Construction of Method for Analytical Evaluation of Turnout Structures

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In Japan, turnout structures are usually designed on the basis of verified experience or JIS specifications. Consequently, there is no established method yet for evaluating original turnout structures. This study examines an evaluation method for turnout structures based on dynamic analyses. When considering new turnout structures, it is necessary to carry out evaluations of running safety and track material strength. In order to implement them efficiently, it was decided to evaluate them separately.

Keywords: turnout, analysis, evaluation, method

1. Introduction

In Japan, “the design standards for railway structures (Track)” were enacted in 2012. Track structure design in Japan is based on these standards which govern the design of structures, through any of the following methods: performance-based design based on limit state design, design based on ‘deemed-to-satisfy regulation’ (deemed-to-satisfy regulation: specification of structures or members whose required performance has been confirmed using the method indicated in the standard) that selects a structure or a member that conforms to the standard. Performance-based design includes verification by experiments and field tests, and verification by experience.

The same methods apply to the design of turnouts. However, in this standard, no specific guidance or design examples have yet been given for the design of a turnout structure that is different from existing ones, so that no formal method has yet been established to evaluate this type of turnout.

In this context, a method was examined for evaluating turnouts based on dynamic analysis.

2. Method for evaluating turnout structures

In order to examine a turnout structure, its train running safety and track material strength have to be evaluated. The evaluation of running safety requires the calculation of wheel loads and lateral forces, while the evaluation of track material strength needs the calculation of stress generated in the track material as trains pass through it. These evaluations can be simultaneously evaluated by modeling turnouts with FEM solid elements and performing vehicle dynamic analysis. However, this approach however present the drawback of generating large computational analysis loads and requires time consuming reconstruction of the model each time the structure is altered for reconsideration.

Therefore, in this study, to make running safety and track material strength evaluations more efficient, they are considered separately as shown in Fig. 1. A simplified track model simulating the geometrical alignment of turnouts using beam elements was used to evaluate running safety. To evaluate material strength, a detailed model constructed by modelling the turnout using solid elements was used, to which the static load of one wheelset, obtained as the output of the simplified track model, was applied.

3. Evaluation of train running safety

Compared to ordinary track structures, turnouts are complicated, and involve the transition of the wheel on the rail and the longitudinal variation of the switch rail section. Modelling a turnout is therefore an onerous task. In addition, if the geometrical alignment of the considered turnout does not meet requirements, it has to be remodel-
eled, creating inefficiency.

Study describes a method developed to model only the geometrical alignment of turnouts as beam elements without consideration of the transition of the wheel on the rail and longitudinal variation of the switch rail section, so that modelling work and calculation loads can be reduced. This method can evaluate running safety of various turnout structures flexibly, with a reduced calculation workload.

A general-purpose simulation software “Virtual Performance Solution” was used for the analysis, that uses a dynamic explicit method that can be speeded up easily by parallelization.

3.1 Analysis model

3.1.1 Simplified track model

As shown in Fig. 2, rails and sleepers are modeled by FEM beam elements. Spring elements modeling fastenings are placed between the rail and the sleeper. Under the sleeper, spring elements simulating the ballasted track are arranged. Each spring constant is shown in Table 1.

The surface of the rail is modeled using shell elements for contact with the wheel. Rail beam elements and rail shell elements are rigidly connected, and the contact force with the wheel acts on the beam elements through the shell elements.

In addition, since the evaluation of the train running safety of turnouts basically depends on trains running on branch line tracks, this method only evaluated running on branch lines and not main lines.

### Table 1 Each spring constant of track model

| Fastening        | Vertical direction | 2000 kN/mm / One fastening (Equivalent) |
|------------------|--------------------|----------------------------------------|
|                  | Horizontal direction | 16 kN/mm / One fastening                |
| Ballasted track  | Vertical direction | 75.2 kN/mm / One sleeper               |
|                  | Longitudinal direction | 9.0 kN/mm / One sleeper               |
|                  | Horizontal direction | 2.6 kN/mm / One sleeper               |

3.1.2 Vehicle model

As shown in Fig. 3, the vehicle is modeled using a MBD (multibody dynamics) model. The car body, the bogie frame, and the wheelset are modeled as rigid bodies. Axial springs, air springs and traction rods are modeled using spring elements and damper elements. The wheel surface in contact with the rail was modeled as a solid element. For contact, a penalty method is used in which springs are applied vertically between the contacting interfaces.

Vehicle specifications are shown in Table 2.

### Table 2 Vehicle specifications

| Item                                | Specifications               |
|-------------------------------------|-----------------------------|
| Track gauge                         | 1067 mm                     |
| Distance between two bogies          | 14.4 m                      |
| Wheelbase                           | 2.1 m                       |
| Suspended load                      | 21.6 ton                    |
| Mass between primary spring and secondary spring | 1.5 ton / One bogie |
| No suspended load                   | 1.5 ton / One axel          |

3.2 Verification of validity

In order to verify the validity of the evaluation method, a simplified track model of a specific turnout was created and analyzed, and a comparison was made with past running tests [1].

3.2.1 Target of the analysis and analysis case

Figure 4 shows an overall view of the constructed analysis model. The target turnout was a simple turnout, and the lead curve radius of the outer rail of the turnout was about 186 m. The running direction of the vehicle was on the facing branch line side, and the running speed was 20 km / h.

Comparison with the running test results was carried out in two cases shown in Table 3. Case 1 is the case where the track irregularity at the time of test is set to the track model in consideration of the geometrical alignment of the turnout. Case 2 is a case where the wheel / rail frictional coefficient of the analysis model of Case 1 is changed in accordance with the time of test. The wheel / rail frictional coefficient in Case 1 was 0.3.

From the purpose of evaluation of the running safety, the verification compares the wheel load and lateral force of the first axis of the analysis result and the test result. In the wheel load and lateral force measurement method used in the running test, 100 Hz low-pass processing was performed on the wheel load and lateral force data, so 100 Hz low-pass processing was also done in this analysis result.
3.2.2 Analysis conditions and results of case 1 [2]

The track irregularity at the time of the running test is shown in Fig. 5. These track irregularities are obtained from restored waveforms for wavelength components of 6 m to 25 m. The measurement results of the alignment show that, in the vicinity of the toe of the switch, displacement occurs such that both the inner rail and the outer rail bulge toward the outer rail.

Figure 6 shows that the analysis results when the geometrical alignment and the track irregularity of the turnout are set in the track model. The timing at which the wheel load and lateral force occur roughly agrees with the test results. However, the wheel load and the lateral force of the outer rail were slightly smaller.

In Fig. 6, the analysis result of the lateral force on the outer rail side is smaller than that of the test result. It may be due to the influence of the wheel / rail frictional coefficient. With regard to the wheel / rail frictional coefficient, it is known that the derailment coefficient (Lateral force / Wheel load) and the frictional coefficient are almost the same value only on the inner rail side of the sharp curve. This running test was conducted in a sharp curve of radius of about 186 m. Therefore, the derailment coefficient (Lateral force / Wheel load) of the inner rail is shown in Fig. 7 as the wheel / rail frictional coefficient of the inner rail side. After entering the sharp curve, at 2 m from the toe of the switch lateral force becomes suddenly increased, followed by some fluctuation but remaining almost constant. The average value in the range of 2 m to 12 m from the toe of the switch was 0.34.

From the test results, an analysis was performed in the case where the wheel / rail frictional coefficient of the inner rail side was 0.34. The wheel / rail frictional coefficient of the outer rail side was kept at 0.3. The analysis results are shown in Fig. 8. The figure also shows the result of Case 1 for comparison. The wheel loads were omitted because there was not much difference with Case 1. The lateral force on the outer rail side slightly increased, and the test results approached slightly.

Table 4 shows a comparison of the maximum values of the lateral force of the analysis result and the test result at a point where the lateral force rapidly increases after entering the curve (around 2 m from the toe of the switch). The difference in the maximum lateral force of the outer rail was 6%, and the inner rail was 29%. Although the inner rail side was somewhat different, the outer rail side where the lateral force was large was almost the same. In addition, Table 5 shows a comparison of the average values of the wheel load and the lateral force at the time of running in a curve (in the range of 2 m to 12 m from the toe of the switch). The difference between the average wheel load of the analysis and the test results was within 15%, and the results agree in general. The difference in average lateral force was 26% in the outer rail and 4% in the inner rail, and although the difference in the outer rail was somewhat large, the analysis and the test results were consistent in general.

The difference between the analysis results and the
test results is that the simplified track model does not consider the difference between the design cross section and the worn cross section of the rails and wheels, and since the target track was a ballast track, static track irregularity was reflected in the analytical model, but dynamic track irregularity may not have been reproduced.

![Fig. 7 Inner rail lateral force / wheel load test results](image)

**Fig. 7 Inner rail lateral force / wheel load test results**

![Fig. 8 Analysis results from case 2](image)

**Fig. 8 Analysis results from case 2**

| Table 4 Difference of maximum lateral forces between analysis and test result of Case 2 |
|----------------------------------|-------------------|-------------------|
|                                  | Test (kN) | Analysis (kN) | Difference (%) |
| Maximum lateral force            | Outer rail | 24           | 25              | 6                |
|                                  | Inner rail | 15            | 10              | -29              |

| Table 5 Difference of average load between analysis and test result of Case 2 |
|----------------------------------|-------------------|-------------------|
|                                  | Test (kN) | Analysis (kN) | Difference (%) |
| Average wheel load               | Outer rail | 53             | 45              | -15              |
|                                  | Inner rail | 32             | 30              | -8               |
| Average lateral force            | Outer rail | 24             | 18              | -26              |
|                                  | Inner rail | 11             | 10              | -4               |

3.2.4 Consideration

It was confirmed that the analysis results of the simplified track model and the running test results agrees generally. From this, it is considered that running safety can be roughly evaluated using a simplified track model in which the geometrical alignment of the turnout is simulated by beam elements.

4. Evaluation of the track material strength [3]

In order to correctly evaluate the track material strength, it is important to understand the load applied to the track material when the car is running. The proposed evaluation method of the track material strength applies the wheel loads and lateral forces, which are the analysis results of the simplified track model described in Chapter 3, to the track member as external force. Specifically, the turnout structure to be evaluated is modeled by solid elements. Then, the wheel load and the lateral force calculated by the simplified track model are statically applied to the solid model as an external force via one wheelset. Then, the response of the track material is evaluated.

4.1 Analysis model

This chapter describes the detailed model built for a movable set-off mechanism (hereinafter referred to as “set-off mechanism”). Figure 9 shows a schematic figure of the detailed model.

In the detailed model, track members to be evaluated are modeled by solid elements. In the model shown in the figure, main rail, support rail, set-off rail were modeled as solid elements. For loading, place one wheelset at an arbitrary position, and then statically apply the wheel load and the lateral force calculated by the simplified track model to the left and right wheels of the one wheelset respectively.
4.2 Verification by comparison

The detailed model was modeled as a solid element including not only the track member but also the geometrical alignment of the turnout. Therefore, it is also possible to arrange a vehicle model instead of one wheelset and carry out a vehicle running analysis.

Checks were therefore made to verify whether the response of the track member due to the load acting on the vehicle traveling and the response of the track member due to the static load of one wheelset, were equal. In addition, it is possible to simultaneously evaluate not only the member strength but also the running safety when the vehicle running analysis is performed using the detailed model. This model is hereinafter referred to as the “simultaneous evaluation model”.

4.2.1 Target of the analysis and analysis case 1

The running direction of the vehicle was facing, and the running speed was 5 km / h. The strain generated in the rail and set-off rail at the positions shown in Fig. 10 was compared with the simplified trajectory model and the simultaneous evaluation model.

Figure 11 shows a comparison of the wheel load and lateral force of the simplified track model and the simultaneous evaluation model when passing through the set-off mechanism. From the figure, it can be confirmed that the results on running safety coincide with the simplified track model and the simultaneous evaluation model. In the subsequent verification of the detailed model, the results of the simplified track model analysis shown in the figure were used as the load to be input to the detailed model.
4.2.2 Analysis result

A comparison of the analysis results of the detailed model and the simultaneous evaluation model with respect to the strain at each evaluation point in Fig. 10, is shown in Fig. 12. It was confirmed that the detailed model and the simultaneous evaluation model almost coincided despite some differences. An explanation for the difference in results could be that since the set-off mechanism to be analyzed has a complicated assembly structure, it is conceivable that the contact state of the wheel and the track member and the contact state of the track members, etc. may have been different whilst the vehicle was running and during static loading.

In addition, it was confirmed that the time required to conduct the analysis to evaluate running safety and member strength using the proposed method could be reduced by about 70%, compared to the simultaneous evaluation model, with the same computer and number of parallel connections.

![Fig. 12 Comparison of analysis results](image)

4.2.3 Consideration

As a result of examining the method to evaluate the member strength using the detailed model, it was confirmed that the analysis results almost agreed with the simultaneous evaluation model and the detailed model.

These results confirmed that it was possible to evaluate the response of a member simulating a running vehicle using the detailed model statically loaded on one wheelset.

5. Conclusions

This study proposed a method for evaluating turnout structures based on dynamic analysis, which was then verified to confirm its validity. The results are summarized as follows:

1. In order to consider a new turnout structure, it is necessary to evaluate running safety and track material strength. In order to carry out these evaluations more efficiently, it was decided as part of this research, to evaluate them separately.
2. Running safety was evaluated using a simplified track model in which only the geometrical alignment of turnouts was simulated with beam elements without considering the transition of the wheel on the rail and the longitudinal variation of switch rail section. In order to verify the validity of this model, running through a simple turnout on the branch line side was examined, and a comparison of these results with a running test confirmed that they almost agreed.
3. The track material strength was evaluated by using a detailed model in which the wheel load and lateral force obtained from a simplified track model were statically loaded via the wheelset on the turnout member modeled by solid elements. In order to verify the validity of this method, a comparison was made of results obtained from a case where the vehicle model ran through a movable set-off mechanism, and results of analyses using the static loading method, confirming that they both roughly agreed. Future work will focus on the design of a new turnout structure using this method with further research on gaining deeper insight into this method.

References

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