Structure born sound, High speed train, interaction, reinforced concrete viaduct

1. Introduction

Structure born sound is generated as a result of a varying acting force (hereafter called an “excitation force”) generated by a railway train running on track irregularity of the order of several micrometers, always existing on the surfaces of the wheel treads and the rail heads, and also by track displacement that has a wavelength of the order of several meters in the direction of the railway track, causing the wheels and the rails to vibrate. This vibration is transmitted to the track structure consisting of sleepers and track slabs, and also to the civil engineering structure (girder bridges, viaducts, truss bridges and so on) that supports the track, causing sound to be radiated from the vibrating surfaces of the various members. Structure born sound generated from concrete rigid frame viaducts or other types of concrete bridges are mainly comprised by the frequency components in the range between several tens of Hz to 1000 Hz, and it is said that in the case of a high-speed railway the dominant zone in this frequency range lies below 200 Hz [1,2].
Many researches have already been carried out concerning structure born sound. However, regarding reinforced concrete (hereafter called “RC”) viaducts and their structure members, there are few cases where structure born sound raises an environment problem, accordingly no systematic studies have been carried out to date. However, in recent years, the drastic increase in the speed of trains has brought about concerns that RC bridges, which have been considered to generate little structure born sound, are also likely to generate structure born sound to be reckoned with due to the increase in the dynamic response of the members.

In order to develop a prediction method of structure born 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, as the final goal of this research. 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, and the boundary element method is used 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. For example, it is possible to predict the vibration and structure borne sound of the new and advanced vehicle, track and structure based on the numerical analysis. In addition, conventional prediction methods of the railway noise are mainly based on measurement data.

It is not easy to analytically calculate the vibration velocity of the structure member. 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 in which a mode-converted equation of motion is solved using the Newmark method is used. In this case, however, the study must deal with not only a frequency region of the design of structure but also higher region. In other words, it must be extended to the high order vibration mode of the member level. In addition, it is necessary to appropriately select the vibration modes to be taken into account, the element division and the time step.

Up until now the authors have developed a numerical analysis model for vibration up to about 200 Hz generated from an RC girder and RC rigid frame viaduct which is the cause of structure born sound, and have first carried out studies focusing on the various parameters of the structure in the overall system composed of vehicle/track/structure [3].

In this research, in the coupling method consisting of the finite element method and the boundary element method, which is the final goal of research, structural analysis using the finite element method was selected as the target, and we paid attention to the following points.

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

(2) To elucidate the phenomenon of vibration up to 200 Hz, concerning the response to each member when a high-speed train is passing, the above model is used.

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

2.1. Analysis Method

Figure 1 shows the outlines of the structure to be analyzed, and Table 1 shows material constants for each element. We conducted analyses of 2 types of structures. One is a standard 3-span RC rigid frame viaduct with a block length of 25 m and adjustable girders of 10 m spans for each end, the other is a standard 3-span RC girder with a length of 20m.

Figure 2 shows an outline of the analysis model. In order to adequately evaluate structure born sound, it is important to take into consideration high order vibration modes of the structure members. In addition, it is necessary to excite the unsprung mass of the vehicle that
constitutes the excitation source up to the high frequency region, and to reproduce the vibration to the high order mode of the rails the frequency of which exceeds 500 Hz. Until now, the entire system consisting of vehicle/track/structure has been modeled [3] (hereafter called an “integrated model”). We used the DIASTARSIII program for numerical analysis, which

(a) Vehicle/tracks model

(b) A standard 3-span RC rigid frame viaduct and adjustable girders

(c) A standard RC girders

Figure 2: Outlines of the analysis model
had been developed by the Railway Technical Research Institute [4], which analyzes the dynamic interaction between the vehicle and railway structure. However, there is a problem that most of the structure vibration modes (mainly lower than 100Hz) existed in order that were even lower than those of the rail vibration modes (mainly higher than several hundreds Hz), so if an attempt is made to reproduce the high order modes of the rails, the number of orders of the vibration modes to be taken into account become extremely large, necessitating a large time to carry out each analysis. Consequently, although the method used to study the effect of each parameter individually enables a solution to be obtained, it is not very practical.

Accordingly, we have focused on the difference in the dominant frequency band of the vibration mode of the track (mainly lower than 100Hz) and structure (mainly higher than several hundreds Hz), and we developed a new method in which the integrated model was divided the system into two parts, namely vehicle/track and track/structure (hereafter called “discrete models”), respectively. The vehicle/track model is used for analysis of the excitation force, and then input the excitation force to the track/structure model in order to analysis the vibration of the structure member. The transfer of the excitation between vehicle/track model and track/structure model is done using a constructed program which automates the pre-processing to the lines of excitation forces. Concretely, in the case of an integrated model, the vehicle/track system model of Figure 2(a) was constructed on the excitation input line (This is shown as “Lines of Excitation Forces” in the figures) in Figure 2(b) and Figure 2(c). In the case of discrete models, the system is separated into two parts on “Lines of Excitation Forces”. This enables the numbers of degrees of freedom of analysis to be greatly reduced compared to the case where the entire system is analyzed at one time. Also, compared to the integrated model, the discrete models enable the analysis time to be greatly reduced and also high order vibration modes of the rails to be reproduced. It is thus an efficient and practical analysis method. Concretely, analysis that takes about 240 hours when the integrated model is used can be performed in a total of less than 30 hours when the discrete models are used. In the latter case, the vehicle/track system model enables the excitation force to be calculated in about 24 hours, and the track/structure system model enables the excitation force to be calculated in about 2 hours.

In the case of discrete models, as is the case of integrated models, numerical analysis using the vehicle/track system model was carried out using the DIASTARSIII program, developed by the Railway Technical Research Institute [4,5], which analyzes dynamic interaction between the vehicle and railway structure. Also, numerical analysis using the track/structure system model was carried out using the DIARIST general purpose structure program for the track structure [6]. We describe the outlines of the theory of those programs, in the next section later.

In this research, we have focused on the difference in the dominant frequency band of the vibration mode of the track and structure, and newly constructed “discrete models”. As described above, the division of the overall system into two systems contributes to more efficient analysis, however there are concerns to the effect that it may not always be possible to reproduce the interaction that takes place in the entire system. However, when the response of a structure due to the passage of an actual train is measured, a certain amount of random variation exists. Details will be described later. Even though the overall system is divided, it was first confirmed that the random variations of the measurement results were within the specified range, and then we recognized the method was a practical method for studying the effect.
of a large number of parameters from a numerical analysis point of view. In this research, we decided to carry out a study using this method.

2.2. Dynamic Model of Vehicle

Figure 3 shows a dynamic model of vehicle, and Figure 4 shows an outline of the axle arrangement of the vehicle. It is assumed that the body, bogies and wheelset are rigid bodies. The model is a 3-dimensional model in which these rigid bodies are linked by springs $K_N$ and dampers $C_N$ (with $N$ as a suffix in Figure 3) according to their respective characteristics. Vehicles have 31 degrees of freedom (5 degrees of freedom for the body, 5 degrees of freedom for the bogies, and 4 degrees of freedom for the wheelset). A train is represented by multiple vehicle models linked together by springs $K_C$ and dampers $C_C$ attached to the ends of the vehicle models. Assuming that the train runs at a constant speed, the equation of 3D motion of train with $N$ vehicles connected is written in a familiar matrix form as reference [7].

$$M^V \ddot{X}^V + C^V \dot{X}^V + K^V X^V = F^V_L + F^V_I(X^V, X^B) + F^V_N(X^V)$$

(1)

where, affixing character $V$ and $B$ are the vehicle and the structure (=bridge), respectively; $X^V$ is a displacement vector of vehicle; $M^V$, $C^V$ and $K^V$ are the mass, damping and stiffness matrices of vehicle, respectively; $F^V_L$ is load vectors; $F^V_I(X^V, X^B)$ is the interaction force between the rail and the vehicle, $F^V_N(X^V)$ is the load vectors of non-linear spring force in the car, respectively. In this research, the train consisted of 6 standard Shinkansen cars, each with a length of 25 m and axle load of roughly 60 kN.

2.3. Dynamic Model of Track and Structure

The track and structure were modeled using the finite element method. Assembling all elements in the model, the equation of motion of the track and structure is obtained in a standard matrix form as

$$M^B \ddot{X}^B + C^B \dot{X}^B + K^B X^B = F^B_L + F^B_I(X^V, X^B) + F^B_N(X^B)$$

(2)
where $X^b$ is a displacement vector of the track and structure, and $M^b$, $C^b$ and $K^b$ are the mass, damping and stiffness matrices, respectively; $F_L^b$ is load vectors; $F_B^{\text{int}}(X^V, X^B)$ is the interaction force between the rail and the car, $F_N^b(X^B)$ is the load vectors of non-linear spring force in the structure, respectively.

In the case of the vehicle/track system model, as shown in Figure 2 (a) the rail and track slabs were modeled using beam elements, and the rail pads and the CA mortar beneath the track slabs were modeled using spring elements. In this model, the spring reaction force equivalent to the CA mortar was obtained, and the resulting value was input as the excitation force to the excitation force input line, taking into consideration the rail position of the track/structure system model and the load dispersion due to the roadbed concrete, for example. The spring constant of the rail pad used in actual analysis was set to three times the nominal value, while making reference to the measurement value of the track spring constant calculated from the results of measuring the displacement of the left and right rails and the axle load. In the case of the track/structure system model, as shown in Figure 2 (b) and Figure 2 (c) handrails, cantilever slabs, center slabs, beams and main girder were modeled using shell elements, columns and piers were using beam elements. Track members were added to slabs in consideration of weight alone. From a prior study, it was found that the effect on the response of a member in the frequency region (roughly above 20 Hz) that contributes to structure born sound was small, even when the footing beam or the ground was modeled. For this reason, it was decided to omit their modeling, and assume that the bottom end of the column is fixed.

In the case of both models, the basic mesh size was set to 0.15625 m, which is 1/4 of the rail fastening interval (0.625 m). In the case of the vehicle/track system model, the total number of nodal point is 1158 and the total number of elements is 1446, while in the case of the track/structure system model, the total number of nodal points of girders is 11448 and the total number of elements is 13776. And the total number of nodal points of rigid frame viaduct is 7869 and the total number of elements is 8030.

### 2.4. Dynamic Model of Interaction Force between Wheel and rail

Figure 5 shows a wheel/rail model. We focused on relative displacement between the wheel and the rail. The vertical interaction force of these components was modeled by Hertzian contact springs so that it can be possible to judge the contact condition between the wheel and rails. The vertical relative displacement between the rail and the wheel $\delta_Z$ is expressed by equation (3).

$$\delta_Z = z_R - z_W + e_Z + e_0(y)$$

where $z_R$ is the rail vertical displacement, $z_W$ is the wheel vertical displacement, $e_Z$ is the vertical track irregularity. $e_0(y)$ is the amount of change of the wheel radius at the current contact point from initial wheel radius. When $\delta_Z \geq 0$, the wheel is in contact with the rail, and $\delta_Z \leq 0$, the wheel is out of contact with the rail. The $z$-direction interaction force $H$ produced due to the contact of wheel and rail is expressed by equation (4).

$$H = H(\delta_Z)$$

The interaction force in the horizontal direction is expressed as a creep force until the wheel flange touches the rail. After 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 oc-
The torsion resistance which is generated by the rail and the rail fastener is expressed by a spring element. Figure 6 shows the track irregularity which was used for analysis. In the case of an integrated model, a 2 m wave length irregularity (hereafter called “short wavelength irregularity”) measured using a measuring device of length 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 displacement of the order of a wavelength of several meters will not be reproduced sufficiently. Consequently, in the case of discrete models, a newly measured 10 m wave length irregularity was added to the short wavelength irregularity measured on the slab track of a meter-gauge railway line, although it was not on the structure to be analyzed, thus a long wavelength component was added (hereafter called “medium wavelength irregularity”). The pseudo peaks due to the measurement wavelength was removed using a filter.

2.5. Numerical Analysis Method

In the previous section, the equation of motion of vehicle, track and structure is respectively derived in the vehicle’s coordinate system and global coordinate system respectively. The coupling of equations of motion of vehicle, track and structure are conducted by applying the compatibility and equilibrium conditions to the sub region where both systems are connected with each other. As the vehicle moves every moment, the compatibility condition is evaluated using a fictitious nodal point at the center between left and right rails where the wheelset is being contacted.

In order to carry out efficient numerical analysis, the equation of motion concerning the vehicle, track and structure shown as equation (1) and equation (2) was modal-converted. The resulting equation of motion on the modal coordinate system of the vehicle, track and structure was progressively solved in time increment Δt units by using the Newmark method. However, because the equation of motion is non-linear, iterative calculation has to be carried out within Δt until the disproportional part becomes sufficiently small. The mode order in the
analysis was set to a value that enabled vibration 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, Δf = 0.49 Hz).

2.6. Analysis Cases

Table 2 shows analysis cases. We carried out parameter studies for rigid frame viaduct. We paid attention to the various vehicle, track and structure parameters. CASE 1 is the basic case. Because each parameter is affected by the train speed, we calculated at 10 km/h intervals over the range between 160 and 370 km/h.

In the CASE 2, we examined parameters related to vehicle. In CASE 2-1, the vehicle was represented not by the dynamic model with 31 degrees of freedom shown in Figure 4, but rather by a series of load of constant force equivalent to the static wheel load (the load interval was the same as that of the vehicle axle arrangement), in order to study the effect of the vibration system of the vehicle. In CASE 2-2, the mass of each of the body, bogies, and wheelset 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 the CASE 3, we examined parameters related to the track. In CASE 3-1, the vehicle runs on smooth rails without applying the irregularity shown in Figure 5, in order to study the effect of rail irregularity. In CASE 3-2, the spring constant was reduced to half of the basic case, in order to study the effect of the applying of low spring constant springs to the rail pad. In CASE 3-3, the normal rail fastening interval of 0.625 m was changed to a continuously supported condition, in order to study of the effect of the rail fastening interval. Here, the spring elements were arranged at all nodal points on the beam element equivalent to the rails, in order to simulate a continuously supported condition. However, the spring constant of the track support spring per unit length was made the same as that of the basic case.

In the CASE 4, we examined parameters related to the structure. In CASE 4-1, the stiffness of the center slab was set to a value of five times the normal value, in order to study the
effect of the stiffness of the center slab. This value was set using the currently proposed center slab reinforcing method, based on the results of an actual bending stiffness test. In CASE 4-2, the effect of the difference of material constants such as the Young’s modulus of inclined concrete, non-structure members such as roadbed concrete, and concrete was studied. Regarding non-structure members, it is known that the difference of material constants affects the response of an actual structure and the natural frequency. Accordingly, it was assumed that the stiffness of structure member is increased by these effects, and the Young’s modulus of concrete was uniformly set to three times the design value, in order to take into account the apparent increase of stiffness. In CASE 4-3, the damping constant of the structure was set to 5%, in order to study the effect of damping of the structure. In this case, that of the vehicle/track system model remained 2%.

3. Analysis Result in Basic Case

3.1. Vibration Mode

Figure 7 shows the vibration modes and natural frequencies of the RC rigid frame viaduct. 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. However, that in this model, the bottom end of the column is fixed, then if the ground, footing beam and foundation structure are appropriately modeled, the natural frequency may change. The vibration mode of a cantilever slab appears at 11.1 Hz. There are also many modes in which cantilever slabs are coupled to 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 numbers of the node and the antinode increase.

Figure 8 shows the vibration modes and natural frequencies of the RC girder. The first vibration mode of girder appears at 7.1 Hz and that of cantilever slab appears at 10.1 Hz.

Reference [8] provides a detailed report concerning validation of these vibration modes by measurement. We considered that analysis results accorded with measurement to some extent.

3.2. Characteristics of Excitation Force

Figure 9 shows the results of time history wave and frequency analysis for the spring reaction force of a single spring element equivalent to CA mortar in a vehicle/track system model. The train speed is 270 km/h. From the time waveform, it can be seen that a response peak appears each time an axle passes the spring element. It can be seen that a peak occurs at an integral multiple of the basic acceleration frequency (3Hz = (270/3.6)/25) which is determined based on a train speed of 270 km/h and a carriage length of 25 m.

3.3. Frequency Characteristics of Each Structure Member

Figure 10 shows a comparison between measurement and analysis concerning the frequency analysis results of the response acceleration at the center slab and cantilever slab of RC rigid frame viaduct. The train speed is 270 km/h, and the reference points are the positions shown in Figure 2 (b). We decided to compare the results of measuring about 10 trains (the number
of trains differs for each member) with the analysis results, in consideration of random variations from one measurement to another.

Figure 11 shows a comparison of the frequency analysis results of the response acceleration at the center slab and cantilever slab of RC girder between measurement and analysis. The train speed is 270 km/h, and the reference points are shown in Figure 2 (c).

From the above figure, it can be seen that a peak appears at an integral multiple of the basic excitation frequency (3 Hz = 1/(25/270/3.6) determined by the train speed and the car-
When comparing 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 (= new model) and the integrated model (= conventional model) of RC rigid frame viaduct, by reproducibility of high order rail vibration modes and adding new irregularities of a long wavelength component, the response of the discrete models agree rather well than that of the integrated model over the frequency band between roughly 20 Hz and 150 Hz.

Measurement for comparison is performed only for a train speed of 270 km/h, therefore it is necessary to validate the 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 effect of the various parameters is to be discussed in the following paragraphs.

4. Analysis Results about Various Parameters
   
   We carried out parameter studies for RC rigid frame viaduct.
4.1. Effect of Train Speed in the Basic Case (CASE1)

(1) Effect on Excitation Force

Figure 12: Effect of the train speed on excitation force in the basic case

(a) Basic case                  (b) No irregularity (CASE 3-1)

Figure 13: Effect of differences in train speed

(a) Center slab                    (b) Cantilever slab

(1) Effect on Excitation Force

Figure 12 (a) shows the effect of the train speed on excitation force in the basic case. Here, a linear scale is used for the vertical axis in order to clearly indicate the peak. Among the spring elements that are continuously distributed in the longitudinal track direction, one excitation force at the center of the model was taken up as the target excitation force. From the figure, it can be seen that peaks such as those generated by a 2.5 m wheelbase shift 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 of in the vicinity of 84 Hz at a speed of about 190 km/h is generated as a result of the approximate matching of the excitation frequency (84.4 Hz = (190/3.6)/0.625) of the spring mass system (consisting of the unsprung mass, rail mass and rail pad) with the excitation frequency due to the reaction received by the unsprung mass at each rail fastening interval of 0.625 m, and that the effect of the excitation force is small.

This natural frequency was obtained by eigenvalue analysis using the vehicle/track model indicated in Figure 2 (a). 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, Figure 12 (b)), it is considered that these peaks are caused by track irregularity.
(2) Effect on Response of Members

Figure 13 shows the effect of differences in train speed on the response of the center slab and the cantilever slab.

Regarding the center slab, it can be seen that 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 have 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 (Figure 8 (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 that had been generated by an excitation force. This peak is generated as a result of approximate matching of the excitation frequency of the spring mass system (consisting of the unsprung mass, rail mass and rail pad) the excitation frequency due to reaction received by the unsprung mass at each rail fastening interval of 0.625 m.

Regarding the cantilever slab, it can be seen that 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 an individual cantilever slab or a cantilever slab that is coupled to a center slab (Figure 8 (c), (d), (g), and (l)). In contrast to the case of a center slab, excitation frequencies may not be necessarily in proximity to those determined by the train speed and the 2.5 m 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) some time. 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 the excitation frequency of 3.7 Hz determined by the train speed and the carriage length of 25 m; therefore, and that there is a possibility that the peak is excited by this excitation frequency. The vibration mode of a cantilever slab is a complicated natural vibration mode in which a cantilever slab is coupled to a center slab and a handrail. We will carry out an evaluation for the mode with due consideration on the effect of other parameters as well in future.

Although it had been limited to a certain train speed in the conventional experiment and analysis, by using the new numerical model, namely “discrete model”, we newly have been able to construct three-dimensional graphics consisting of train speed, frequency and acceleration power spectrum.

4.2. Effect of Various Parameters

Figure 14 shows the influences of the difference of various parameters on the acceleration response at the center slab. Because the response of the center slab is bigger, we carried out the parameter studies for center slab.

In CASE2-1, the response was obtained by representing vehicle not as a dynamic model with 31 degrees of freedom shown in Figure 4, but rather as a series of load of constant force equivalent to the static wheel load (the load interval was the same as that of the vehicle). 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. Conversely, in the high frequency region above 20 Hz, the response according to a series of the load is smaller, thus indicating the effect of the vibration system of the carriage. Generally, the natural frequency of the body is of the order of several Hz, suggesting that for
the frequency band concerned, the effect of the unsprung mass is large in the vibration system of the carriage.

In CASE2-2, the mass of each of the body, bogies and wheelset is all reduced equally by about 30% compared to the basic case. The spring constant is left unchanged. The figure shows the effect of the reduction of mass of the carriage 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, it can be seen that the peak determined by the carriage length has changed from 3 Hz of the basic case to 3.2 Hz (= (270/3.6)/23.75), because the carriage length changes from 25 m to 23.75 m. It can also be seen that this effect causes the response peak to shift to a frequency that is an integral multiple of 3.2 Hz.

In CASE3-1, the track irregularity shown in Figure 6 is not set on the traveling surface of the vehicle. Instead, a setting is made for the case where the vehicle travels on a smooth surface. From the figure, it can be seen that in the frequency band of about 20 Hz or less there is no difference in the response according to whether or not there is track irregularity. It can also be seen that in the frequency band higher than 20 Hz, the response in the case where the rail irregularity is not taken into account is significantly lower compared to that in the basic case. On the other hand, it can be seen in the frequency band between about 100 and 140 Hz, the response is more or less the same regardless of whether or not there is rail irregularity. It is thought that this is due to the fact that the excitation frequency caused by the reaction received by the unsprung mass at each rail fastening interval of 0.626 m when the train is traveling at a speed of 270 km/h is 120 Hz (= (270/3.6)/0.625).

In CASE3-2, if the spring constant of the rail pad is set to one half of 60 MN/m which is normally used for high-speed trains, the response in the region above about 70 Hz is even
lower than that of the basic case. The method of supporting the rails with a soft rail pad in order to reduce the track support spring constant is widely used as a countermeasure against ground vibration, and the trend of the frequency bands in which the vibration reduction effect was obtained at an center slab by the use of a low spring constant roughly agrees with the trend of past ground vibration measurement results [9].

In CASE3-3, it can be seen that if the rail is supported continuously, the response in the vicinity of 100 to 140 Hz is lower than that of the basic case. As mentioned previously, this is because the excitation frequency caused by the reaction received by the unsprung mass at each rail fastening interval of 0.625 m when the train is traveling at a speed of 270 km/h is 120 Hz (= (270/3.6)/0.625), and when the rail is supported continuously, the unsprung mass is not subjected repeatedly to an excitation force, at each rail fastening interval of 0.625 m.

In CASE4-1, it can be seen from the figure that by increasing the stiffness by a factor of 5, the response decreases compared to the basic case in the low frequency zone less than around 60 Hz. It can be seen that the natural frequency changes as a result of the increased stiffness of the center slab. For example, in the vicinity of 80 Hz or 180 Hz, an increase of stiffness induces an increased response, and frequencies that produce the reverse effect also exist.

In CASE4-2, the apparent increase in stiffness is taken into account by increasing the Young’s modulus to three times. It can be seen from the figure that the response is reduced compared to the basic case in the frequency zone below about 60 Hz. It can also be seen that the case of the stiffness increase and the trend of the abovementioned center slab resemble each other.

In CASE4-3, the damping constant of the track/structure system model changes uniformly to 5%. In the case of the vehicle/track model used to calculate the excitation force, the damping constant is 2%. It can be seen from the figure that even if the damping constant of the structure is changed to 5%, the effect on the response of the center slab is not very large.

By using the new numerical model, namely “discrete model”, it has become possible to conduct parametric analyses effectively. Consequently, it have been able to elucidate the dominant factor for each frequency band, as shown in Figure 15 which will be described later.

5. Conclusion

In this research, by using the new numerical model, namely “discrete model”, we conducted phenomenon elucidations of structure borne sound which no systematic studies have been carried out to date. 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 method of efficiently analyzing vibration of members up to about 200 Hz, which is the main cause of Structure born sound, was newly developed. As a result of comparing the result of analysis using this method with the result of measurement, it was confirmed that the result obtained using this method is more or less within the range of random variations of measurement.

2. Conventional prediction methods of the railway noise were based on the measurement data. However, by using newly developed method in this research, it is possible to predict the vibration and structure borne sound of the new and advanced vehicle, track and struc-
The train speed was increased from 160 km/h to 370 km/h, and a study was carried out on the effect of the parameters for vehicle and structures on the vibration of the structure members. The results are summarized as follows.

(3) There are cases in which large peaks occur in the response of the cantilever slab or center slab, at specific speeds and frequency bands, with the change 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 or otherwise of rail irregularity, and the fact that the natural frequency of the spring mass system consisting of the unsprung mass, the rail mass and the rail pad approaches the excitation frequency due to the reaction received by the unsprung mass at each rail fastening interval of 0.625 m.

The authors focused their attention on the response of the center slab at a train speed of 270 km/h, and set out their findings below. Figure 15 indicates the dominant factors for each frequency concerning the acceleration response of the center slab at a train speed of 270 km/h.

(4) 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 high in the frequency region below 100 Hz and the frequency region above 180 Hz, and that the axle arrangement affects the frequency at which peaks occurred.

(5) Regarding the parameters of the track, it was found that the effect of rail irregularity is large in the frequency region between 20 Hz and 100 Hz and also in the frequency region above 150 Hz, the effect of the stiffness of the rail pad is large in the frequency region 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 large in the region between 100 and 140 Hz.

(6) Regarding the parameters of the structure, it was found that when the case where the stiffness of the center slab is 60 Hz, and the stiffness of the non-structure members and also the Young’s modulus of the concrete are greater than the design values is examined, there...
is a simple tendency for the reaction to decrease due to the increased stiffness in the frequency region below 30 Hz, there is an increase or decrease in the response at higher frequencies than this, and also the effect of the damping coefficient of the structure is not very great as far as the change of damping constant is in the range from 2% to 5%.

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