Technical Report / Technology

Strength Design and Fatigue Damage Evaluation Method of Railway Axles for Rolling Stock

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Abstract

Railway axles are an important component in ensuring train safety. In order to review the current design methods of axles and develop more reasonable design methods, two standards on strength evaluation methods and fatigue damage evaluation methods for railway axles were established on September 1, 2016. The establishment of the strength design method enables the reduction of axle diameter for high-speed train axles, and the establishment of the fatigue damage evaluation method led to the creation of reasonable criteria to prevent an increase in diameter beyond what is necessary to achieve the maximum design speed. This paper reports the features of both standards with case studies.

1. Introduction

The railway rolling stock axle is an important component to ensure the safety of a rolling stock (Refer to **Fig. 1**). As design methods for axle strength, there are two standards: one is the "JIS E 4501:1995 Railway rolling stock−Design methods for strength of axles[1)]" (hereafter referred to as JIS E 4501), and the other is the "JRIS D1201-1:2016 Rolling stock−Axle strength−Part 1: Design method[2)]" (hereafter referred to as JRIS D1201-1) which complements the JIS. Furthermore, intended for the development of more reasonable railway axle design methods based on fatigue damage is the standard for methods of evaluation of axle fatigue damage "JRIS D1201-2:2016 Rolling stock−Axle strength−Part 2: Fatigue damage evaluation method[3)]" (hereafter referred to as JRIS D1201-2). Proposed by Nippon Steel Corporation, the basic study pertaining to JRIS D1201-1 and JRIS D1201-2 by the "Research Committee for High-Speed Rolling Stock Axle" was started in August 2010. Later, as a substructure of the Committee, a subcommittee consisting of experts for establishing a standard was formed. Discussions continued from May 2012 until March 2015, and the respective method came into effect on the 1st of September, 2016 as the Japan Association of Rolling Stock Industries Standard (JRIS). This report introduces the concepts of the present axle strength design and the fatigue damage estimation based on JRIS D1201-1 and JRIS D1201-2.

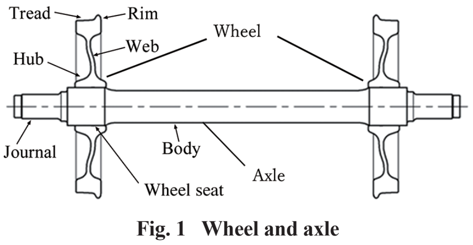

Fig. 1 Wheel and axle

2. Standard for Axle Strength Design Method (JRIS D1201-1)

The axle design standard JIS E 4501 was established in 1972. However, as only the fundamental items were provided, until the establishment of JRIS1201-1, JIS E 4501 that complemented it in detail was used as the design method (hereafter referred to as the conventional design method).

However, the conventional design method was not established as an official standard, and additionally, there was a concern about the calculation overestimating the load caused by braking and driving. Therefore, solutions for such issues were sought, and the conventional design method was improved, and was officially standardized.

2.1 Standard based on the conventional design method

For standardization as JRIS D1201-1, the following items that complemented JIS E 4501 by the conventional design method were

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(1) Although, in JIS E 4501, the subject part of the stress evaluation was limited to the wheel seat of the axle of the pressed-in wheel, the design procedure was provided in detail so as to enable the design of the entire axle including the starting point of a fitted part and the starting point of the change in shape in the stepped area.

(2) Although, in JIS E 4501, braking force is not taken into consideration, and the driving force is evaluated as the ratio of added stress, the load calculation method taking into account the load generated by braking and driving was provided in detail. Thus the force exerted on the axle by braking and driving from the component equipment installation location and the force acting on the journal via the bogie frame as the reactional force of the motor armature shaft are now taken into account.

(3) As opposed to JIS E 4501 which evaluates only the bending stress, the design method was developed to take into account the combination of bending stress σ and torsional stress τ based on the following formula of Nagashima-Nakamura[4)].

$$S_{\mathrm{D,N}} = \frac{1}{\sqrt{\left(\frac{\sigma_{\mathrm{N}}}{\sigma_{\mathrm{s,N}}}\right)^2 + \left(\frac{\tau_{\mathrm{N}}}{\tau_{\mathrm{s,N}}}\right)^2}} \tag{1}$$

Where, $S_{\mathrm{D,N}}$ denotes the design safety factor, the suffix N denotes the division of the evaluated section, and s denotes the allowable stress. Furthermore, the allowable stress is unchanged from those of the conventional design method and JIS E 4501.

2.2 Provision reviewing the conventional design method

The following items which had been considered as issues to be revised in the conventional design method were reviewed.

(1) For calculating the braking force and the driving force, the load acting in the vertical direction used for obtaining the tangential force in the circumferential direction of the wheel was reviewed from the velocity-dependent dynamic load to the static load without speed-dependency.

(2) Regarding the standard safety factor, in response to the review of the evaluation method of the braking force and the driving force, to secure safety equivalent to that of the conventional design method, the safety factor was converted and newly set.

(3) The upper limit of the applicable speed in the high-speed railway system (System 1 Applied division SA[*1]) was increased from the current 350 km/h specified in JIS standard to 370 km/h.

Details of the above three items will be described in Sections 2.4 to 2.6 later.

2.3 Strength design procedure

Herein, the procedure of the strength design method based on the standards described in Section 2.1 and Section 2.2 is introduced.

(1) Setting an axle shape plan and the positions of the sections for strength design. The dimension of the respective part is to be tentatively decided.

(2) Calculation of the acting stress caused by the load exerted on the axle.

(3) Calculations of the bending moment and the bending stress.

(4) Calculations of the torsional moment and the torsional stress (Refer to Section 2.4).

(5) Calculation of the design safety factor. Apply the bending stress and the torsional stress obtained by (3) and (4) above to Formula (1).

(6) Evaluation. Confirm the design safety factor of (5) above exceeds the standard safety factor (Refer to Section 2.5).

2.4 Review of the speed-dependency of the tangential force in the circumferential direction of the wheel

In the conventional design method, the tangential force F_{t} in the wheel circumferential direction used for the calculation of the braking torque and the driving torque employed the following Formulae from (2) to (4). Namely, F_{t} was considered to be dependent on the design maximum speed.

$$F_{\mathrm{t}} = \mu \times 2W_0 \tag{2}$$

Where,

W_0: vertical load acting on a journal consisting of the load acting on an axle and an additional dynamic load

μ: coefficient of friction between a rail and a wheel

$$W_0 = (1 + \alpha_{\mathrm{V}}) \times \frac{W_{\mathrm{A}}}{2} \tag{3}$$

Where,

α_{V}: vertical dynamic force coefficient

W_{A}: static force by the mass on an axle

α_{V} is sought by the following formula for a high-speed rolling stock. V is the maximum design speed (km/h).

$$\alpha_{\mathrm{V}} = 0.0027V \tag{4}$$

In order to evaluate the validity of this formula, the torsional stress generated while actually traveling on-track was investigated. **Figures 2** and **3** show the relationship between the speeds before braking and accelerating (“Initial speed” in the figures) and the variations of the torsional stress, and it is found that the torsional stress is not dependent on speed. Based on this result, in JRIS D1201-1, the following revised formula employing static force and without speed-dependency is employed.

$$F_{\mathrm{t}} = \mu \times W_{\mathrm{A}} \tag{5}$$

2.5 Establishment of new standard safety factor

Due to the review described in the preceding section, the design load is reduced in JRIS D1201-1. However, in order to avoid the deterioration of the actual safety on account of this revision, the standard safety factor was converted and reviewed. Specifically, the standard safety factor in the conventional design method was replaced through the following procedure, and a new standard safety

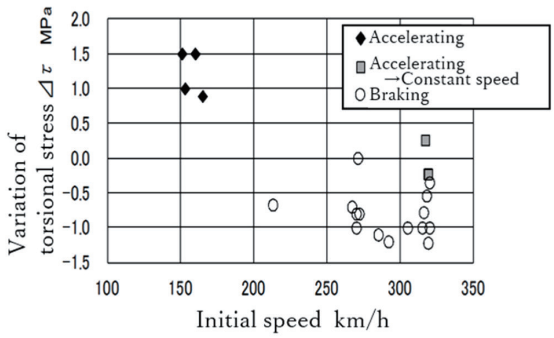


Fig. 2 Initial speed - variation of torsional stress (Motor car)

[*1] The classification set forth herein corresponds to the rolling stocks, tracks, and the relevant maintenance and management, and corresponds to the high-speed railway system having high-speed rolling stocks and tracks (System 1), and the conventional railway system of another (System 2). Its Applied division corresponds to the design speed.

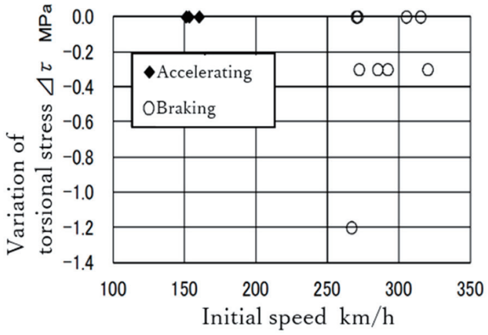


Fig. 3 Initial speed - variation of torsional stress (Trailer car)

factor was provided.

Table 1 shows the standard safety factor defined in the conventional design method, and the axle diameter satisfying the standard safety factor based on the concept of the conventional design method was determined. Then, the safety factor of the axle diameter determined based on the concept of JRIS D1201-1 was provided by counting backwards, and was employed as the candidate standard safety factor. Herein, in the conventional design method, the braking force and the driving force increase as the design speed goes higher. However, since in JRIS D1201-1, there is no speed-dependency, the candidate standard safety factor value grows higher in line with the increase of the design maximum speed. In general, since the change in the standard safety factor depending on speed is unnatural, the safety factor of the rolling stock with the lowest design speed in the same applied division for which safety is confirmed based on its proven result was provided as the "standard safety factor in the respective part".

However, fundamentally, the change in safety factor depending on part[*2] cannot be explained logically. Therefore, a design was studied which realizes the result equivalent to that of the design which employed to the extent possible the standard safety factor of individual parts, and as a conclusion, the unified standard safety factors as shown in **Table 2** were decided.

Furthermore, concerning System 2, in the case of the design using Formula (5) which does not take into account the speed-dependency and the new safety factor, the axle diameter may become larger in the actual design, and the use of the conventional design method is accepted.

As for the Shinkansen, in the conventional design method, due to historical circumstances, the standard safety factor actually differed depending on users. However, in establishing this standard, the safety factors were unified, and for the Applied divisions of SA and A of System 1, the value at 210 km/h of the 0 Series Shinkansen having a proven performance result was employed as the standard value. With this, the axle diameter could be made smaller than that of the conventional design, the design example of which is explained in Section 2.7.

2.6 Expansion of applied speed range in high-speed railway system (Applied division SA of System 1)

In response to the increase of the rolling stock speed, the design speed range in the high-speed railway system (Applied division SA of System 1) was increased from 350 km/h to 370 km/h.

[*2] The part herein referred to consists of the respective region corresponding to the fitting of various unit parts and the region where there is no fitting of unit parts.

Table 1 Standard safety factor defined in conventional design method

Railway system	Applied division		Design speed V (km/h)	Standard safety factor
System1	SA		$200<V\leq 350$	Consultation with each customer for each part
	A		$150<V\leq 280$	
System2	A	Limited express (inter-city) type	$V\leq 160$	1.6
		Commuter type		1.2

Table 2 Standard safety factor defined in new design method (JRIS D1201-1)

Railway system	Applied division		Design speed V (km/h)	Standard safety factor S_R	
				Fitted part	Non-fitted part
System1	SA		$200<V\leq 370$	1.99	1.97
	A		$150<V\leq 280$	2.00	1.95
System2	SA	Limited express (inter-city) type	$120<V\leq 160$	2.08	2.04
		Commuter type		1.56	1.52
	A		$V\leq 120$	1.35	1.26

The applied speed range in Applied division SA is up to 350 km/h in JIS E 4501. However, since high-speed rolling stocks with a design speed exceeding this speed range were planned, the test data of Shinkansen travelling at a speed exceeding 350 km/h were added, and the relationship between the travelling speed and the maximum stress was determined. The result is shown in **Fig. 4**. In the range above 350 km/h also, the maximum stress stays below the stress-velocity relationship formula (straight line in the figure) of JIS E 4501:1995 in which the dynamic coefficient of additional force was reviewed in 1995. It is found that the gradient of the increase of the stress vs. velocity is almost equal to that in the range below 350 km/h. With this, the dynamic coefficient of additional force in the Applied division SA of System 1 was unchanged, and the application range was expanded up to the maximum speed of 370 km/h in the measured data.

Furthermore, as for the conventional railway line, the Applied division was divided at 120 km/h, and for below 120 km/h, the dynamic coefficient of additional vertical force of 0.4, and the dynamic coefficient of additional lateral force of 0.3 provided in JIS E 4501:1986, and for 120 km/h–160 km/h, values in Applied division A of System 2 provided in JIS E 4501:1995 are respectively employed, and Applied division B of System 2 was abolished. These provisions are based on those of the conventional design method. Namely, for below 120 km/h of the conventional railway, speed-dependency is not taken into account even in the conventional design method.

Table 3 is a list of the dynamic coefficients of additional force provided by JRIS D1201-1 based on the above study. The formula of the dynamic coefficient of additional force of Applied division SA of System 1 is divided into the dynamic coefficient of additional vertical force α_V and the dynamic coefficient of additional lateral force α_L based on the formula of the bending stress-velocity rela-

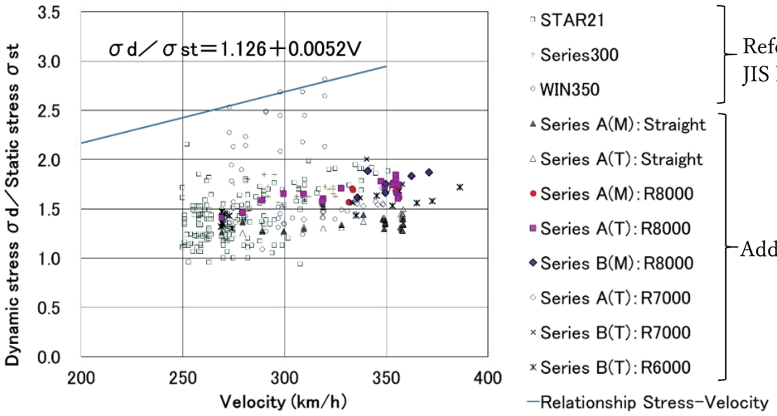
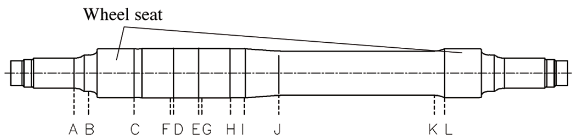


Fig. 4 Velocity dependence of bending stress

Table 3 Dynamic coefficient of vertical and horizontal additional force[2)]

Railway System	Applied division	Design speed V (km/h)	Dynamic coefficient of vertical additional force α_V Dynamic coefficient of horizontal additional force α_L
System1	SA	$200 < V \leq 370$	$\alpha_V = 0.0027V$ $\alpha_L = 0.030 + 0.00060V$
	A	$150 < V \leq 280$	$\alpha_V = 0.0027V$ $\alpha_L = 0.030 + 0.00085V$
System2	SA	$120 < V \leq 160$	$\alpha_V = 0.0027V$ $\alpha_L = 0.040 + 0.0012V$
	A	$V \leq 120$	$\alpha_V = 0.4$ $\alpha_L = 0.3$

Variation of diameter (mm) *												Variation of mass (kg) *
Evaluated section												
A	B	C	D	E	F	G	H	I	J	K	L	
0	-2	-6							-8	-6		-29



*: Designed value by JRIS D1201-1 minus that by conventional design method

Fig. 5 Estimated variation of diameter and mass estimation by JRIS D1201-1 compared to conventional design method

tionship in Fig. 4.

2.7 Change in axle diameter by application of new standard

In the design of Shinkansen axles based on JRIS D1201-1, the minimum required axle diameter to satisfy the standard safety factor for a rolling stock with a design maximum speed exceeding 210 km/h is smaller than that of the previous design due to the review of the speed-dependency of the braking force and the driving force and the review of the standard safety factor based on 210 km/h.

Then, axle diameters were compared between that of the conventional design method and that of the JRIS D1201-1 method, assuming a maximum speed of 360 km/h. The resulting changes in diameter and mass are shown in **Fig. 5**. With this, the diameter of the wheel seat is made smaller by 6 mm, and the weight is reduced by 29 kg.

Furthermore, as the minimum required axle diameter has become smaller, it may become possible for the existing axles to be used as they are for the increase of the maximum speed and the increase of the axle load of the existing rolling stock.

3. Standard for Axle Fatigue Damage Evaluation Method (JRIS D1201-2)

The conventional design method and JRIS D1201-1 described in the preceding chapter are based on the concept of fatigue strength. In recent years, since no axle breakage accidents due to fatigue have ever happened, there is no problem of safety in these design methods. However, upon developing more reasonable design methods, it is considered effective to focus on the accumulation of fatigue (fatigue damage). However, a method to evaluate the damage of axles due to the accumulation of fatigue caused by the stress in actual on-track operation has not yet been standardized not only in Japan, but also throughout the world.

Therefore, we tackled the development of a new standard for various evaluation results for easy comparison and examined it by eliminating the difference in results due to the difference in evaluation methods by standardizing the fatigue evaluation method of railway axles.

3.1 Concept of the standard and subject axle material

This standard provides the evaluation method for the fitted part and the non-fitted part of SFA640 and S38C-QA by employing the linear cumulative damage rule based on the Modified Miner's rule. In addition, the evaluation based on the concept of this standard is applicable to other steel grades.

Figure 6 shows the evaluation flow of fatigue damage, which is explained sequentially hereunder.

3.2 Evaluation index

As the index for the evaluation of the fatigue damage design method, equivalent stress is provided. This is because in recent years, there have been no domestic axle breakage accidents at all, and furthermore, the data which evaluate the fatigue damage of the actual size of axle fatigue damage are very scarce; accordingly, we are unable to discuss the fatigue damage focused on axle breakage life. Therefore, by grasping the equivalent stresses of the existing axles that have been used safely, the evaluation of safety was conducted by comparing stresses with these stresses.

As shown in **Fig. 7**, the "equivalent stress" is sought for the stress having a constant amplitude stress which will develop fatigue damage at the same number of cycles at which number the actual on-track stress having varying amplitudes will develop fatigue damage based on the Modified Miner's rule. This is an index which enables fatigue damage evaluation.

In the case of setting a standard value of an equivalent stress for evaluation, select an existing rolling stock having a sufficiently proven safety record, grasp the actual state of operation of the subject rolling stock such as the operating section, operating ratio and average load factor, and calculate the equivalent stress per steel grade and per part, and the highest value among them is to be selected as the standard equivalent stress.

However, in this standard, the standard equivalent stress is not provided specifically. Therefore, hereafter, it is necessary to set proper standard values by gathering the actual values of axles having sufficient proven safety records.

3.3 Gathering of measured stress spectrum (base data)

In order to obtain the equivalent stress, a stress spectrum is necessary, which should be composed of the data of one both-ways consisting of the data of the first axle on the outward train direction and the data of the fourth axle on the return train direction (Refer to **Fig. 8**, both-ways in acceptable reverse direction).

In this standard, two kinds of spectrum are employed for calculating the equivalent stress. One is the spectrum composed of the stress measured during actual rolling stock operation, and the other is the spectrum estimated based on the measured data spectrum. The former is defined as "base data". The estimation method of the latter is described in Sections 3.5 to 3.7.

The base data of the first axle is essential. As for the data of the fourth axle, as a matter of course, measurement is desirable; however, it is not necessarily essential. Refer to Section 3.7 for the estimation method of a spectrum when data of the fourth axle are not available.

3.4 Base data analysis

As a result of the analysis of the spectrum of the data measured on an actual rolling stock, it was found that such spectrum can be approximated mostly by a normal distribution. **Figure 9** shows the spectrum of the actual measured values on a curved section of a track line and a normal distribution curve produced by using the mean value and the standard deviation of the spectrum. Both are almost equal. Therefore, when estimating the spectrum under similar conditions based on the base data, all spectrums were assumed to be normal distributions. Therefore, the base data obtained is assumed as a normal distribution, and the mean value and the standard deviation are sought, and a subsequent study is conducted, employing these values as standard values.

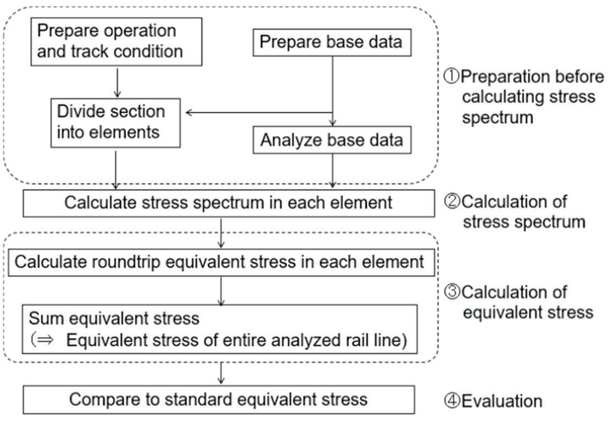


Fig. 6 Evaluation flow of fatigue damage

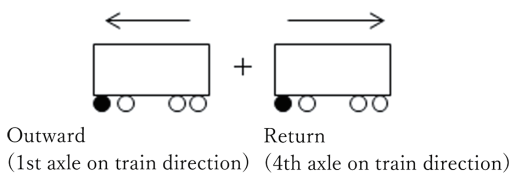


Fig. 8 Axle position of data acquisition

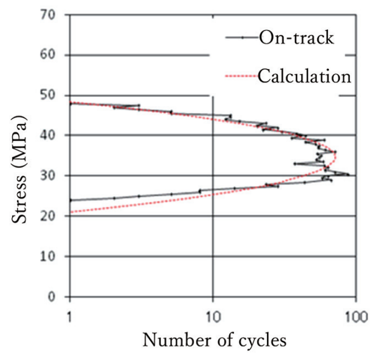


Fig. 9 Spectrum of on-track stress and calculated stress (On curve)

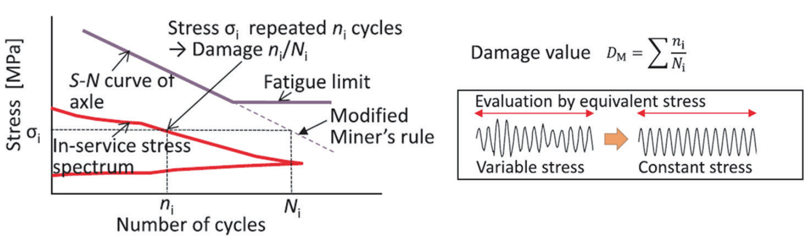


Fig. 7 Modified Miner's rule and equivalent stress[6)]

3.5 Calculation of correction parameter for the section without base data

For the axle of a new type of rolling stock, since the track condition and/or the operation condition are not always the same as those of the existing base data, the equivalent stress cannot be obtained from such base data. In such a case, a new spectrum is developed by correcting the base data obtained under a similar condition. As aforementioned, these spectrums are assumed to be normal distributions.

Then firstly, the bending stress generated on an axle while a rolling stock travels on a track was studied by multibody dynamics simulation. The results are plotted in **Fig. 10**, and the mean value of the distribution in each curved section can be approximated mostly by the curve in the figure. This curve is created by Formula (6), and the mean value of the axle bending stress σ_{av} can be obtained by inputting the excessive centrifugal acceleration α_y, curved section radius R, and the static axle bending stress σ_{st}. The correlation between the mean values thus obtained by calculation (approximation) and the actual measured values is shown in **Fig. 11**, and both are in good agreement.

In addition, when the bending stress spectrum is assumed as a normal distribution on each curve, the standard deviation σ_{dv} is obtained from Formula (7).

$$\frac{\sigma_{av}}{\sigma_{st}} = \mathrm{A} \times \alpha_y + \mathrm{B} \times \mathrm{C}^{e^{-\mathrm{D}\times\log\left(\frac{\mathrm{E}}{R}\right)}} + 1 \qquad (6)$$

$$\frac{\sigma_{dv}}{\sigma_{st}} = \mathrm{a} \times \left(\frac{1}{R}\right)^{\mathrm{b}} + \mathrm{c} \qquad (7)$$

Where, A, B, C, D, E, a, b, and c are undetermined constants, and this standard introduces the Newton-Raphson's method as an example of seeking these undetermined constants[7, 8]. By using Formulae (6) and (7) with the input of these constants, a stress spectrum based on a normal distribution can be obtained by calculation even when the static bending stress, excessive centrifugal acceleration, and the curved section track radius in the subject evaluation section are different from those of the base data.

3.6 Division of evaluation section

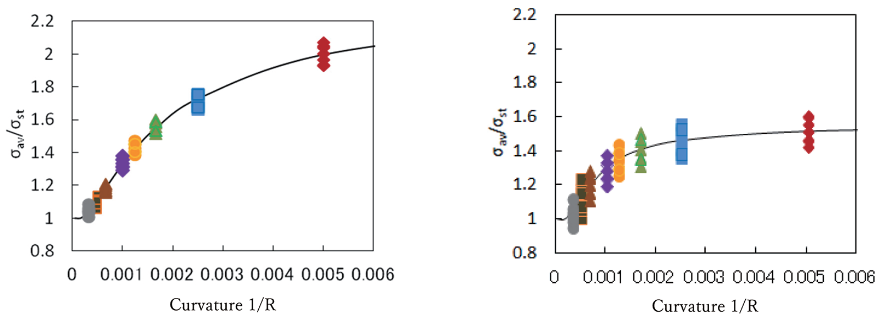
The track line for evaluation is divided into several sections at the boundary positions where the track line information, operation parameters, existence of base data, operating conditions, and so forth change, each section of which is to be addressed as an independent evaluation section.

3.7 Preparation of stress spectrum in respective evaluation section

A stress spectrum is to be prepared per section divided for evaluation. The stress spectrum to be prepared herein is based on that of a both-ways per section (outward direction: first axle, return direction: fourth axle). Furthermore, in a rolling stock workshop, high stress tends to be generated. However, as a result of a comparison of the data obtained only during travelling on a main track line and that added with the data in a workshop, since no significant difference was noticed between the two, data in a workshop was omitted.

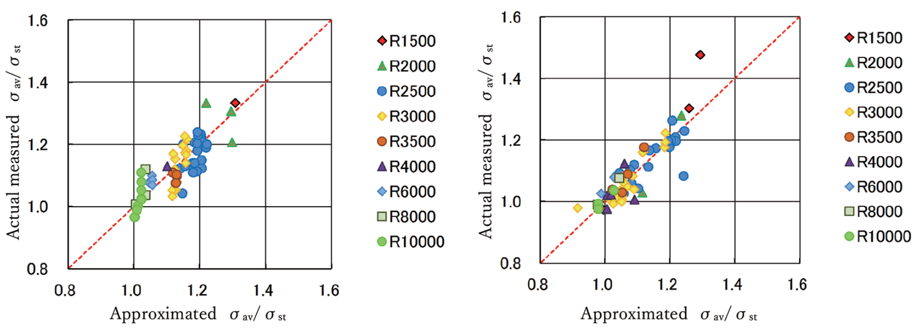
When the condition in a section for evaluation is the same as that of the actually measured base data, the base data itself is assumed as the spectrum to be evaluated.

When the condition in a section for evaluation differs from that of the base data, the normal distribution obtained by inputting into Formulae (6) and (7) the constants introduced by using the base data, the excessive centrifugal acceleration, the radius of the curved track in the subject evaluation section, and the static axle bending stress, and then by calculating the mean value and the standard deviation of the bending stress thereby is assumed as the spectrum in



(a) Outer rail side (b) Inner rail side

Fig. 10 Simulated relationship between curvature and bending stress



(a) Outer rail side (b) Inner rail side

Fig. 11 Correlation of bending stress between approximated and actual measured

the subject section.

Furthermore, regarding the fourth axle, even when the base data does not exist at all, with respect to the static bending stress σ_{st}, a normal distribution having the mean value equal to $1.0\sigma_{st}$ and the standard deviation equal to $0.1\sigma_{st}$ is assumed as the spectrum. This is because, as a result of an investigation, noticeable influence of the travelling speed on the stress of the fourth axle was not clearly recognized, and the change in the stress while passing a curved track was also small.

Furthermore, the bending stress needs to be calculated per axle evaluation position, and nominal stress is taken at the fitted part, and net stress incorporating the stress concentration coefficient is taken at the non-fitted part.

3.8 Calculation of equivalent stress in each section

In the case of studying the axle strength from the viewpoint of fatigue damage, the S–N curve of the part for evaluation is necessary. An S–N curve is generally expressed by Formula (8). In this standard, the parts for evaluation are broadly classified into the fitted part and the non-fitted part, and the values of m required for the calculation of fatigue damage are provided in the **Table 4**, respectively. These values are based on the result of the evaluation of the fatigue strength of an actual size axle conducted by Makino et al.[6].

Furthermore, in JRIS D1201-2, since the data for the S–N curve index m at the non-fitted part of S38C-QA presently used for high-speed rolling stocks are not sufficient, the same value as that of SFA640 is used. Currently, a study is in progress to gather new data, and to revise the standard.

Next, upon evaluating fatigue damage, the concept of the linear cumulative damage rule (Modified Miner's rule) was employed.

The equivalent stress σ_{eq} is calculated by Formula (9) based on the Modified Miner's rule respectively per spectrum constituent element, namely, the first axle in the outward direction, and the fourth axle in the return direction.

$$N = A^* \times \sigma^{-m} \tag{8}$$

Where,
N: number of repeated cycles
σ: bending stress of axle
A^*: S–N curve coefficient
m: S–N curve index

$$\sigma_{eq} = \sqrt[m]{\frac{\sum_i \sigma_i^m \times n_i}{\sum_i n_i}} \tag{9}$$

Where,
σ_i: representative stress at ith rank in stress spectrum
n_i: frequency (number of repeated cycles) at ith rank in stress spectrum

Table 4 Index of S–N curve[3]

Mark	Part	m
SFA640	Fitted	5
	Non-fitted (Surface roughness: Ra = 3.2 μm)	9
S38C-QA	Fitted	6
	Non-fitted (Surface roughness: Ra = 3.2 μm)	9 [a]

Note [a] Determined as the same value as that of SFA640

3.9 Sum up of equivalent stress

The equivalent stress in the respective evaluation section is weighted based on the travelling mileage, and the equivalent stress in the entire evaluation section is required.

3.10 Evaluation

The equivalent stress in the entire evaluation section so obtained is compared with the standard equivalent stress corresponding to the parts for evaluation. When the value of the standard equivalent stress divided by the equivalent stress of the entire evaluation section is 1 or above, it is judged as safe.

3.11 Utilization of the standard hereafter

3.11.1 Subject of application

This standard can be utilized in the following cases hereafter.

(1) To verify the safety of an axle designed by JRIS D1201-1 described in the preceding chapter. The safety of the high-speed rolling stock axle is to be confirmed especially by applying the fatigue damage evaluation since the diameter of the axle designed by D1201-1 becomes smaller than that designed by the conventional design.

(2) To judge the necessity by the fatigue damage evaluation of increasing the diameter of an existing axle designed by the conventional design method for the application to higher speed which may require increase of the axle diameter from the viewpoint of fatigue strength design.

(3) To evaluate safety by the fatigue damage evaluation towards the changes in the track line condition and/or the changes in the operation condition anticipated in a track line extension.

3.11.2 Introduction of expected utilization case

Assuming the application case of (1) of Section 3.11.1 above, the application procedure is illustrated below (Refer to **Fig. 12**).

(1) An axle designed by the conventional design method has already been provided to an existing rolling stock series (defined as rolling stock A). The actual result of the bending stress of this axle is shown in **Fig. 13** with a full line.

(2) To this, a new axle was designed by JRIS D1201-1 described in Chapter 2 based on the design specification for the rolling stock A. Compared with the existing axle, the diameter became smaller by 2 mm at the fitted part, and by 7 mm at the non-fitted part (body part).

(3) Based on the axle diameter so decided, the spectrum of the existing axle in Fig. 12 was modified, the result of which is shown in the same figure with a broken line. As a result of applying the abovementioned fatigue damage evaluation based on this spectrum, the equivalent stress was $\sigma_{eq\,A'}$=33.24 MPa.

(4) On the other hand, as a result of seeking the equivalent stress of an existing rolling stock having a sufficient proven safe operation record (defined as rolling stock B) with the above mentioned method, σ_{eqB}=35.50 MPa was obtained at the same evaluation position.

(5) Since the value of the equivalent stress of the rolling stock B divided by the equivalent stress of rolling stock A is above 1, the newly designed axle is judged to be safely serviceable.

4. Conclusions

(1) For a high-speed rolling stock, based on the axle strength design standard JRIS D1201-1, increase of the allowable design speed and the allowable axle load of the existing axle has be-

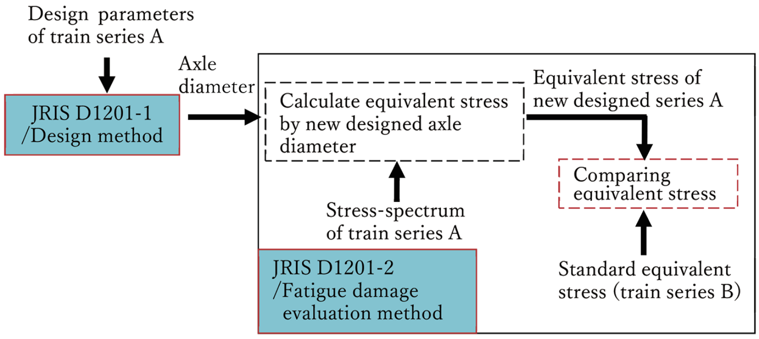


Fig. 12 Evaluation flow of design shape

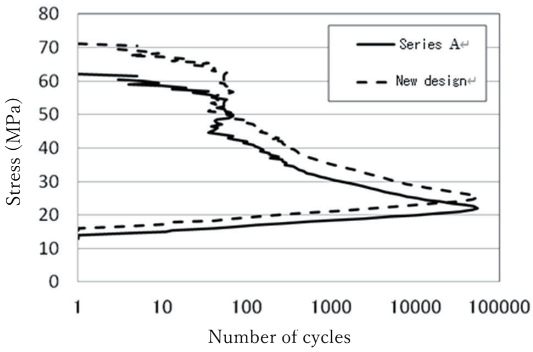


Fig. 13 Modifying stress spectrum

come possible by reducing the design driving force and the design braking force, and by setting the evaluation standard at one of the 0 Series Shinkansen (design maximum speed 210 km/h). As compared with that of the conventional design, the reduction of the axle diameter, namely, reduction of weight is enabled by the new design.

(2) Based on the axle fatigue damage evaluation standard JRIS D1201-2, the evaluation of axle strength from the viewpoint of fatigue damage has become possible, and since reasonable evaluation toward the increase of the maximum speed based on the stress under actual operation has become possible, avoidance of the increase of the axle diameter to more than necessary is considered possible. Concerning the fatigue damage evaluation method, future standardization as further reasonable axle design methods is desired.

References

1) Japanese Standards Association: JIS E 4501 (Railway rolling stock – Design methods for strength of axles), 1995
2) Japan Association of Rolling Stock Industries: JRIS D 1201-1 (Rolling stock – Axle strength – Part 1: Design method), 2016 English translation mentioned in tables
3) Japan Association of Rolling Stock Industries: JRIS D1201-2 (Rolling stock – Axle strength – Part 2: Fatigue damage evaluation method), 2016 English translation mentioned in tables
4) Nagashima, Nakamura: On the Strength of Axles for Electric Rail Car. The Japan Society of Mechanical Engineers Proceedings. 17 (63), (1951)
5) Tezuka, Kanbayashi, Tajima, Yoshie, Nakata: Strength Design Methods of Axles of Railway Rolling Stock. SHARYO GIJUTSU. (207), (1995)
6) Makino, T. et al.: Int. J. Fatigue. 132, 105361 (2020)
7) Shinozaki, Matsushita: Introduction to Applied Numerical Computation Method for Engineering (First volume). 1976
8) Mori: Asakura Modern physics Lecture 7 Numerical Analysis Method. 1984

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