THE REPUBLIC OF INDONESIA

REPORT ON STUDY FOR INCREASE OF TRAIN SPEED ON JAVA, INDONESIA

FEBRUARY 1974

PREPARED FOR

OVERSEAS TECHNICAL COOPERATION AGENCY

BY

JAPAN RAILWAY TECHNICAL SERVICE, JAPAN

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PREFACE

In response to the request made by ECAFE and the Government of Indonesia, the Government of Japan decided to conduct the study for the Train Speed-Up Project of the Indonesian State Railways (PJKA) on Java and entrusted the Overseas Technical Cooperation Agency (OTCA) with this task.

In view of the importance of the project, OTCA deputed its execution to the Japan Railway Technical Service (JARTS).

JARTS, thereupon, formed a study team consisting of 6 members headed by Mr. Seinosuke Arai and dispatched it to Indonesia for 31 days from June 4 to July 4, 1973 to let it make required investigations.

The study team conducted running tests on the three sections of Jakarta-Cikampek, Jogjakarta-Surabaya and Jogjakarta-Madiun with the purpose of increasing the maximum train speed up to 100 km/h and, at the same time, investigated the existing conditions of the track and rolling stock, demonstrating how to evaluate them.

This report contains the results of the study made by the team, and we shall be very happy if it can contribute to the acceleration of the Train Speed-Up Project of PJKA and serve to promote closer relations between Indonesia and Japan.

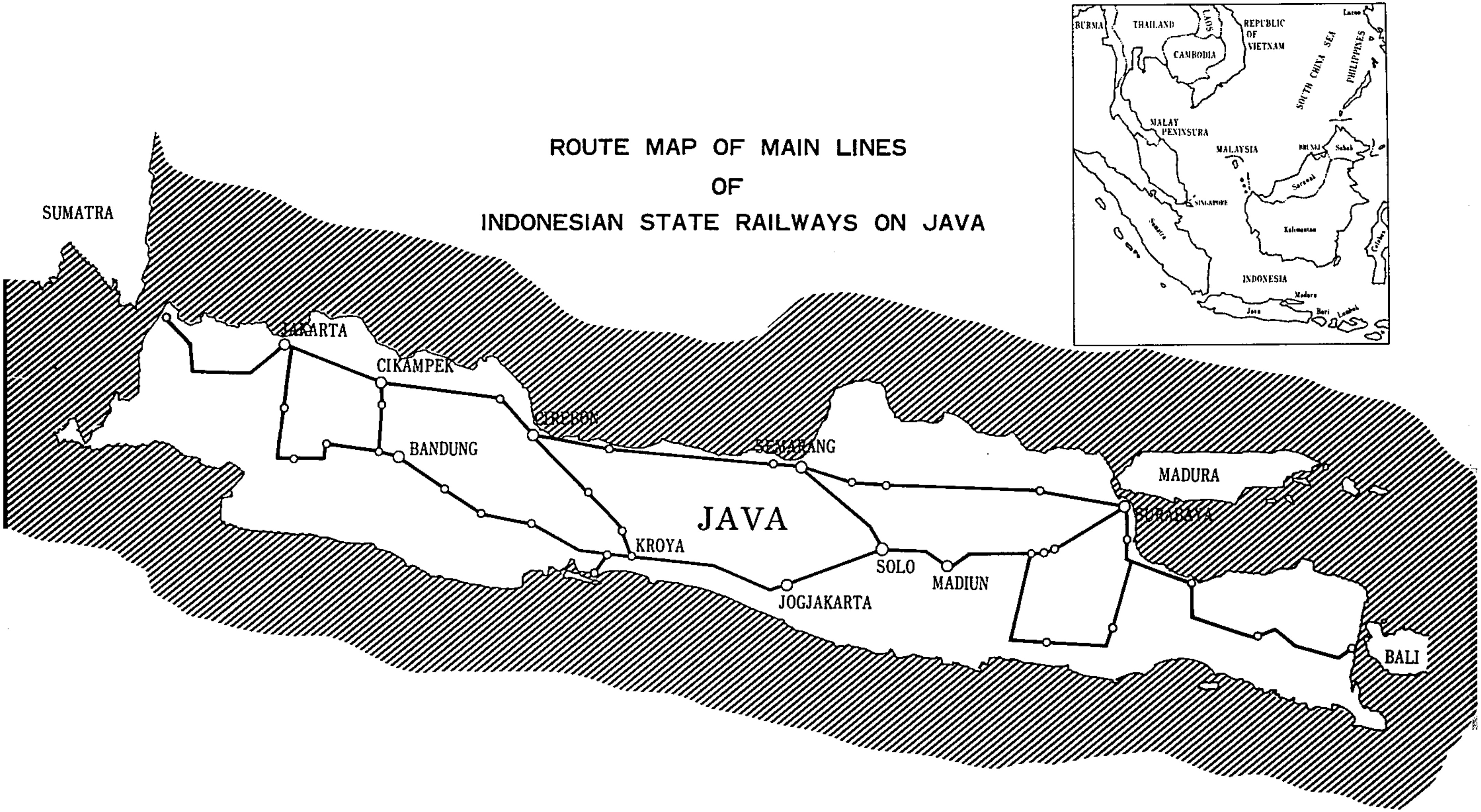
Finally, we would like to express our deep thanks to the ECAFE Secretariat, the Government of Indonesia and PJKA for their warm assistance and cooperation extended to the study team.

February 1974

K. Tatsuke

Keiichi Tatsuke
Director General
Overseas Technical Cooperation Agency

ROUTE MAP OF MAIN LINES OF INDONESIAN STATE RAILWAYS ON JAVA



OUTLINE AND RECOMMENDATION

1. Introduction

Generally speaking, the increase of train speed needs a large investiment for improving or modernizing railway track, signal device, rolling stock and so forth. However, it contributes to the railway management to a great extent, such as increment of traffic revenue by inducing freight and passengers, improvement of working efficiencies of rolling stock and train crew by decrement of train hour [(train running time + train stopping time) x number of train operations], increment of line capacity, etc.

As regards increasing train speeds on existing lines, discussion was made in detail in No.4 Working Group of the symposium on the subject "High Speed Railways and other Forms of Guided High Speed Surface Transport", which was held in Wien, Austria for 6 days from 17th to 22nd of June, 1968 under the joint anspices of IRCA and UIC, and the following is the summarized conclusion reached at that time:

(1) People eagerly hope the elevation of the quality of railway service including speed-up of trains.

(2) There exist many proofs of the fact that traffic quantity increases as train speed goes up.

(3) As regards speed-up of trains, financial problem is more serious than technical problem.

(4) As regards economy of speed-up of trains, it can be said that reveniue increases as train speed goes up.

(5) Attention should also be paid to the social benefit by speed-up of trains.

Increasing the speed of a train means shortening the running time of the train, that is, increasing the so-called "Scheduled Speed". It goes without saying that the alleviation of speed restriction on curved sections, etc. is effectual for this purpose, but it seems better to increase the maximum train speed on the trunk-lines on Java, as they are almost straight excluding a few sections.

It is necessary to investigate the following items in general for train speed-up:

(1) Running safety and riding comfort of rolling stock

(2) Structure and strength of track

(3) Strength of roadbed

(4) Strength of bridge and other structures

(5) Improvement of grade and curvature

(6) Doubling of track

(7) Signal system

(8) Motive power system of rolling stock (power concentrated or dispersed)

(9) Train operation system

This time, however, emphasis was placed on the investigation into the existing conditions of track and rolling stock and the study on the methods and works required for improving these facilities.

2. Purpose of study and its outline

The purpose of the study is to tender technical advice for increasing the maximum train speed of the truck-lines on Java up to 100 km/h and at the same time, to demonstrate how to evaluate the track and rolling stock therefore.

The study was conducted for about one month starting from June 4, 1973, during which were made the train running test and the investigation of the existing conditions of track and rolling stock with the measuring instruments carried with the study team from Japan.

The running test was conducted in three kinds, i.e. short-distance test, long-distance test and high-speed test.

The short-distance test was conducted by operating the test train on the section, Jakarta to Cikampek, several times repeatedly with the train speed changed from 60 km/h to 100 km/h stepwisely. The long-distance test was conducted by operating the test train on the section, Jogjakarta to Surabaya, one time at the train speed of the express passenger train specified on the existing train diagram. The distance of the test run was about 320 km. The high-speed test was conducted by operating the test train on the section, Jogjakarta to Madiun, at speeds 10 to 20 km/h higher than in the case of the long-distance test, based on the result of the said test.

3. Result of study

The track on the straight section generally has poor cross level, showing values far exceeding the allowable limit of PJKA at many places. Settling of track is seen at both ends of every bridge, causing large drooping in longitudinal level. The track in the curved section shows the track gauge and cross level exceeding the allowable limit at many places. Longitudinal level and alignment are almost within the allowable limit.

Conspicuous corrugation of about 40 mm in wave length is seen covering a considerably long extension of rail.

Sleepers used are mostly wooden sleepers (teak wood), though steel sleepers are also used to some extent, and rigid fastening is applied. The track structure seems to belong to a better class of railway standards and have a full strength against lateral pressure of wheels. Ballast is noticed to be insufficient in replenishment for the whole test section. A large quantity of soil mixes into the ballast, and insufficient is the ballast thickness under sleeper. Track settles for almost the whole test sections. There are many rail joint parts showing large depression, or hardened ballast after pumping phenomenon. Large irregularities in alignment and longitudinal level are noticed at the point of and near level crossings. Long stretches of track are noticed being stained by the oil dropping from rolling stock of passing trains.

As for the wear of wheel tread, we have measured that of 3 test coaches. The result shows that the wearing condition of wheel tread differs for respective car, some showing a large wear at the wheel flange part and some showing wear at the tread part. The result of test run shows that there is no tendency of the test rolling stock generating snake motion at up to 100 km/h. It is noticed that the lateral vibration *acceleration of carbody tends to increase monotonously in proportion to the increase* in train speed, whether on the straight section or the curved. It is also noticed that the vertical vibration acceleration of carbody tends to increase monotonously by about 30 to 50% from 60 to 100 km/h, whether on the straight section or the curved, while the said acceleration tends to increase in proportion to the square of train speed when passing a bridge. In the long-distance test conducted between Jogjakarta and Surabaya, it has been found that the riding comfort is the best for the Jogjakarta-Solo section supposedly due to the maximum train speed being restricted at 70 km/h, followed by the Solo-Madiun section and the Madiun-Surabaya section in that order. In the high-speed test, it has been found that the vertical vibration acceleration of each car is 0.4 g or less, making us believe that the maximum value of vertical vibration acceleration on this section may not exceed 0.5 g even under the proposed 100 km/h operation, not to speak of 90 km/h operation.

4. Observation and recommendation

Consolidating the results of the study, it can be said that there are several problematical points in the track and rolling stock requiring remedy or improvement, in the point of view of riding comfort to passengers and safe train operation, before realizing the proposed maximum 100 km/h train operation.

The proposed train operation requires at least the replacement of existing 15P rails with heavier ones (R14A) and the increase of the ballast thickness to 20 cm to let the track have a sufficient load bearing capacity. After train speed is raised, it becomes necessary to strengthen the track and its maintenance force as track irregularities progress faster. And therefore, it is preferable to change the distance between sleepers for the range 62 cm to 65 cm after train speed is increased in order that the track maintenance quantity may be kept as small as possible. It is also preferable to improve the structure of rail joint, as part of strengthening the track. As for track irregularities, it is necessary to better the longitudinal level, making smaller the allowable limit of the level, and special attention must be paid to such places as rail joint, level crossing and bridge. Further, it is necessary to make track measurements as often as possible to grasp track irregularity conditions as correctly as possible, and maintain the track so that track irregularities or sway of rolling stock is within the allowable limit or the goal value.

As for track maintenance work, especially to heighten the efficiency of the ballast work, i.e. overall tamping, surfacing, etc., it is preferable to improve the work tools and method. Also as for the ballast, it is necessary to keep the specified ballast thickness under sleepers at all times by raising the track level by overall tamping or executing the screening of ballast.

It is necessary to correctly grasp the actual status of deterioration of track materials, especially deterioration ratios of rail, rail joint, rail fastening device, sleeper, etc. so as to be able to set up a desirable track maintenance plan when the proposed train speed-up is realized.

It is necessary to stop letting oil drop from passing trains on the track especially on the rail, for better track maintenance condition.

As for deteriorated bridges (only those bridges presenting problems in the strength of bridge girders, abutments and piers) and soft roadbed sections, it is necessary to measure the stress, deflection and vibration for the former and the bearing capacity, settlement and vibration for the latter and to establish new restricted speeds for trains passing on them or improve them before realizing the proposed train speed-up.

As for the rolling stock to be used for the proposed 100 km/h operation, it is especially necessary to take measures to decrease vertical vibration of their carbodies.

For this, it becomes necessary to use vertical-moving dampers and keep their normal characteristics at all times.

Brake performance becomes an important factor to obtain a fixed brake distance. Further, the structure of coupling device of rolling stock influences the riding comfort to a great extent. From these points of view, the cars of the same types as No. 1, No. 2 and No. 3 coaches which were all used as test car, are usable for the proposed high-speed operation.

The lateral vibration acceleration of carbodies is comparatively small. This indicates that PJKA rolling stock are fairly stabilized against lateral vibration and that the track of PJKA is comparatively free from large irregularity in track alignment. No snake motion was noticed even at 100 km/h train operation. As snake motion tends to appear as wheel tread wears, it is necessary to pay special attention to the wear of wheel treads under high-speed train operation. From the results of the long-distance test and the high-speed test, it can be determined that the proposed speed-up of trains can be possible for some sections under the existing conditions, but on the pre-condition that the maintenance condition of the track is kept at the existing level at all times.

5. Improvement for proposed train speed-up

As for the track improvement, Jogjakarta to Surabaya, 322 km, the following works are necessary in general, though the quantities of the works differ on each section:

(1) To replace the existing rail and turnouts with heavier ones

(2) To improve rail joints

(3) To add sleepers

(4) To raise track

(5) Miscellaneous

Further, it is necessary to prepare the machines and tools for these works.

The following is an example of the cost estimate for the above improvement works:

(1) US$5,200,000 (equivalent to Rp 2,200,000,000) for preparation of rail, turnouts, machines, tools and others

(2) Rp 2,000,000,000 for welding of rail joints, addition of sleepers, raise of track and others

As for improvement of rolling stock, the following is an example of the cost estimate in the case where all the existing express trains (15 trains) are sped up on Java:

(1) US$7,500,000 (equivalent to Rp 3,100,000,000) for new construction or remodelling of 2/3 of the cars necessary for 15 trains

(2) Rp 340,000,000 for others

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REPORT ON STUDY FOR INCREASE OF TRAIN SPEED ON JAVA INDONESIA

1. Foreword

This study was carried out at the request of ECAFE and the Government of Indonesia made to the Government of Japan, for tendering technical advice and demonstrating how to evaluate the track and rolling stock of the trunk-lines of the Indonesian State Railways on Java, in order to increase the maximum train speed up to 100 km/h, as part of the transportation system development plan of the Railways.

The Government of Japan entrusted this study to the Overseas Technical Cooperation Agency (OTCA). In view of the importance of the study, OTCA deputed its execution to the Japan Railway Technical Service (JARTS).

The study team formed by JARTS flew to Indonesia leaving Tokyo on June 4th, 1973, carried out required investigations and tests in Indonesia under cooperation of PJKA, and returned to Japan on July 4th, 1973, calling at the ECAFE Secretariat on the way to make a provisional report on the results of the study to the Secretariat.

Study was made mainly on the Southern trunk line of PJKA, and this report is prepared based on the test results together with the data and materials which were presented to the study team during their stay in Indonesia.

2. Dispatchment of study team

2.1 Composition of study team

Seinosuke ARAI (Chief of team)	Chief, Car Dynamics Laboratory, Railway Technical Research Institute, Japanese National Railways
Yasuo KATO (Track evaluation)	Chief, Track Materials Laboratory, Railway Technical Research Institute, Japanese National Railways
Shinichi TANAKA (Car dynamics)	Chief researcher, Car Structure Laboratory, Railway Technical Research Institute, Japanese National Railways

Goto HIRATA (Track measurement)	Chief researcher, Track Laboratory, Railway Technical Research Institute, Japanese National Railways
Tsutomu YOSHIDA (Car structure)	Secretary General, Japan Railway Technical Service
Eiichi MUTSURO (Co-ordinator)	Staff, Development Surveys Department, Overseas Technical Cooperation Agency

2.2 Schedule of study team (1973)

Date	Schedule	Activities, etc.	Accommodation at
June 4 (Mon)	Tokyo–Jakarta (by air)	Flight JL 719	Jakarta
" 5 (Tue)		Visit to Japanese Embassy & OTCA for preparatory talk and courtesy call at Land Transport and Inland Waterways, MOT	"
" 6 (Wed)	Jakarta–Bandung (by rail)	Courtesy call at PJKA Head Office	Bandung
" 7 (Thu)		Conference on contents of investigation and investigation schedule with PJKA at PJKA Head Office	"
" 8 (Fri)		Same as above, plus collection of data	"
" 9 (Sat)	(Bandung–Jakarta) (by rail)		" (Mr. Mutsuro stayed at Jakarta)
" 10 (Sun)		Sorting & arrangement of data collected	"
" 11 (Mon)		Adjustment of details of schedule & collection of additional data at PJKA Head Office	"
" 12 (Tue)	Bandung–Jakarta (by rail)	Conference on schedule with West Railway Division at Jakarta station	Jakarta
" 13 (Wed)		(Rolling stock team) Preparation for short-distance test (Track team) Static measurement of track on short-distance test section	"
" 14 (Thu)		"	"
" 15 (Fri)		Short-distance test; (Track team) Dynamic measurement at the site of test	"
" 16 (Sat)		Consolidation of test results	"
" 17 (Sun)	Jakarta–Jogjakarta (by air)	Conference with Central Java Region about schedule	Jogjakarta
" 18 (Mon)	Jogjakarta–Madiun (by rail)	Long-distance test	Madiun

Date	Schedule	Activities, etc.	Accommodation at
" 19 (Tue)	Madiun–Surabaya (by rail)	Long-distance test	Surabaya
" 20 (Wed)		Collection & analysis of results of test	"
" 21 (Thu)	Surabaya–Jogjakarta (by rail)	Forwarding the test train (without test) to Jogjakarta	Jogjakarta
" 22 (Fri)	Jogjakarta–Madiun Madiun–Jogjakarta (both by rail)	High-speed test	"
" 23 (Sat)		Check of test results with Central Java Region; Collection of results of test	"
" 24 (Sun)		Analysis of test result; Putting in order of data	"
" 25 (Mon)	Jogjakarta–Jakarta (by air)	AM: Preparation of provisional report PM: Return to Jakarta	Jakarta
" 26 (Tue)		Preparation of provisional report Call at Japanese Embassy to make report	"
" 27 (Wed)	Jakarta–Bandung (by rail)	Team leader and 2 members: Study of test results at PJKA Head Office Other members: Preparation for shipment of measuring instruments, etc. to Japan	Bandung Jakarta
" 28 (Thu)	PM: Bandung–Jakarta (by rail)		Jakarta
" 29 (Fri)		Call at Land Transportant and Inland Waterways, MOT to make provisional report & explanation	"
" 30 (Sat)		Farewell courtesy call at Japanese Embassy, and Indonesian organs concerned	"
July 1 (Sun)		Preparation for return to Japan	"
" 2 (Mon)	Jakarta–Bangkok (by air)	Flight CX700; Call at Japanese Embassy for reporting	Bangkok
" 3 (Tue)		Call at ECAFE Secretariat to make provisional report and explanation	"
" 4 (Wed)	Bangkok–Tokyo (by air)	Flight JL716	

2.3 **Personnel concerned**

(1) ECAFE

Mr. S. Masood Husain	Chief, Transport and Communications Division
Mr. Aldo Manos	Chief, Technical Assistance Unit
Mr. M. E. Saleh	Chief, Railway Section, Transport and Communications Division
Mr. Shwe Shane	Regional Railway Network Advisor, Transport and Communications Division
Mr. F. Löhr	Railway Expert, Transport and Communications Division
Mr. V. N. Timopheyev	Economic Affairs Officer, Transport and Communications Division

(2) Ministry of Transport of Indonesia

Mr. Sumpono Bayuaji	Director General of Land Transport and Inland Waterways
M. Sc. Soerono	Superintendent of Secretariat, Land Transport and Inland Waterways

(3) Indonesian State Railways

General information

Ir. R. Soemali	Chief Director, Bandung Head Office
Ir. Achmad Rochaeli Ardiwikarta	Director, Supply and Workshops, Bandung Head Office
Mr. Imam Rustadi S.H.	Secretary, Bandung Head Office

Preparation and execution of tests

Research and Development Institute, Bandung Head Office:

Ir. Partosiswojo	Chief
Mr. Soepar	Chief, Mechanical Engineering Department
Drs. M. Subyanto	Chief, Industrial Engineering Department

Planning and Control Institute, Bandung Head Office:

Ir. Sandjojo	Chief
Ir. Boedijoewono	Deputy Chief

Way and Works Department, Bandung Head Office:

Ir. Soeharso	Chief
Ir. Sajid	Deputy Chief
Ir. Tantular	Chief, Project and Extension Division
Mr. R. Mohamad Soegiarto	Chief, Railway Track Section

Contacts during field tests

Ir. Pantiarso	Manager, West Java Region
Ir. Moerhadi	Manager, East Java Region

3. Investigations and tests

3.1 Purpose

The purpose of this study is to indicate concretely, to the Indonesian State Railways (PJKA), the ways and means to increase the train speed on its trunk-lines on Java, Indonesia, to max. 100 km/h. To be more concrete, it is to indicate the method to decide the maximum train speed technically, after executing the running tests with the existing track and rolling stock and checking effects on the vibration characteristics of the carbody of the rolling stock.

3.2 Kinds of investigations and tests conducted

3.2.1 Investigation items

(a) Track

The existing track structure was checked on the design drawing. On the short-distance test section, measured were track irregularities, rail subsidence and rail wear using the measuring instruments which were carried to Indonesia.

In the long-distance test and high-speed test, investigation was made by eye observation from the inspection car coupled to the test train at its rear end.

(b) Rolling stock

The design drawing of trucks was checked. Measurement by sketching was made of the profiles of wheel tyres of the rolling stock selected for test using the measuring instruments which were carried to Indonesia to determine their wearing condition.

3.2.2 Test items

The following three kinds of running test were carried out:

(a) Short-distance test

(b) Long-distance test

(c) High-speed test

3.3 Test-run sections

3.3.1 Short-distance test section

The short-distance test section was selected for 2 km of the track for trains bound for Jakarta between Jakarta and Cikampek (double-track).

The selection was made based on the requirement that the test train could be operated at high speeds to allow to check the relation between vibration of the rolling stock and their speed, especially their high-speed operation.

Among the 2 km stretch of the test section chosen, the sections where measurement and test of the track were made are as follows:

	Kilometerage from Jakarta
Straight section	22 km 770–23 km 270
Curved section	23 km 280–23 km 735

Particulars of the curved section are as follows:

Radius of track curvature:	900 m
Starting point of transition curve:	23 km 281
Starting point of circular curve:	23 km 347
End point of circular curve:	23 km 669
End point of transition curve:	23 km 735
Cant:	65 mm

3.3.2 Long-distance test section

The long-distance test section was selected for about 322 km between Jogjakarta and Surabaya (single track). This selection was made based on the requirement that the selected section should include various kinds of representative track structure to allow to check interrelation between the rolling stock and track.

The test section selected is the South trunk-line stretching from Jogjakarta to Surabaya via Solo running through a flat land with gentle gradient and large radius of curvature, excepting several sharp curves located near major stations.

The roadbed is in just ordinary or good condition and there is no tunnel on this section.

The track structure of the test section is as shown in Table 1.

3.3.3 High-speed test section

As for the high-speed test section, four sections with a total length of about 29 km were selected between Jogjakarta and Madiun after checking the result of the long-distance test. This selection was made with the purpose of finding out ways and means to increase the train speed with the existing track and rolling stock.

3.4 Test cars

Three passenger cars and one locomotive were especially selected for tests. Measurements were made on the three passenger cars in the short-distance test and on two passenger cars of the three and the locomotive in the long-distance test and high-speed test. Table 2 shows the main particulars of the test cars.

Table 1 Track structure of test sections

Kind	Short-distance test section (2 km)	Long-distance test section (322 km)					
		High-speed test section (156 km)					
Section	Jakarta–Cikampek (2 km)	Jogjakarta – Solo (59 km)			Solo – Madiun (97 km)		Madiun–Surabaya (166 km)
Section No. & kilometerage		(I) 165~156	(II)-1 121~117	(II)-2 117~113	(III) 220~210	(IV) 191~188	
Rail weight (kg/m)	41.52 (R14)	38.0 (15P)	38.0 (15P)	41.52 (R14)	41.52 (R14)	42.59 (R14A)	41.52 (R14)
Rail length (m)	85	14	14	85	85	85	85
Rail joint	Suspended joint Even joint	"	"	"	"	"	"
Rail fastening	Rigid fastening 3 screw spikes per tie plate	Rigid fastening 3 screw spikes per tie plate	Rigid fastening	Rigid fastening	Rigid fastening 3 screw spikes per tie plate	Pandrol	Rigid fastening 3 screw spikes per tie plate
Sleeper pitching (cm)	68	65~70	70	70	68	68	65
Kind of ballast	Crushed stone	Screened gravel	Crushed stone	Crushed stone	Crushed stone Screened gravel	Crushed stone	Crushed stone
Ballast thickness (cm)	10~20	5~15	5~10	5~10	5~15	5~15	10~30
Max. gradient (o/oo)	–	10			5		7
Min. radius of curvature (m) (Note)	900	–			900		500
Present max. speed (km/h)	90	70			75		75, 80

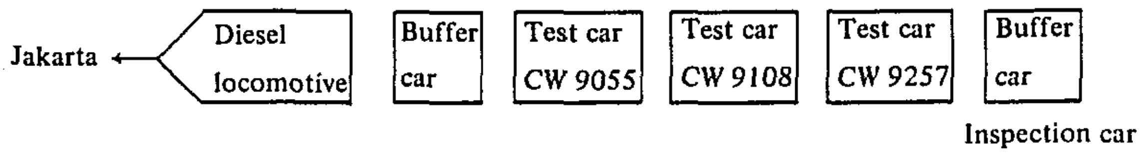
Note: Min. radius of curvature excludes several sharp curves located near major stations.

Table 2 Specification of test cars

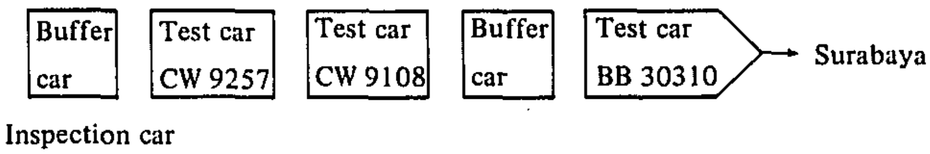
Classification Car number		2nd class coach (No. 1) CW 9055	2nd class coach (No. 2) CW 9108	2nd class coach (No. 3) CW 9257	Diesel locomotive BB 30310
Manufactured by		Simmering-Graz-Pauker A.G.	Nippon Sharyo Seizo Kaisha Ltd.	VEB Waggon Bau Bautzen	Rheinstahl Henschel AG
Year of delivery		1963	1963	1965	1973
Type		2-bogie	2-bogie	2-bogie	2-bogie (B-B)
Total weight (ton)		23.90	29.20	29.00	42.80
Length (m)		20.920	20.920	20.920	12.160
Max. height (m)		3.595	3.610	3.715	3.690
Max. width (m)		2.990	2.990	2.990	2.800
Length between bogie centers (m)		14.000	14.000	14.000	5.800
Track gauge (m)		1.067	1.067	1.067	1.067
Seat capacity (person)		80	80	80	–
Truck	Bogie type	Ferrostaal-type bogie	NT-11-type bogie	Görlitz-type bogie	Henschel-type bogie
	Number of axles	2	2	2	2
	Wheel base (m)	2.360	2.200	2.200	2.200
	Wheel diameter (m)	0.774	0.774	0.774	0.904
	Axle spring	Coil spring	Coil spring	Coil spring	Coil spring
	Bolster spring	Laminated spring	Coil spring	Coil spring	Coil spring
	Oil damper	None	For bolster (Vertical)	For bolster (Vertical & Horizontal)	For primary and secondary springs (Vertical)
	Horizontal damping	Swing bolster hanger	Swing bolster hanger	Lateral damper	None
	Axle-box suspension		Pedestal type	Pedestal type	Pedestal type
	Journal bearing	Roller	Roller	Roller	Roller
Note					Hydraulic

3.5 Composition of test train

(1) Short-distance test



(2) Long-distance test and high-speed test



3.6 Test running speed and method of test train operation

(1) Short-distance test

The short-distance test was conducted repeatedly up and down on the short-distance selected for the purpose. The train was operated, for each run, in a manner allowing the train to run the test section at a pre-determined speed.

The test run was made 6 times, changing the speed from 60 km/h to 100 km/h in the direction to Jakarta.

(2) Long-distance test

The long-distance test was conducted one time, letting the test train run, at the speed specified in the existing train diagram. The measured maximum speed was 84 km/h.

(3) High-speed running test

The test train was operated on the four test sections selected based on the result of the long-distance test at the speed higher than the speed in the long-distance test by 10 to 20 km/h. The test train was operated in the direction, Jogjakarta to Surabaya, the same as for the long-distance test. The measured maximum speed of the test train was 92 km/h.

3.7 Items measured and measuring methods

3.7.1 Items and methods of track measurement

For the short-distance test section, the following items were measured on the track:

Track gauge

Cross level

Longitudinal level

Alignment

Dynamic settlement of rail

Wear of rail

Measurements were made by setting measuring points every 5 m covering whole the curved section and 500 m for the straight section. At each measuring point, the track gauge was used for the measurement of gauge and cross level. And, for the longitudinal level and alignment, measurement was made using the longitudinal level and alignment measuring instrument by reading the amount of irregularity at the middle point of a base line of 10 m long.

The dynamic settlement of the rail was measured, for the straight section, by establishing 4 measuring points near the rail joint and at the middle of the rail by fixing the settlement gauge.

The wear of rail was measured at 3 points each for the straight section and curved section, by using rail section meter.

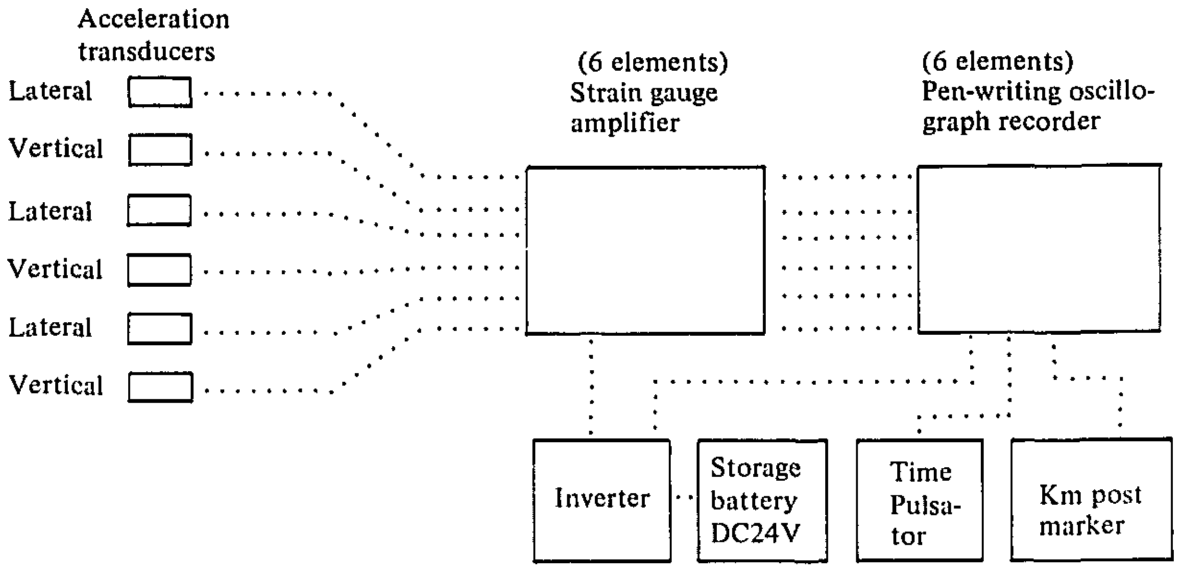
3.7.2 Items and methods of on-the-car measurements

(1) Measurement of profile of wheel tyre

Measurement was made on all wheels of the front-side trucks of all test passenger cars. The wheel tyre profile charting instrument was fixed on the wheel to record tread profile by moving pen by hand.

(2) Measurement of carbody vibration acceleration

The vertical acceleration and lateral acceleration of the car body vibration were measured, for the test cars, at the position of their front truck. The electric connection of the measuring instruments and recording device was as shown in the following block diagram:



3.8 Measuring instruments

3.8.1 Track measurements

The following measuring instruments were used for track measurements:

Track gauge, cross level;	Track gauge
Longitudinal level, alignment;	Track alignment and longitudinal level gauge
Dynamic settlement of rail;	Rail settlement gauge
Wear of rail;	Rail wear gauge

3.8.2 Measurement of rolling stock

The following instruments were used for measurement of rolling stock:

Profile of wheel tyre;	Wheel tread profile drawing instrument
Car body acceleration;	Acceleration transducer (6 sets),
	Strain-gauge amplifier (6 elements),
	Pen-writing oscillograph (6 elements)
Time;	Time pulsater
Km post position;	Electric contacts
Others;	Rotary inverter (as power source of 300 W)

4. Dates of investigations and tests

The following is the list of the dates on which investigations and tests were conducted:

Dates	Kinds of investigations and tests	Sections
June 13 (Wed)	Measurement of track (static)	
" 14 (Thu)	Measurement of rolling stock (standing)	
" 15 (Fri)	Short-distance test Measurement of track (dynamic)	Cikampek to Jakarta
" 18 (Mon)	Long-distance test	Jogjakarta to Madiun
" 19 (Tue)	"	Madiun to Surabaya
" 22 (Fri)	High-speed test	Jogjakarta to Madiun

5. Results of investigations and tests

5.1 Investigation of track

5.1.1 Short-distance test section

The results of measurement of static track irregularities of the short-distance test section, i.e. track gauge, cross level, longitudinal level and alignment, are shown in Fig. 1 and Fig. 2.

Table 3 shows the allowable static track irregularities prescribed by PJKA, RSR and JNR. The allowable irregularities of longitudinal level and alignment are larger than those of RSR and JNR.

Table 3 Allowable static track irregularities

	PJKA	RSR	JNR
Track gauge	–2 ~ +5 mm	–3 ~ +6 mm	–4 ~ +6 mm
Cross level	7 mm	8 mm	9 mm
Longitudinal level	13 mm	10 mm	9 mm
Alignment	15 mm	–	9 mm

Note: The figures of JNR are for the third-grade line.

Fig. 1 Track irregularities in the straight line part of short-distance test section (22K770-23K270)

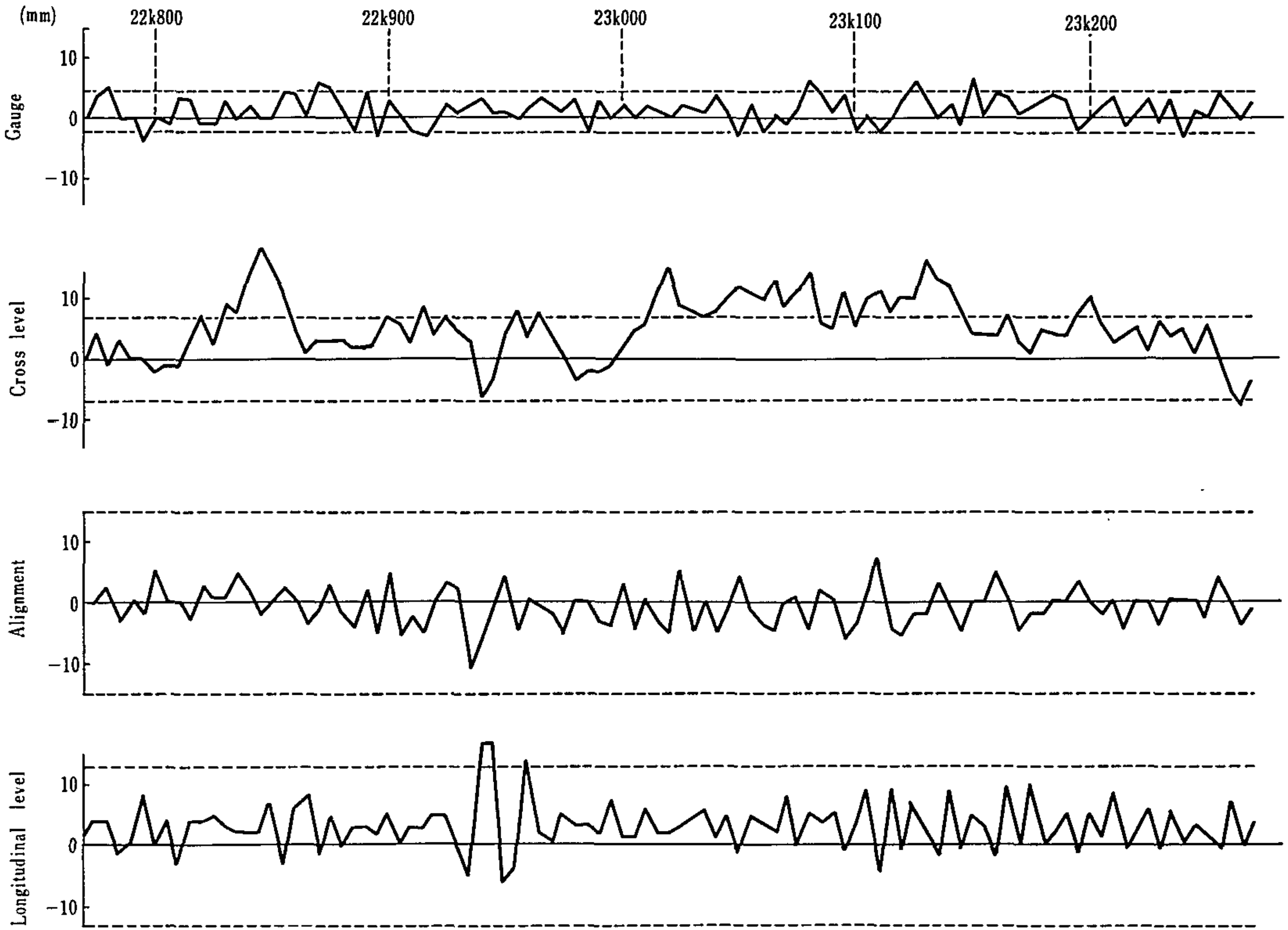
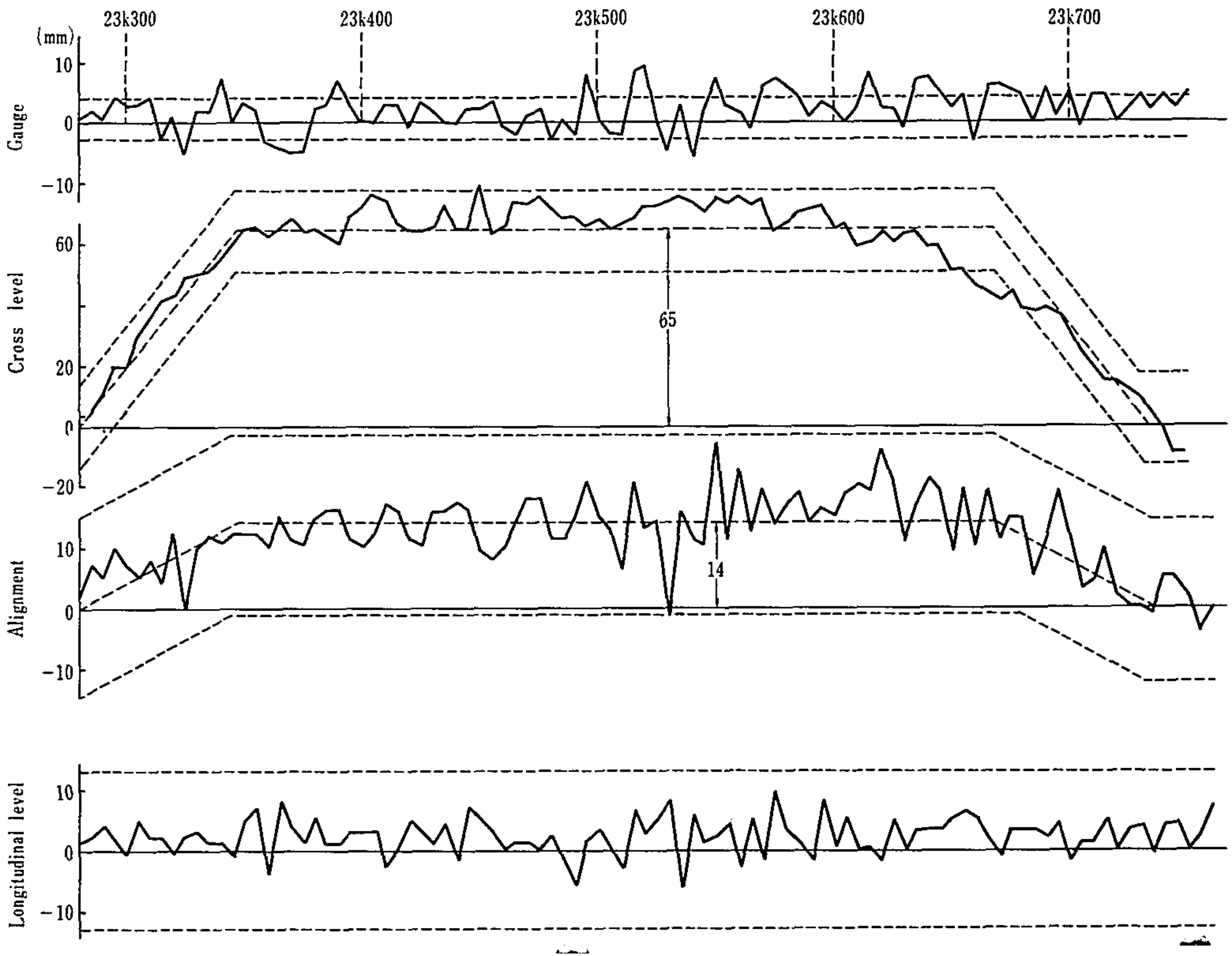


Fig.2 Track irregularities in the curved line part of short-distance test section
(23K280-23K735)



(1) On the straight section, cross level is generally bad, exceeding the allowable value largely at many points. The track just in front and rear sides of bridge is found settling, i.e. the track is sagging largely causing a large irregularity in longitudinal level.

(2) On the curved section, the track gauge and cross level are exceeding the allowable values at many points. Although the longitudinal level and alignment are almost within the allowable values, many are found to be near the limit.

(3) Dynamic settlements of rail are within the range of 3.5 mm ~ 4.3 mm. But, as only a few measurements were made, it was impossible to determine clearly the difference in the value between the rail joint area and rail center area. Further, the train-speed effect was not seen in the dynamic settlement of rail.

(4) The wearing condition of rail (manufactured in 1961) is as shown in Fig. 3 and Fig 4. Corrugation of 40 mm in wave length is seen covering the whole straight section.

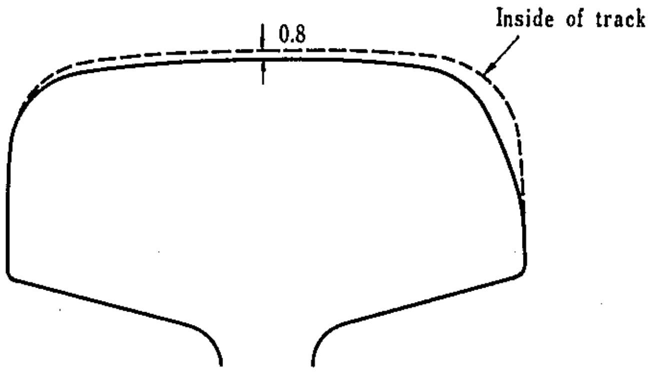
5.1.2 Long-distance test section

On the long-distance test section, the results of on-the-car observation and measurement covering the whole section are as follows:

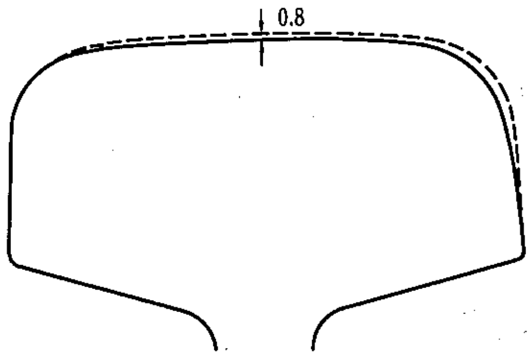
(1) Although steel sleepers are partly used, adopted are, for the most part, wooden sleepers made of teak wood with rigid fastenings having ample strength against side thrust of wheels.

(2) Supplement of ballast is insufficient for the whole section. Soil is mixed into the ballast in a large quantity, the ballast thickness under sleeper is insufficient, and the track is settling in general.

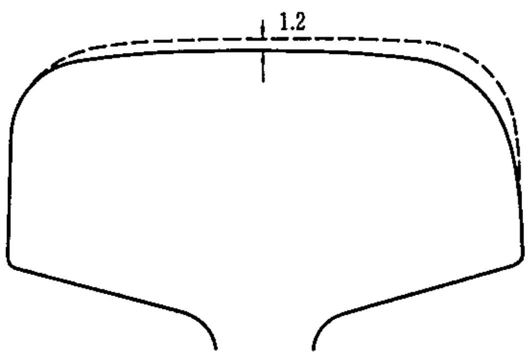
Fig.3 Cross-section of rail (straight section)



Near the rail joint

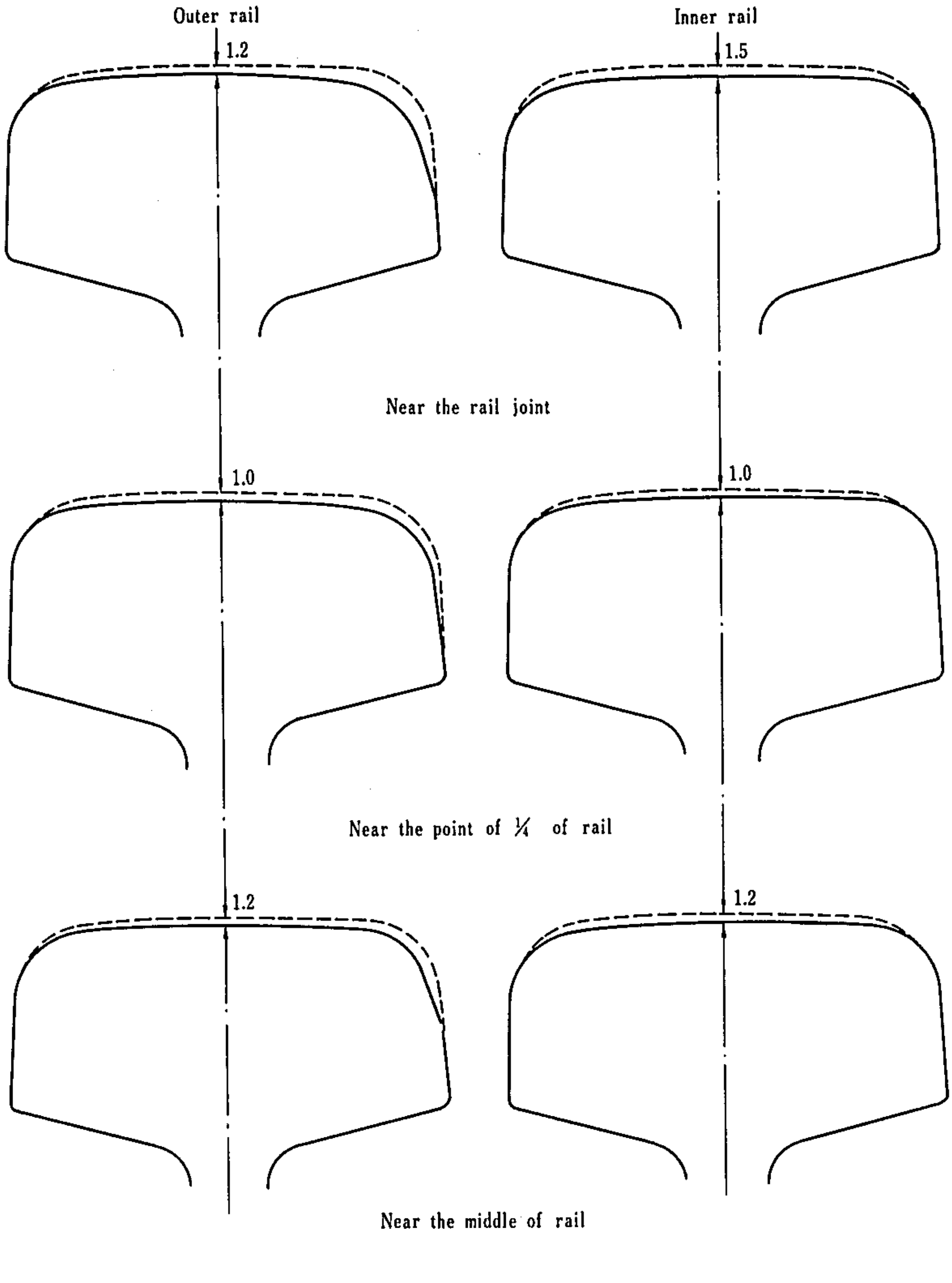


Near the point of ¼ of rail



Near the middle of rail

Fig.4 Cross-section of rail (curved section)



(3) As for aged rail joints, depression of joints and solidification of ballast by mud-pumping are seen at many places.

(4) At places just in front and on the rear side of permanent way structures, especially at embankment just in front and rear sides of bridge, ballast is seen settling exceedingly, causing the track to settle for a large length in a sagging fashion. Irregularities are noticed in longitudinal level of the track near bridges.

(5) Near level crossings, large irregularities in alignment and longitudinal level are noticed.

(6) It was noticed that oil from passing trains had stained the track at many places for a long distance.

(7) Wave-like corrugation of rail was seen at many places for a long distance. For this reason, trains passing on such places are causing considerable shocks and noises at high speeds.

5.2 Investigation of test cars

5.2.1 Profile of wheel tyres

The profile of wheel tyres of the test passenger cars is as shown in Fig. 6. Fig. 5 shows the standard profile of wheel tyre.

As for the wheel treads of No. 2 coach, excessive wear is noticed at the flange part. Wear of the flange part will increase lateral play of the wheel axle and thus affect lateral vibration of the car. Therefore, it is important to keep them within the limit.

As for the wheel treads of No. 1 coach, wear is noticed at the flange part and tread part. Wear of the tread part will finally enlarge the equivalent tread gradient. So, it may worsen the snake-motion characteristic of the car.

As for the wheel treads of No. 3 coach, comparatively little wear is noticed.

The limit of the wear of flange is 8 mm in PJKA, while it is 10 mm in JNR.

The limit of the wear of tread is 8 mm in both PJKA and JNR.

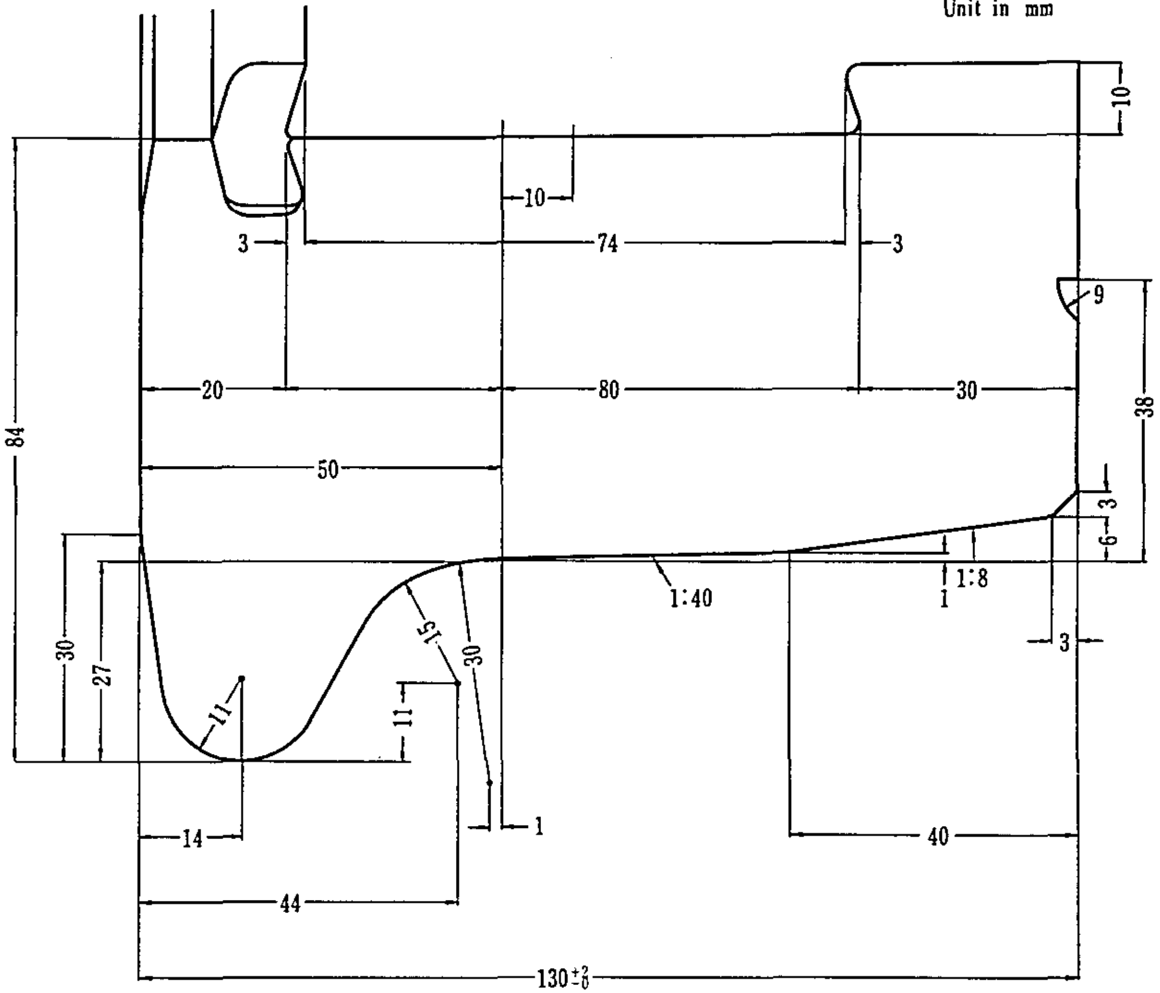
5.2.2 Truck structure and others

The vertical vibration characteristics of the test cars are given in Fig. 7. The method of calculation and values used for calculation are shown in APPENDIX III.

In general, when the composite spring constant combining bolster spring and axle spring is constant, the carbody vibration acceleration in the lower frequency range will become larger in relation to the value of κ which means the ratio between bolster spring constant and axle spring constant. Consequently the value of κ should be made smaller in order to suppress vibration in the lower frequency range.

Fig.5 Standard profile of wheel tyre

Unit in mm



Note : Diameter is 774mm.

Fig.6-1 Wheel tyre profile of No.1 coach (CW 9055)

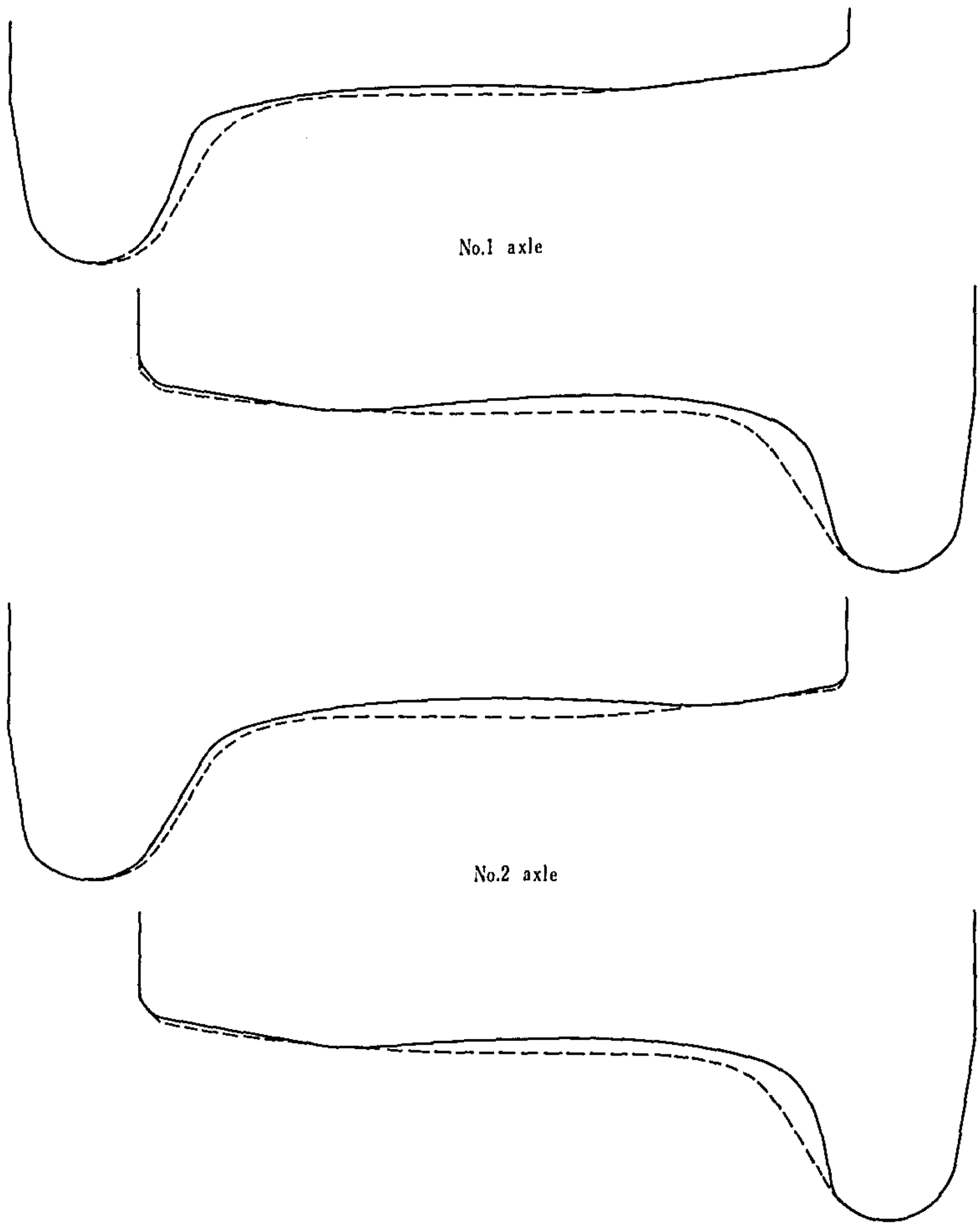


Fig.6-2 Wheel tyre profile of No.2 coach (CW 9108)

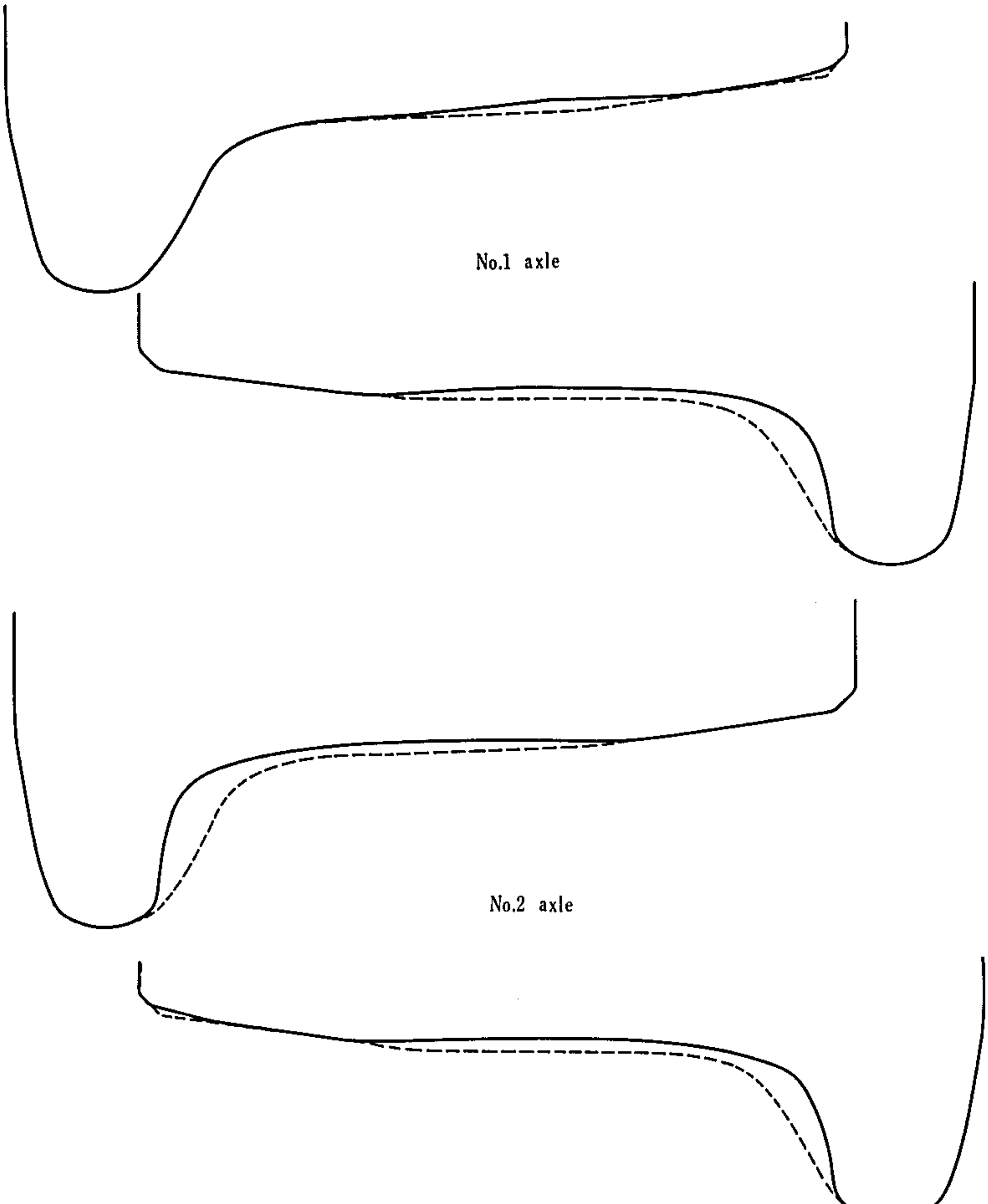


Fig.6-3 Wheel tyre profile No.3 coach (CW 9257)

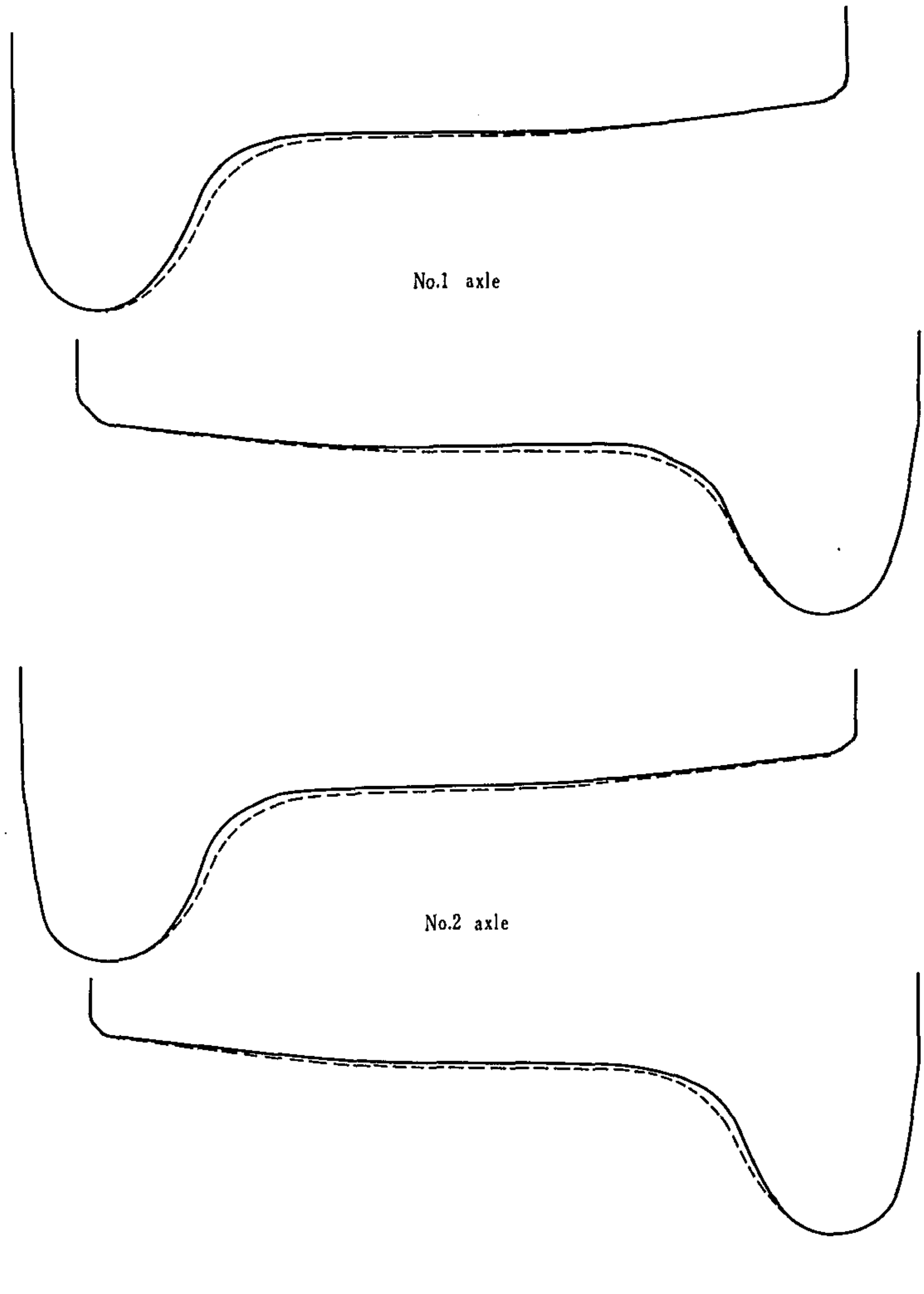
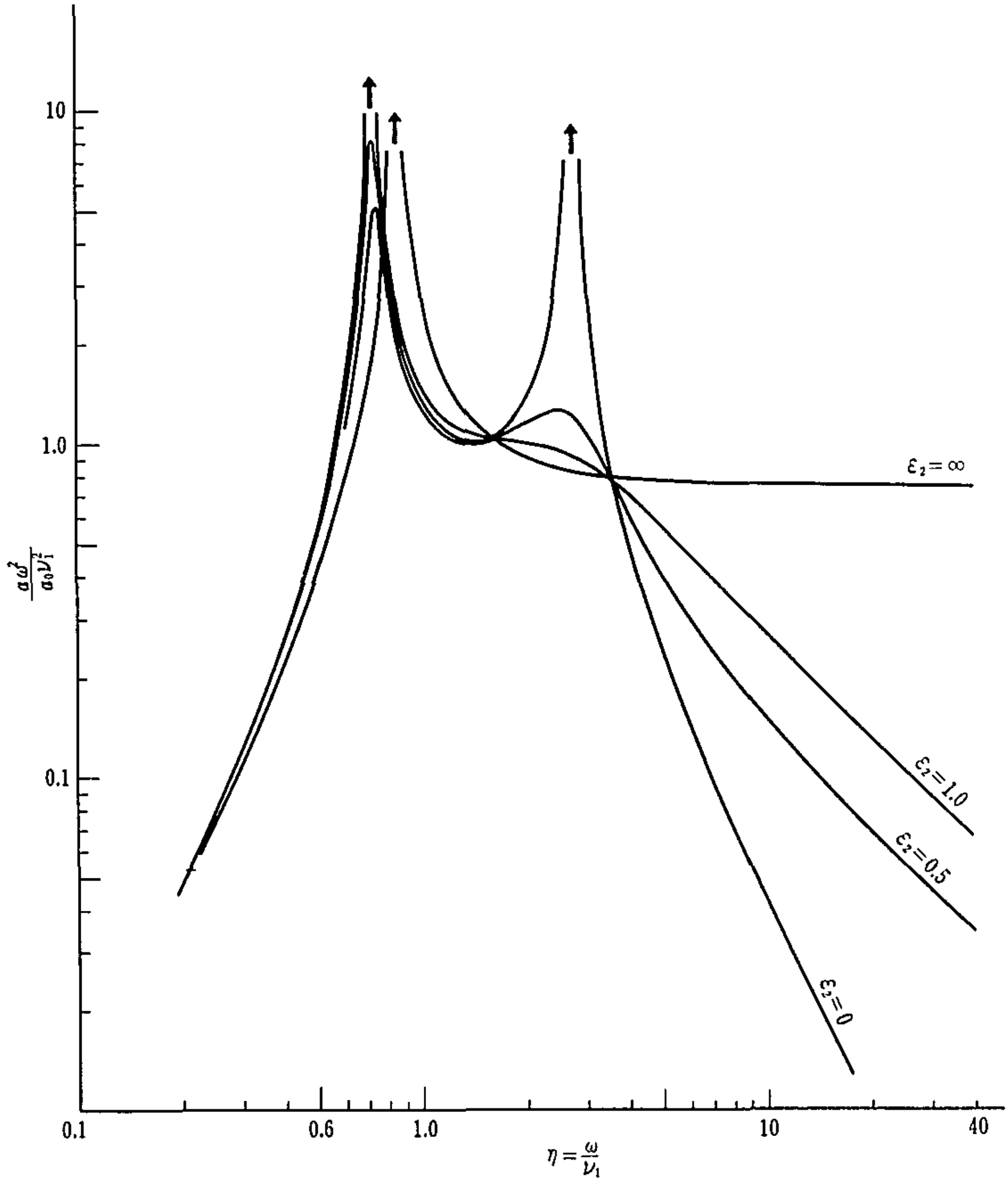
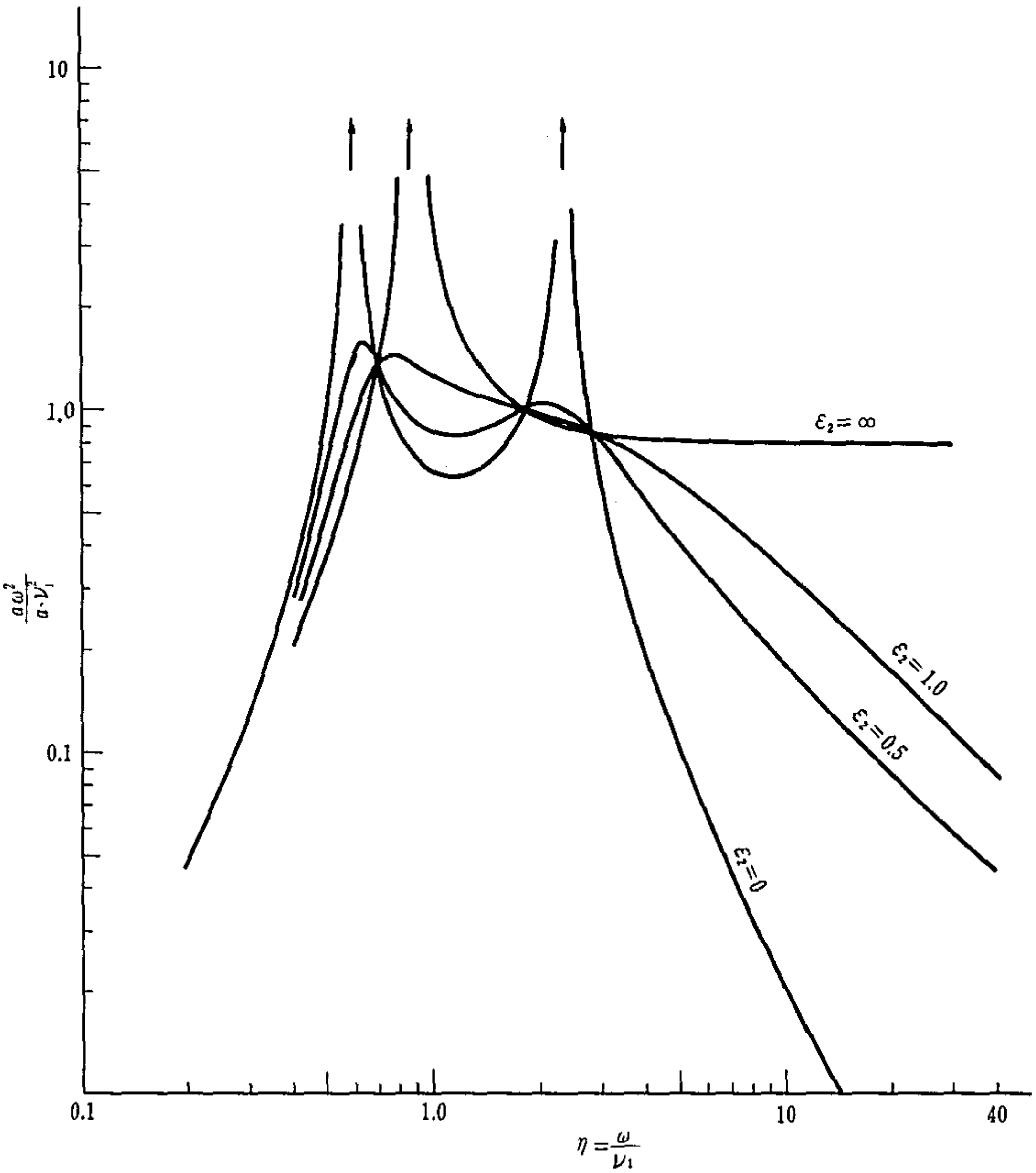


Fig.7-1 Vertical vibration characteristics of No.1 coach (CW 9055)



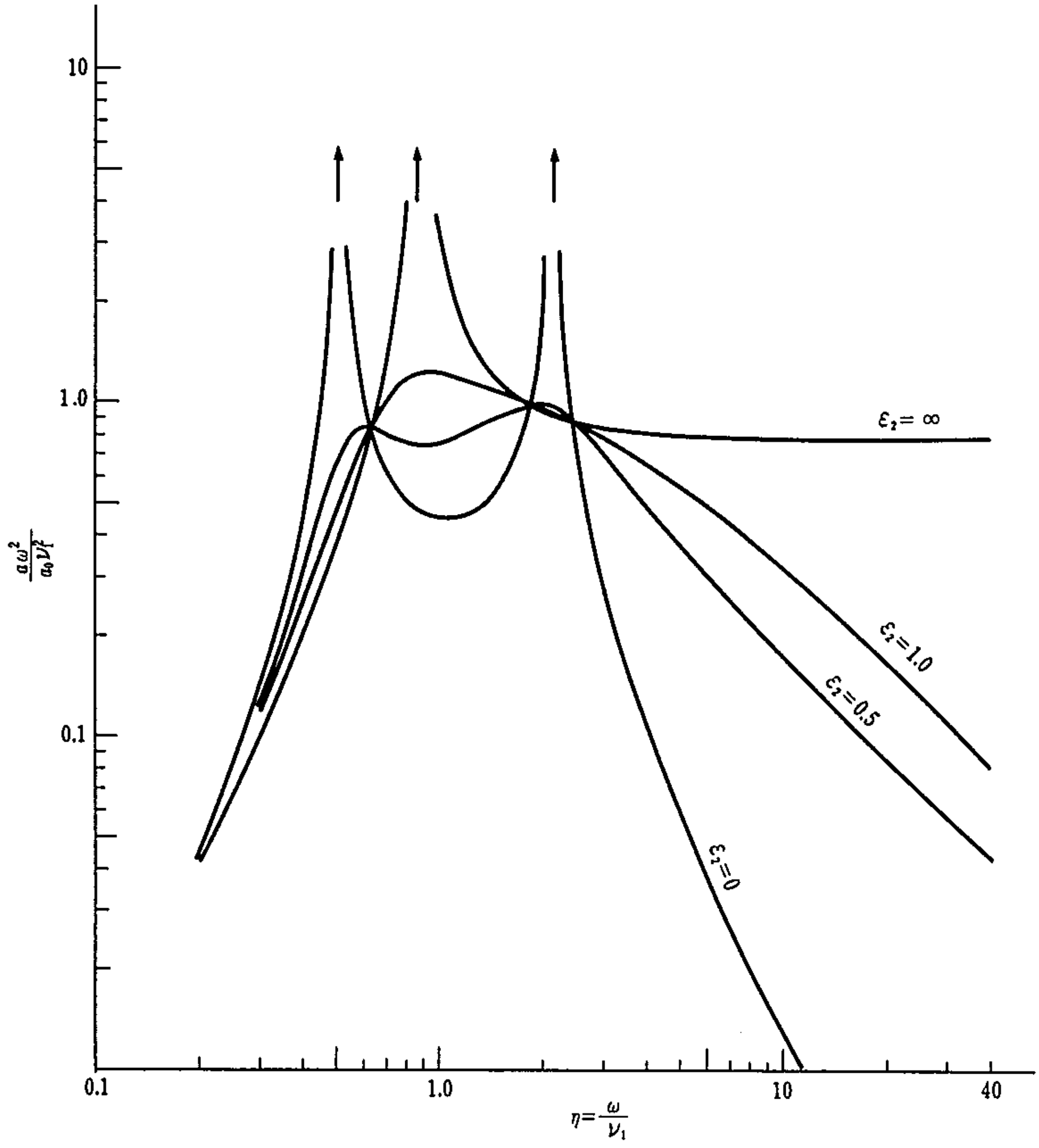
Note: Explanation of symbols is given in Page 118.

Fig.7-2 Vertical vibration characteristics of No.2 coach (CW 9108)



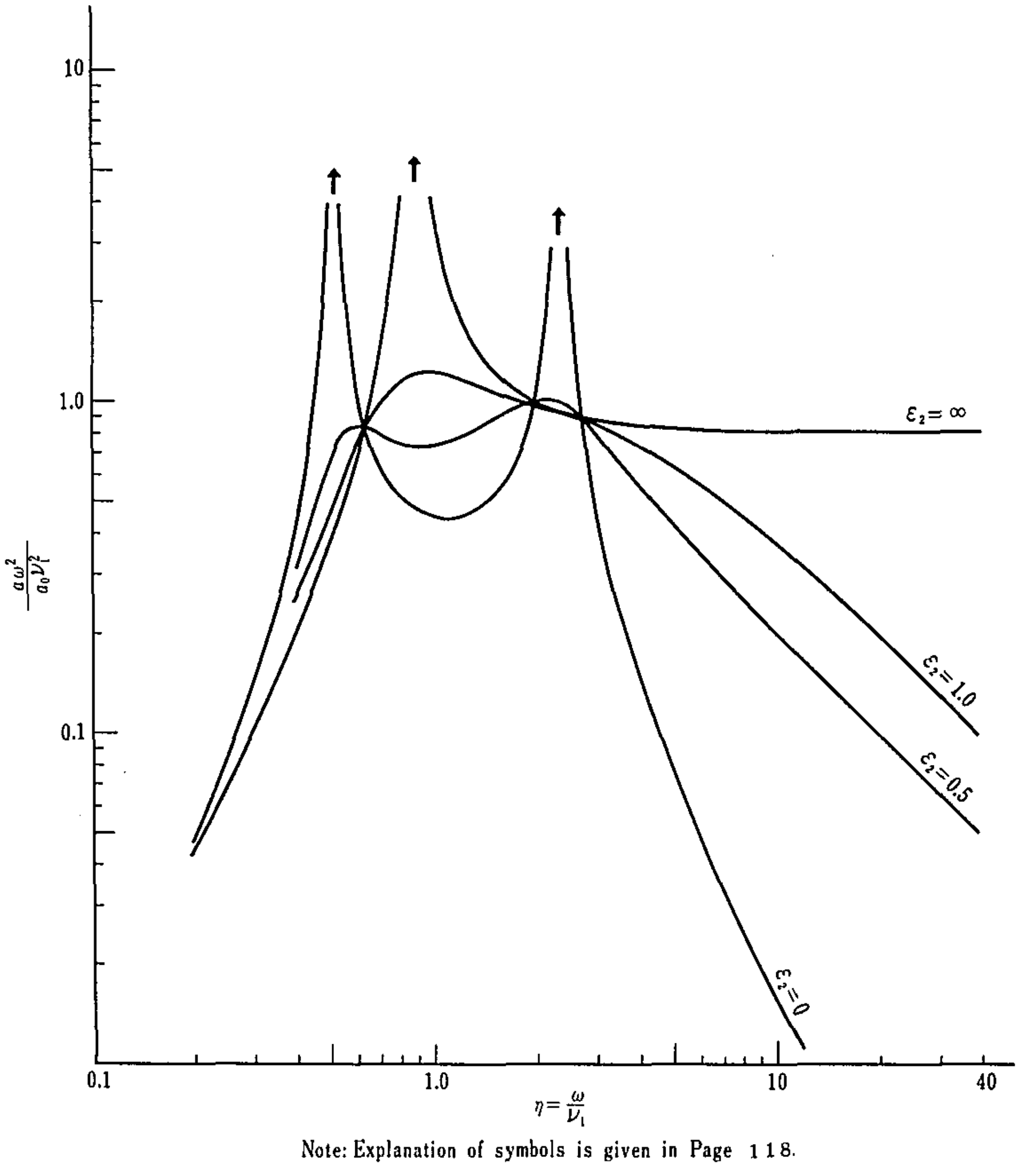
Note: Explanation of symbols is given in Page 118.

Fig.7-3 Vertical vibration characteristics of No.3 coach (CW 9257)



Note: Explanation of symbols is given in Page 118.

Fig.7-4 Vertical vibration characteristics of locomotive (BB 30310)



Note: Explanation of symbols is given in Page 118.

As the vibration induced by rail joints, etc. is considered to be in the lower frequency range, it is preferable to make the value of κ smaller.

As clearly seen in Fig. 7, the damper is especially effective to suppress large vibration. So, it is necessary to let the damper maintain its normal characteristics at all times.

5.3 On-the-car measurements

5.3.1 Short-distance test

(1) Comparison of vibration wave-forms

Fig. 8 shows the wave forms of the carbody vertical and lateral vibration accelerations of the 3 test cars measured in the short-distance test conducted on the straight section and curved section (R = 900 m) located at around 23 K. Fig. 8-1 is at the train speed of 65 km/h and Fig. 8-2 is at the train speed of 101 km/h.

Lateral vibration

Any of the test passenger cars has the component of high frequency in lateral vibration, but no tendency of generation of snake motion of trucks is noticed up to 100 km/h.

Vertical vibration

For No. 1 coach and No. 3 coach, large components of high-frequency vibration are noticed. For No. 2 coach, a tendency of increasing of high-frequency components is noticed as the train speed increases, though not so conspicuous as for No. 1 and No. 3 coaches.

For all passenger cars, a low-frequency vibration of 1.5–2.2 Hz is noticed continuously, and for No. 1 and No. 2 coaches, a tendency to cause frequently large vibration accelerations at higher speeds is noticed.

On and around the bridge located at the point 22 km 950, the vibration acceleration increases remarkably as the train speed increases.

(2) Relation between vibration acceleration and train speed

Fig. 9 shows the relation between the vibration acceleration and the train speed obtained for each passenger car, in the six running tests on the short-distance test section. In each case, the ordinate axis shows the maximum value of full amplitude of the acceleration wave-form, including the high-frequency vibration component, expressed in the unit of gravity acceleration g.

Lateral vibration

The lateral vibration acceleration is noticed having a tendency to increase monotonously as the train speed increases for all cars, whether on the straight section or on the curved section. The vibration acceleration increases to about 2 times when the train speed goes up from 60 km/h to 100 km/h.

Vertical vibration

The vertical vibration acceleration increases monotonously as the train speed increases for all cars, whether on the straight section or on the curved section. The vibration acceleration increases by about 30% when the train speed goes up from 60 km/h to 100 km/h.

However, on the bridge located near the point 22 km 950 (Fig. 9-4 to Fig. 9-6), the vibration acceleration is different from the ordinary straight and curved sections and increases in proportion to the square of the train speed, which may be caused by a large irregularity in longitudinal level of track.

No. 1 and No. 3 coaches show larger vibration accelerations compared with No. 2 coach, and tends to show an acceleration near 0.6 g at about 100 km/h.

5.3.2 Long-distance test

Fig. 10 shows the maximum values, for every 500 m, of the vertical and lateral vibration accelerations of the locomotive and No. 2 and No. 3 coaches, together with the train speeds measured in the long-distance test conducted between Jogjakarta to Surabaya.

The ordinate axis shows the maximum value of full amplitude of vibration including high-frequency component in the unit of gravity g, as the same with the case in Fig. 9.

In the comparison by vibration acceleration, the locomotive showed the smallest value and No. 3 coach the largest. In the comparison by test section, the Jogjakarta–Solo section showed the best result, which may be partly due to the maximum train speed being restricted at 70 km/h, being followed by the Solo–Madiun section and the Madiun–Surabaya section. Especially, on the Saradan–Bagor section and Djombang–Boharan section, many vertical vibration accelerations exceeding 0.5 g appeared probably due to the maximum speed being 80 km/h. Although, the magnitude of vibration acceleration does not solely correspond to the maintenance condition of track, at least the track condition at the points showing a large vibration must be worse than at other points, judging from the results on the short-distance test.

5.3.3 High-speed test

Four high-speed test sections were chosen based on the results of the vibration

Fig.8-1 Typical wave form of vibration (Cikampek→Jakarta)

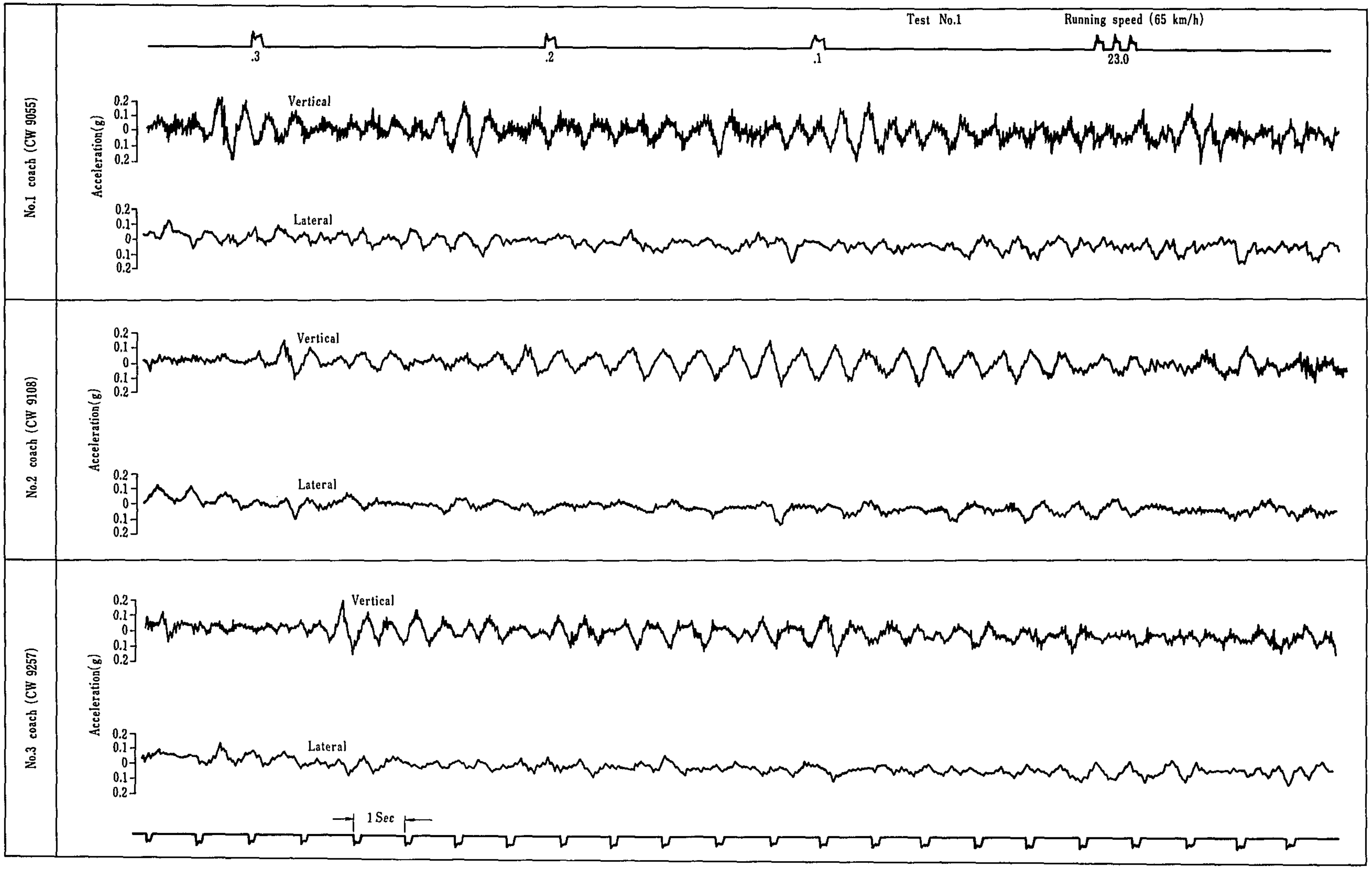


Fig.8-2 Typical wave form of vibration (Cikampek→Jakarta)

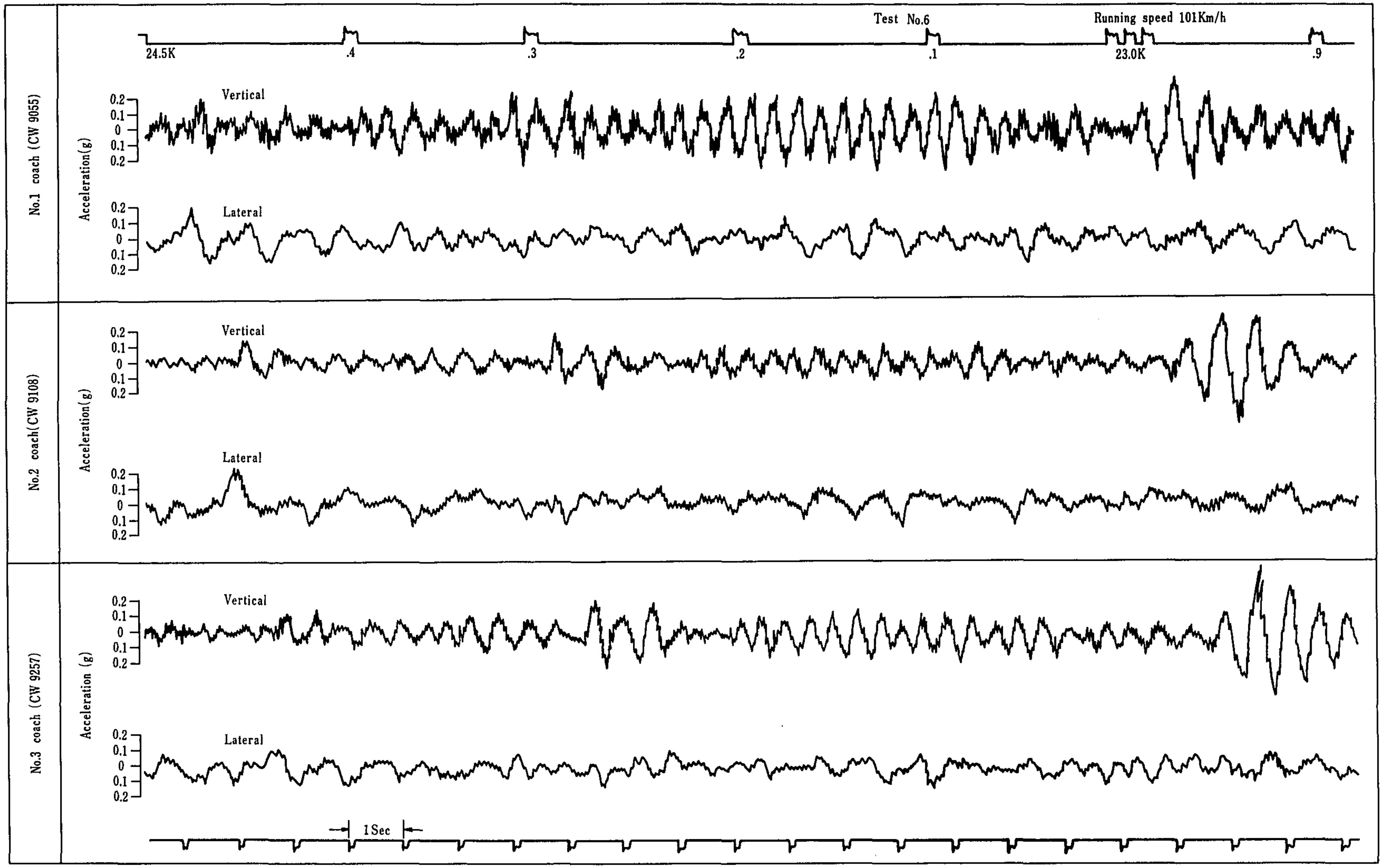


Fig.9-1 Maximum acceleration amplitude (straight and curved sections) of No.1 coach (CW 9055, Ferrostaal bogie)

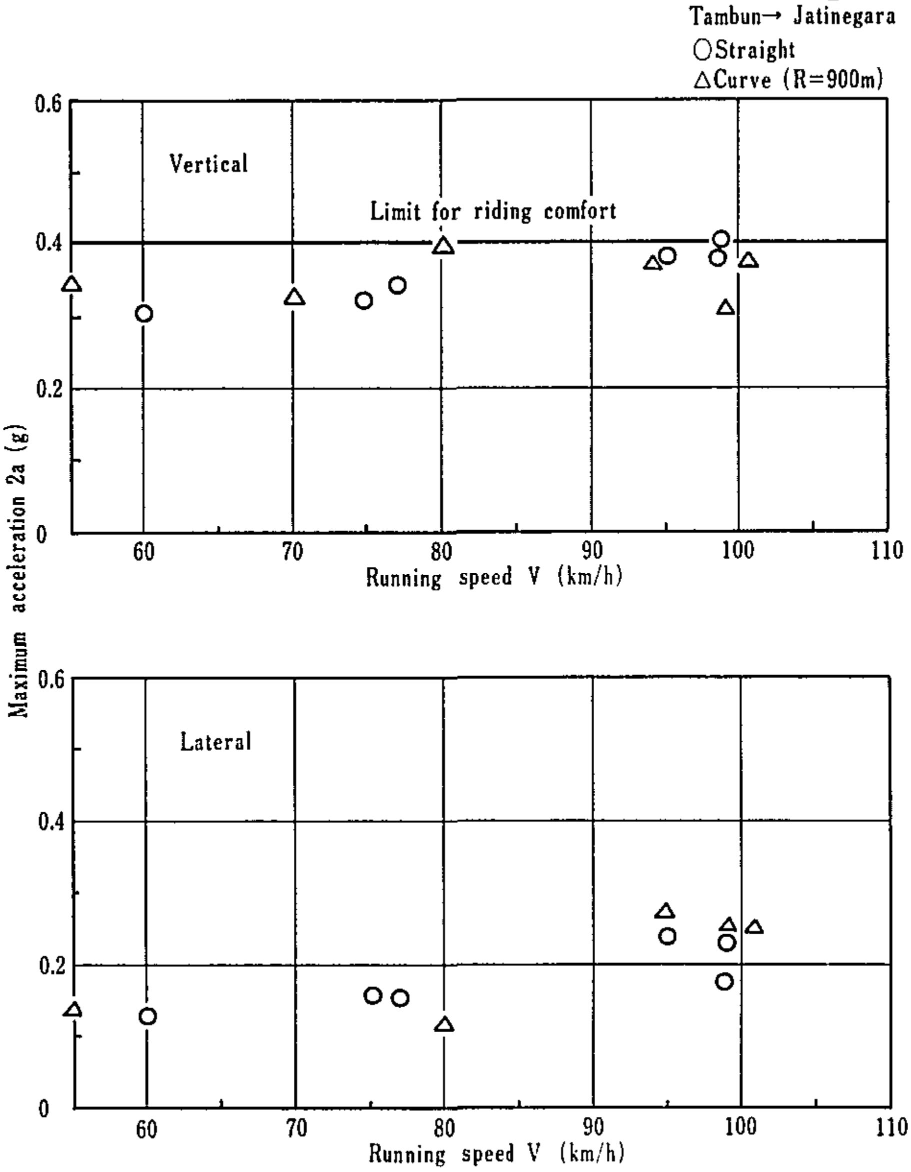


Fig.9-2 Maximum acceleration amplitude (straight and curved sections) of No.2 coach

(CW 9108, NT-11 bogie)

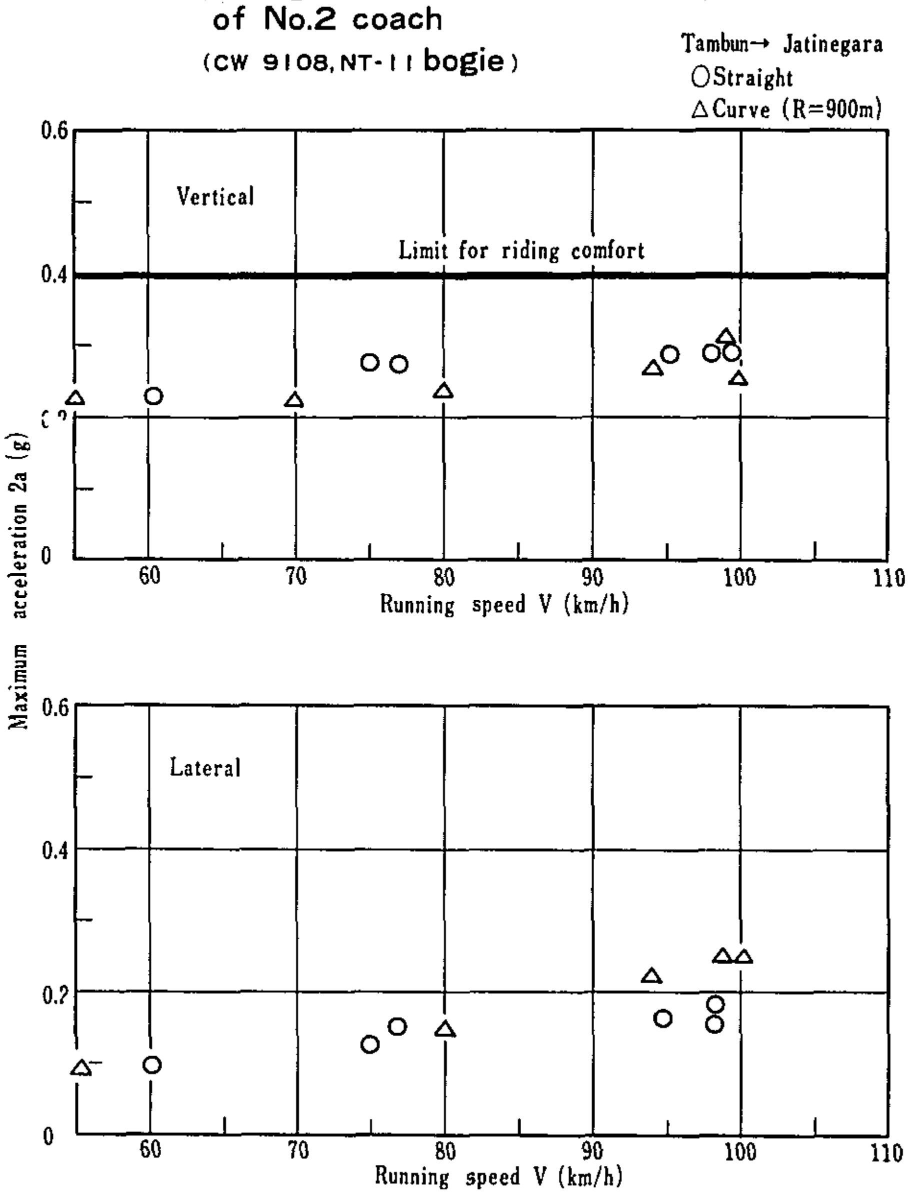


Fig.9-3 Maximum acceleration amplitude (straight and curved sections) of No.3 coach (cw 9257, Görlitz bogie)

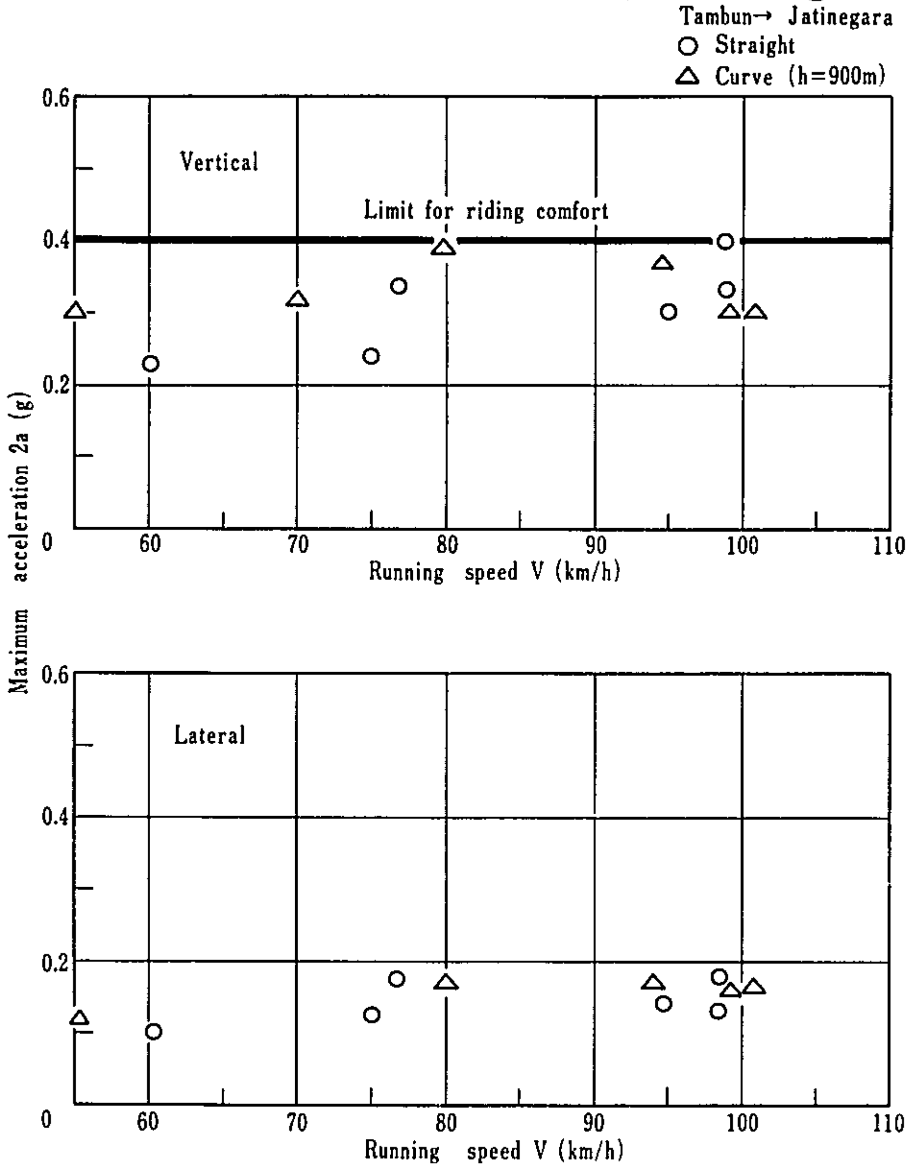


Fig.9-4 Maximum acceleration amplitude (bridge section) of No.1 coach (cw 9055, Ferrostaal bogie)

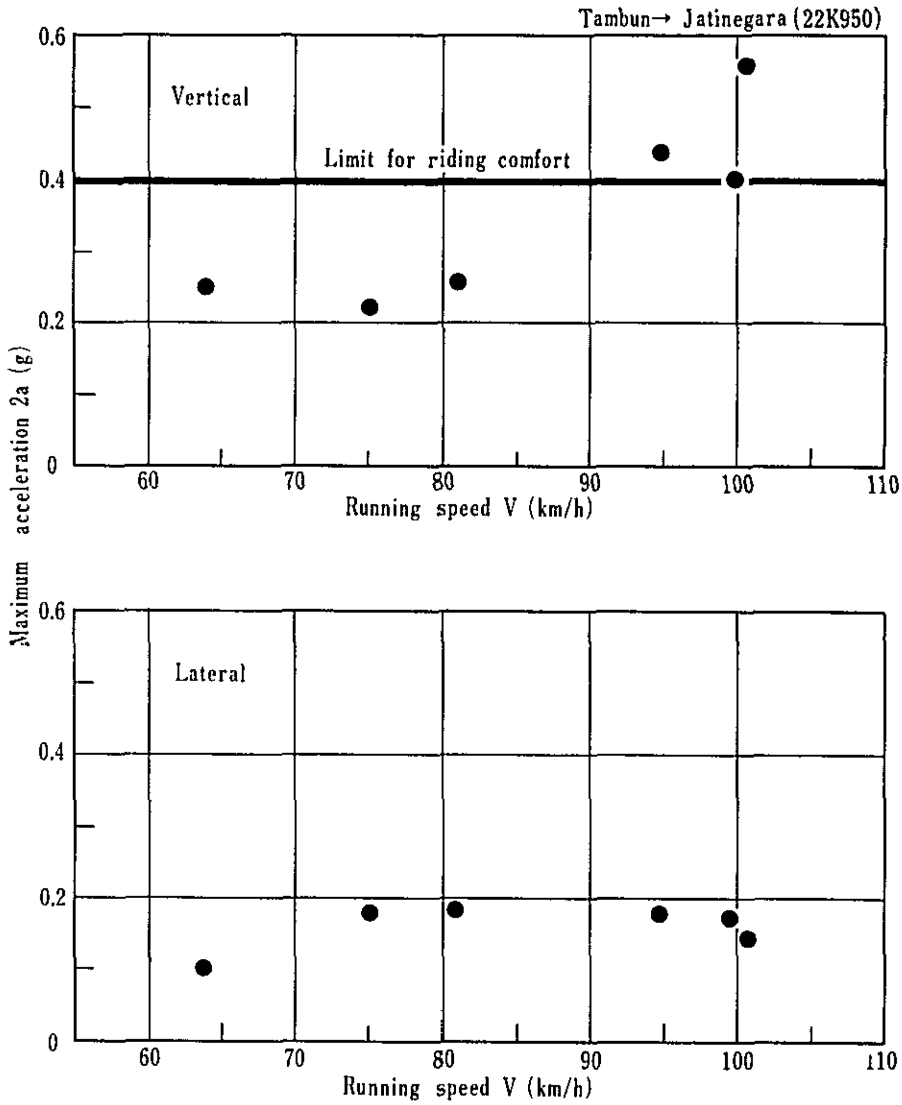


Fig.9-5 Maximum acceleration amplitude (bridge section) of No.2 coach (CW 9108, NT-11 bogie)

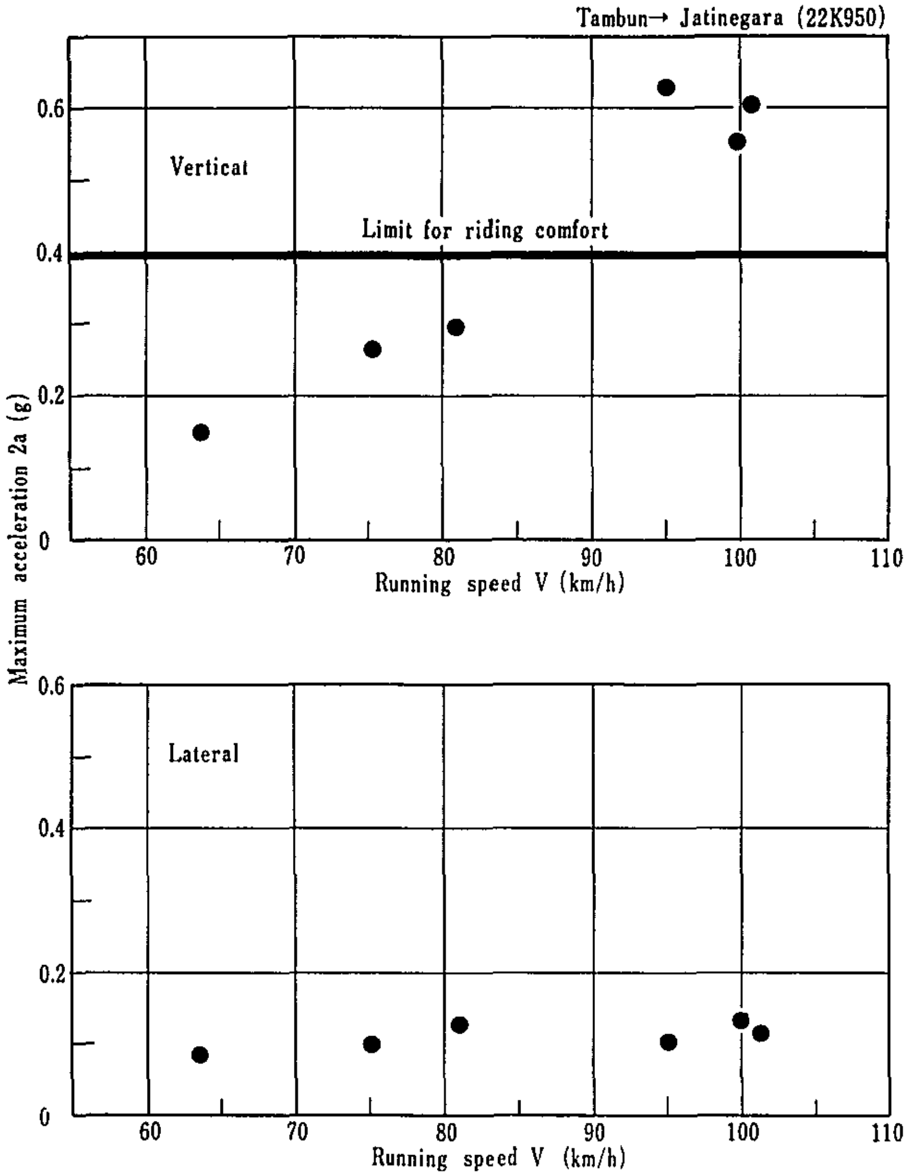


Fig.9-6 Maximum acceleration amplitude (bridge section) of No.3 coach (cw 9257, Görlitz bogie)

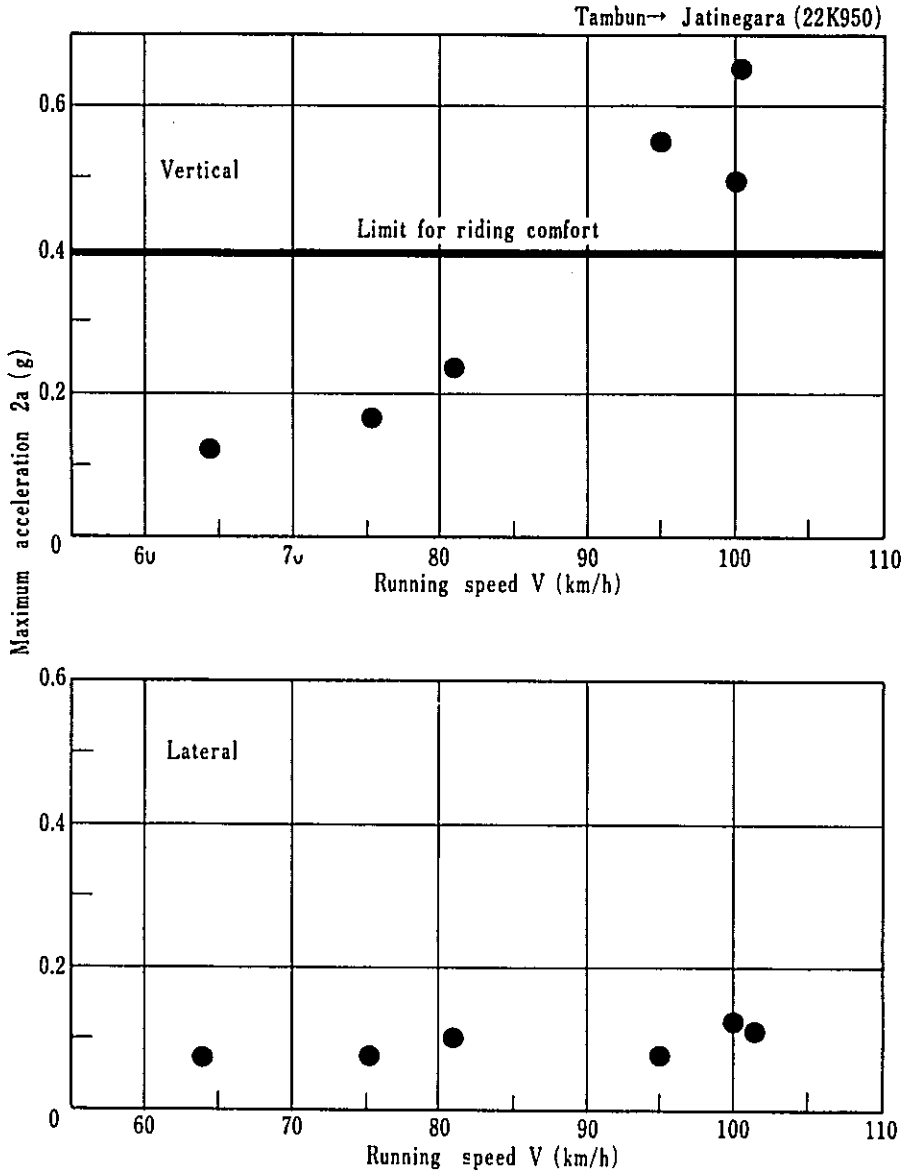


Fig.10-1 Values of vibration acceleration (long-distance test)

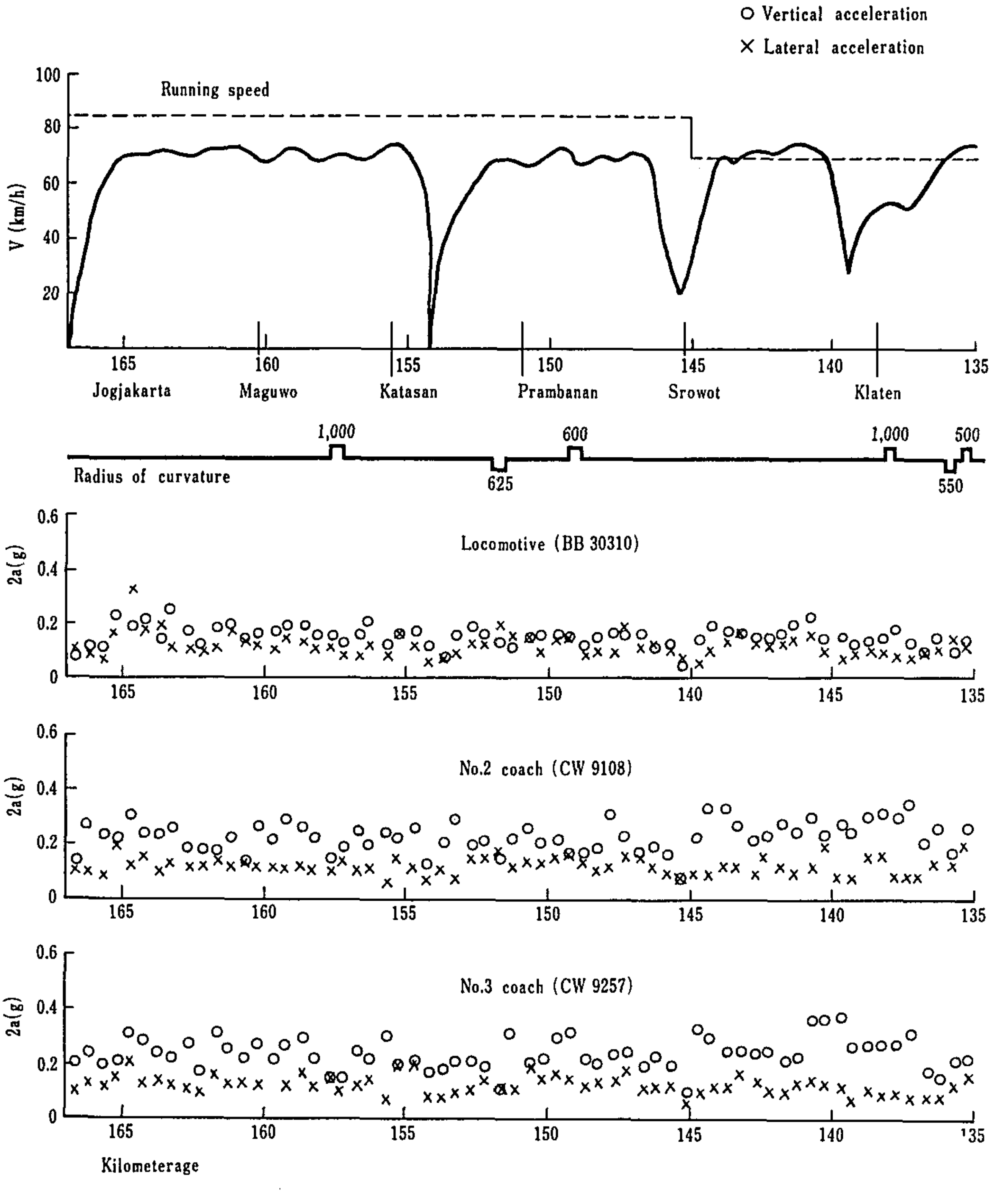


Fig. 10-2 Values of vibration acceleration (long-distance test)

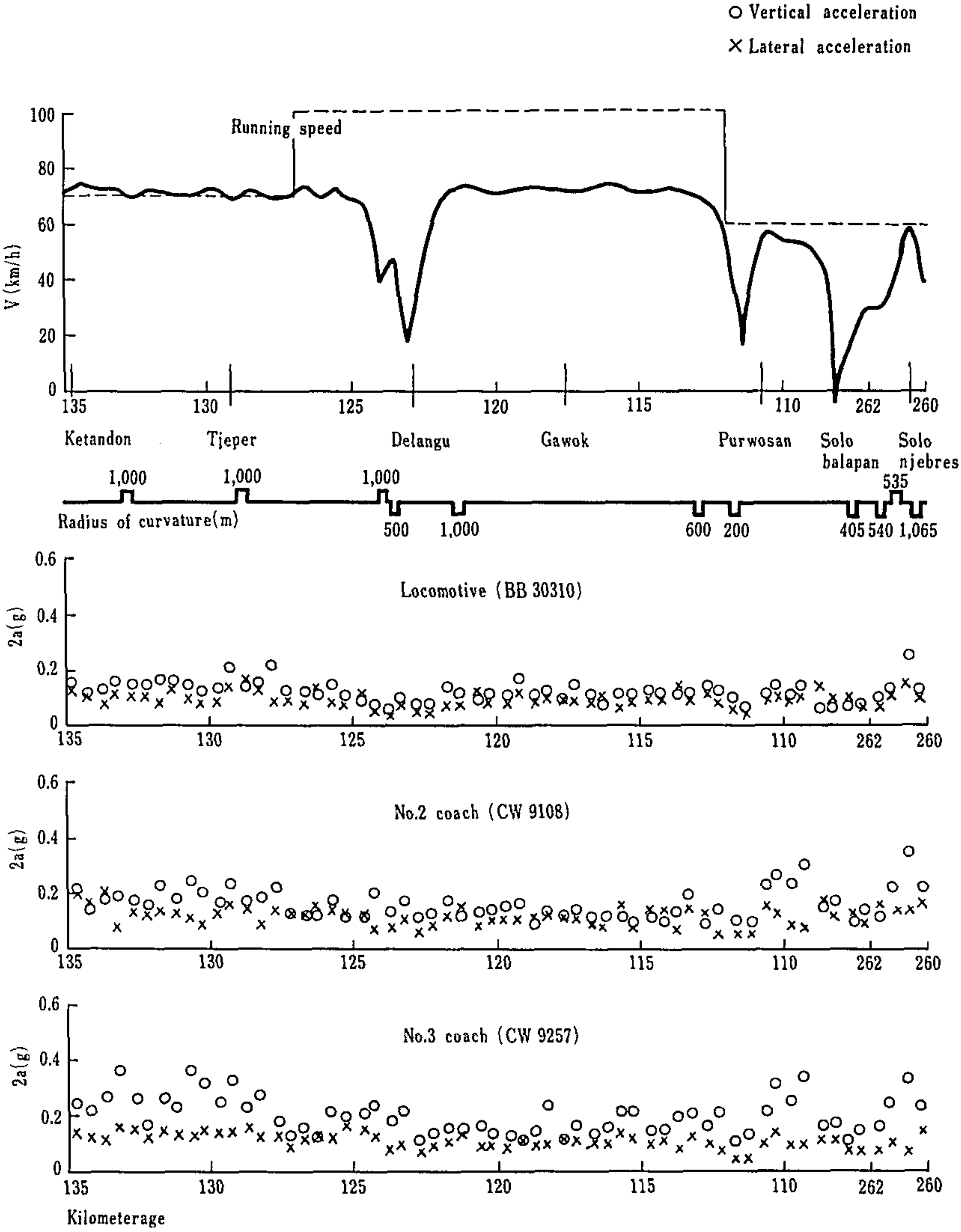


Fig. 10-3 Values of vibration acceleration (long-distance test)

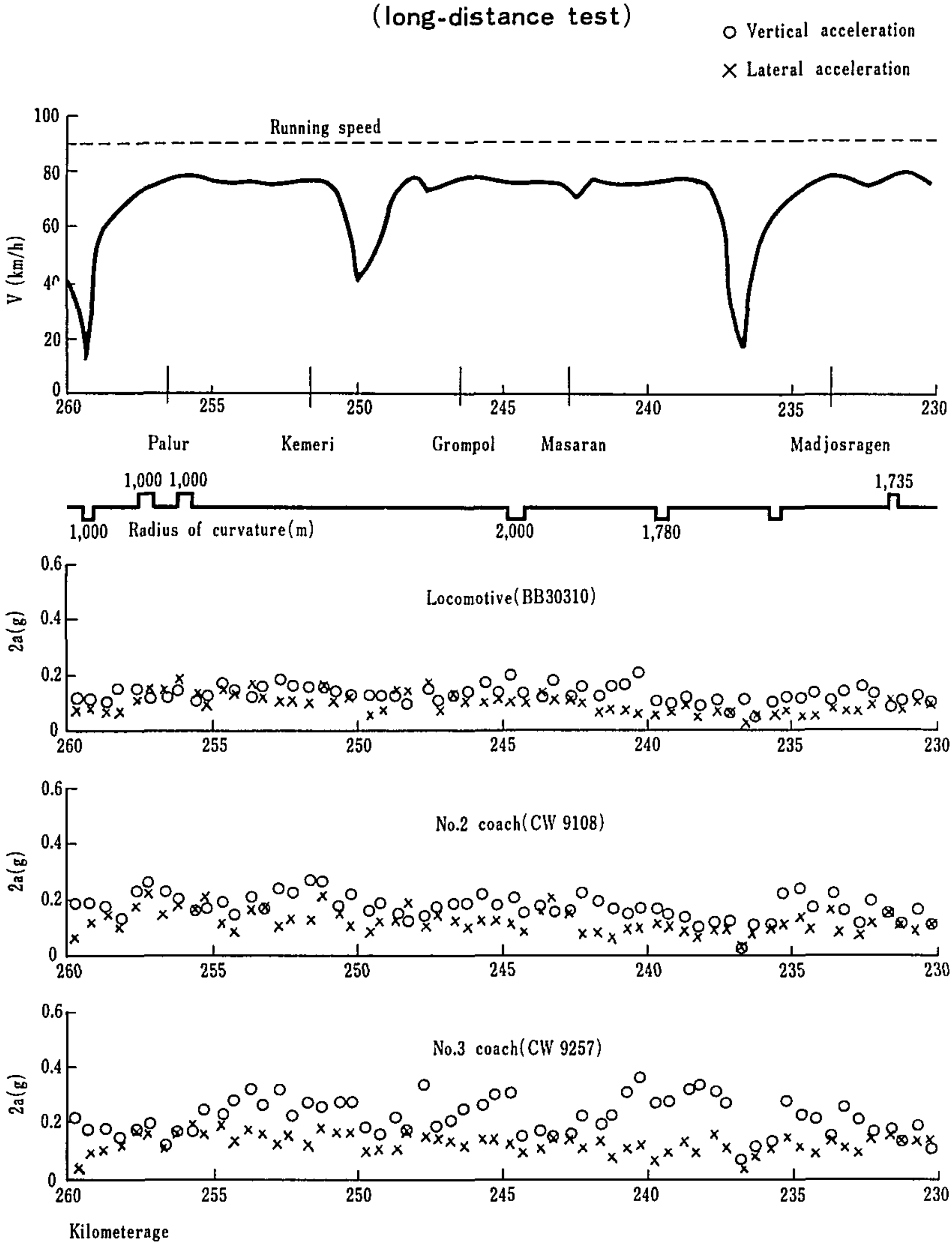


Fig. 10-4 Values of vibration acceleration (long-distance test)

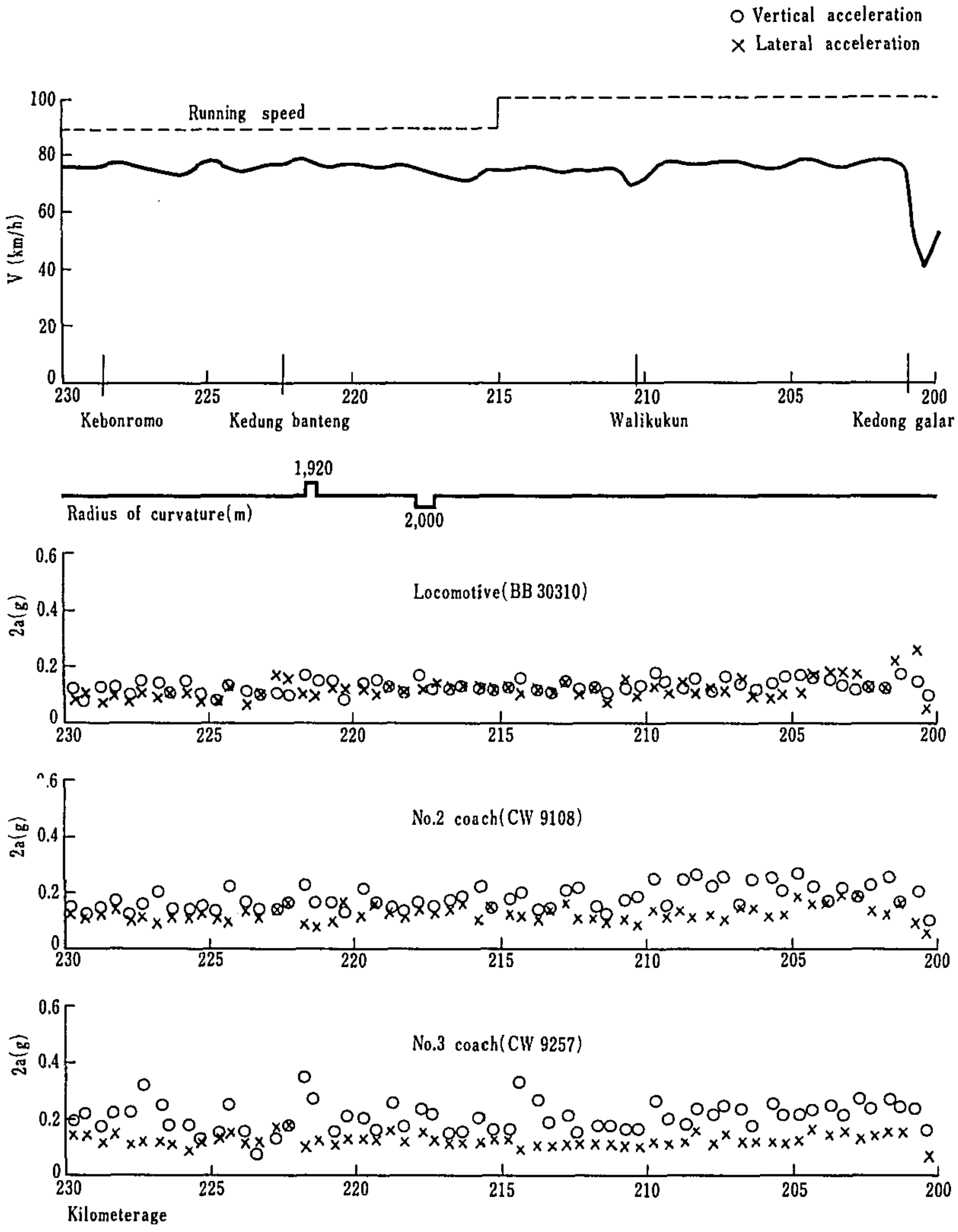


Fig. 10-5 Values of vibration acceleration (long-distance test)

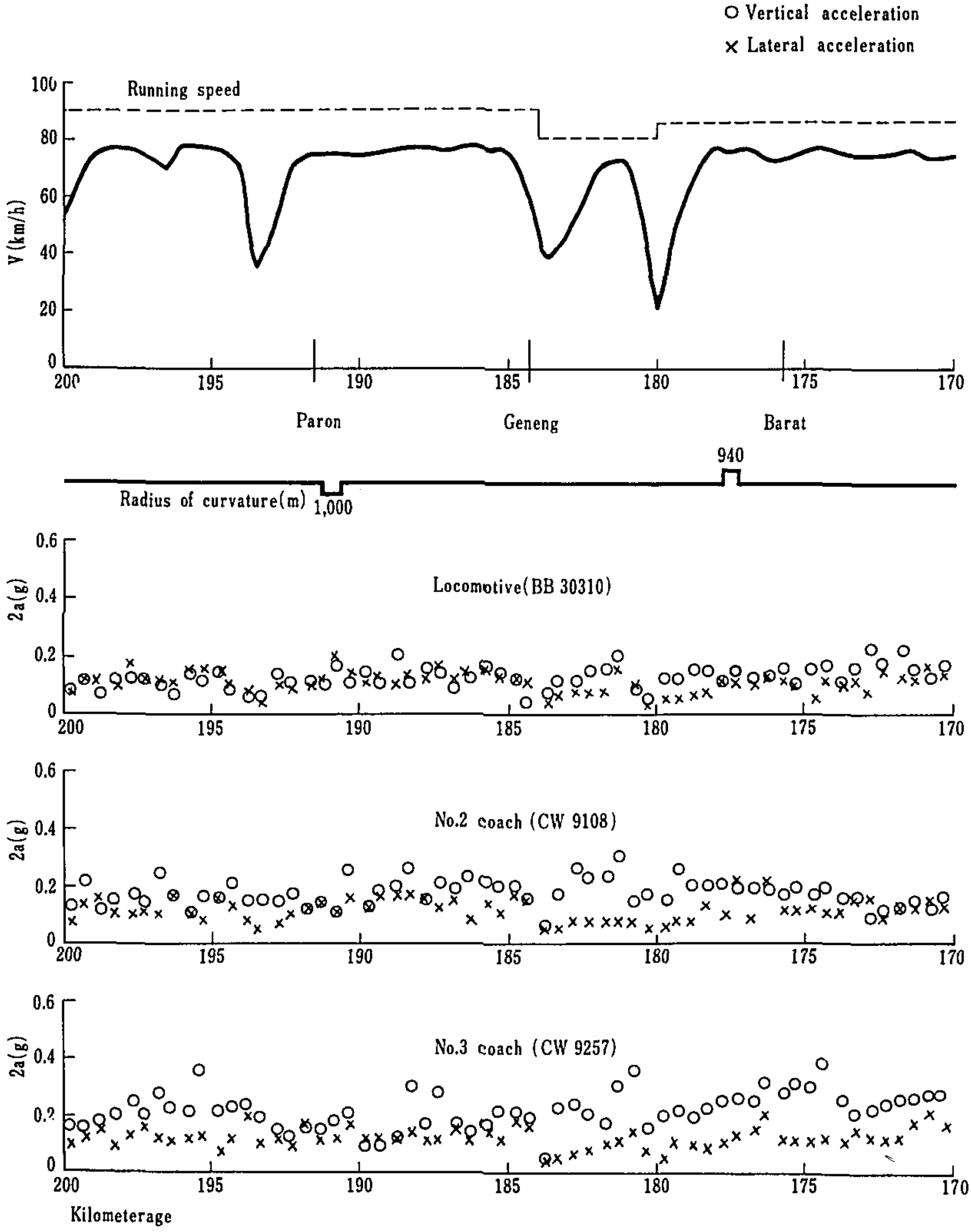


Fig. 10-6 Values of vibration acceleration (long-distance test)

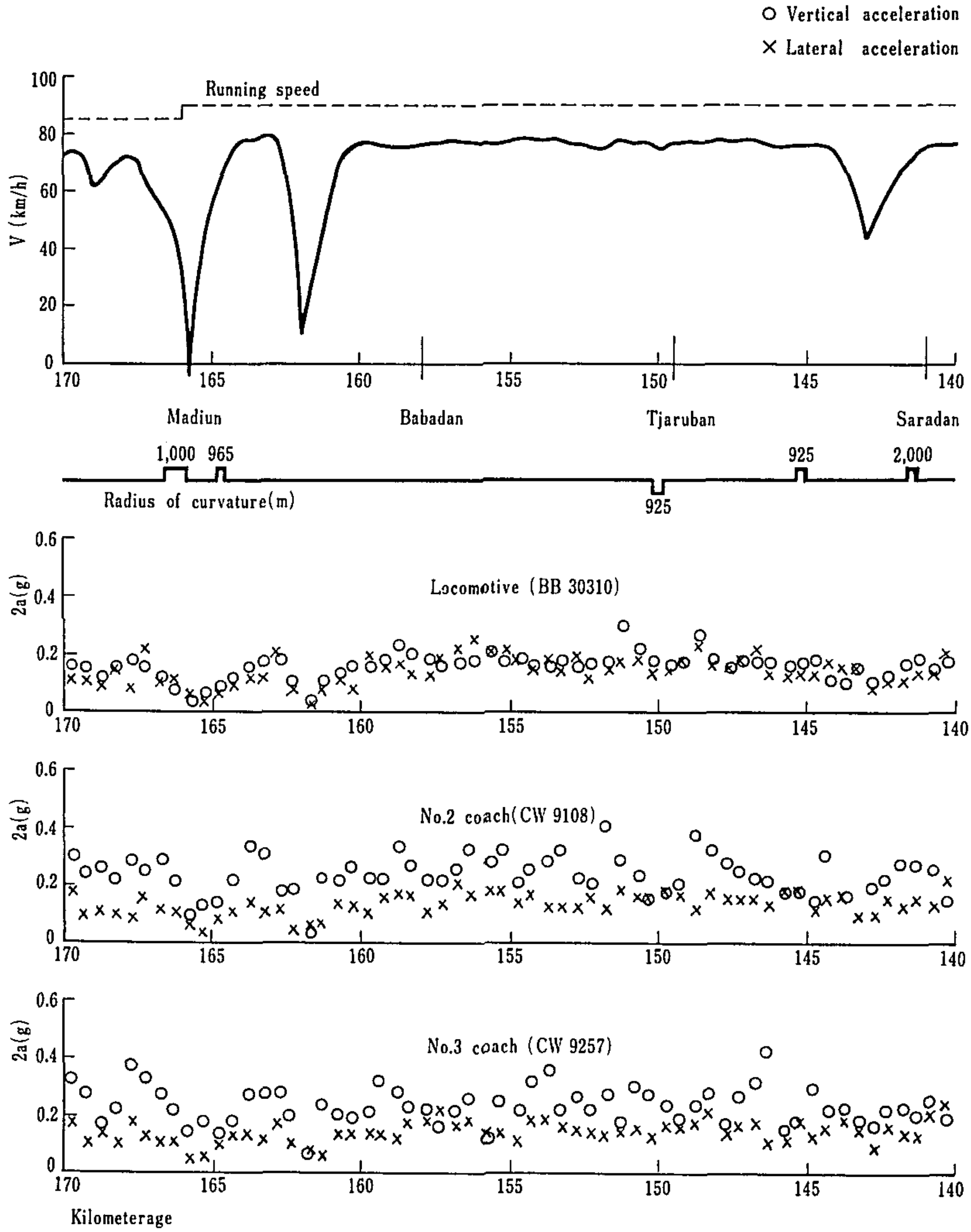


Fig. 10-7 Values of vibration acceleration (long-distance test)

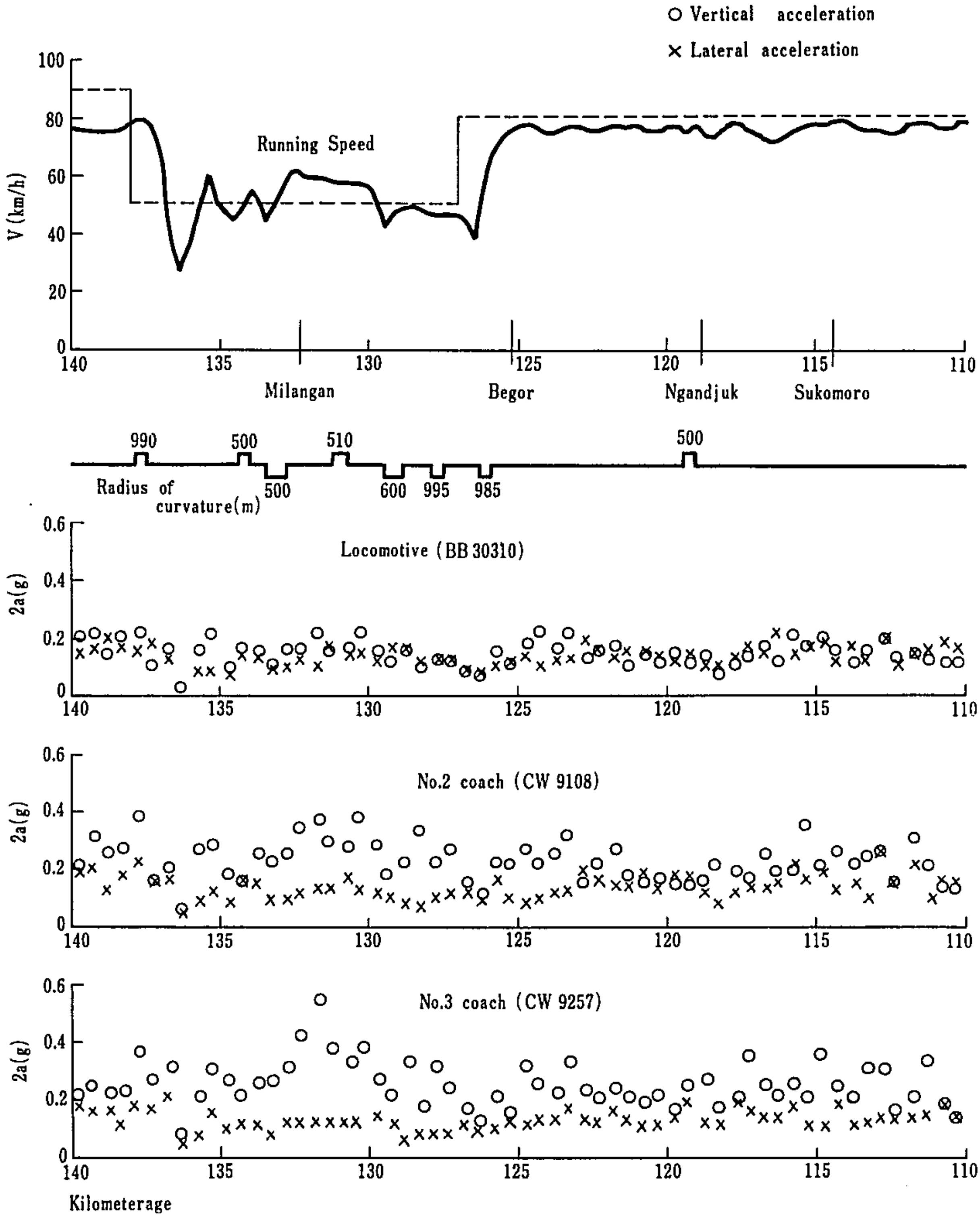


Fig. 10-8 Values of vibration acceleration (long-distance test)

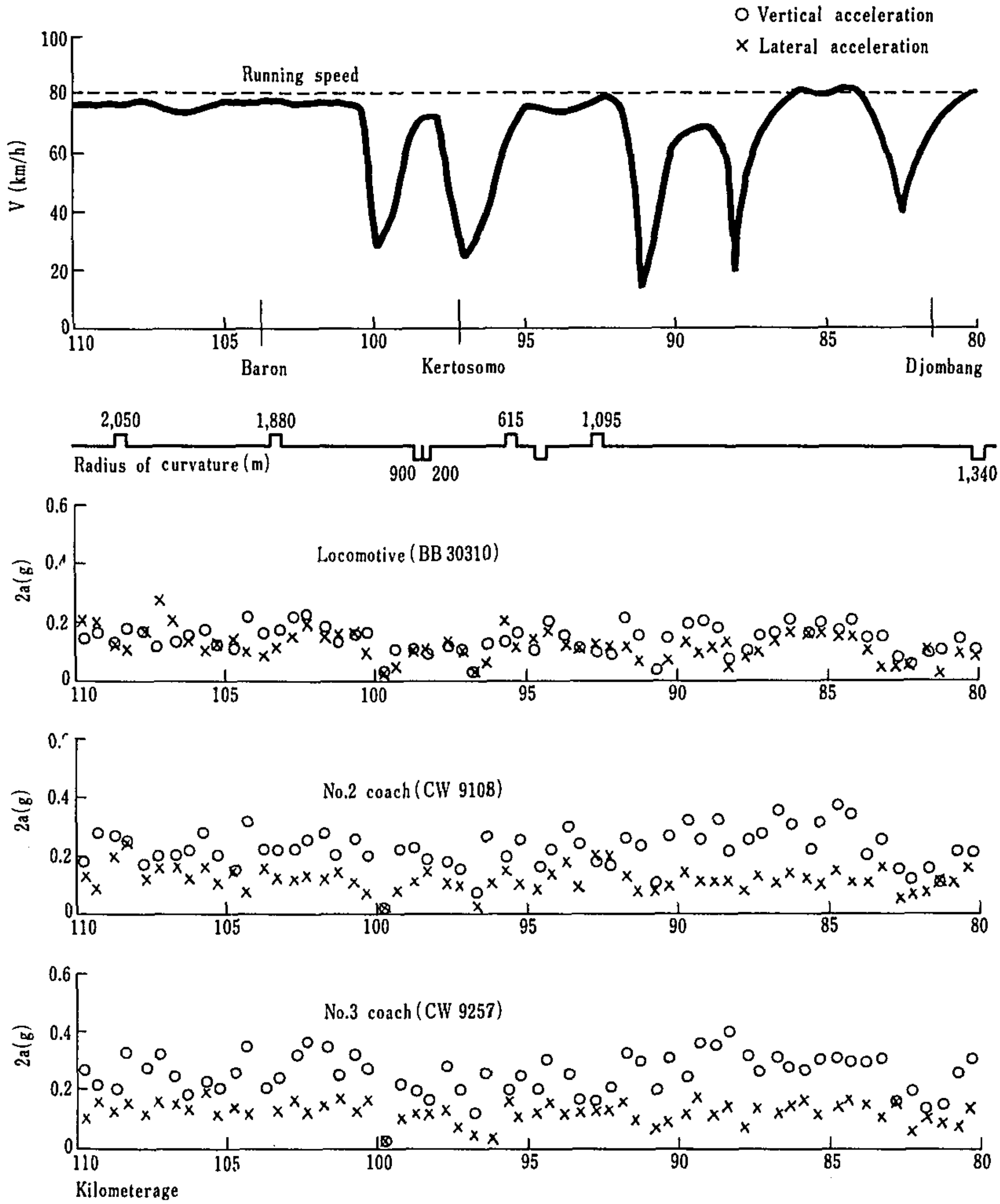


Fig. 10-9 Values of vibration acceleration (long-distance test)

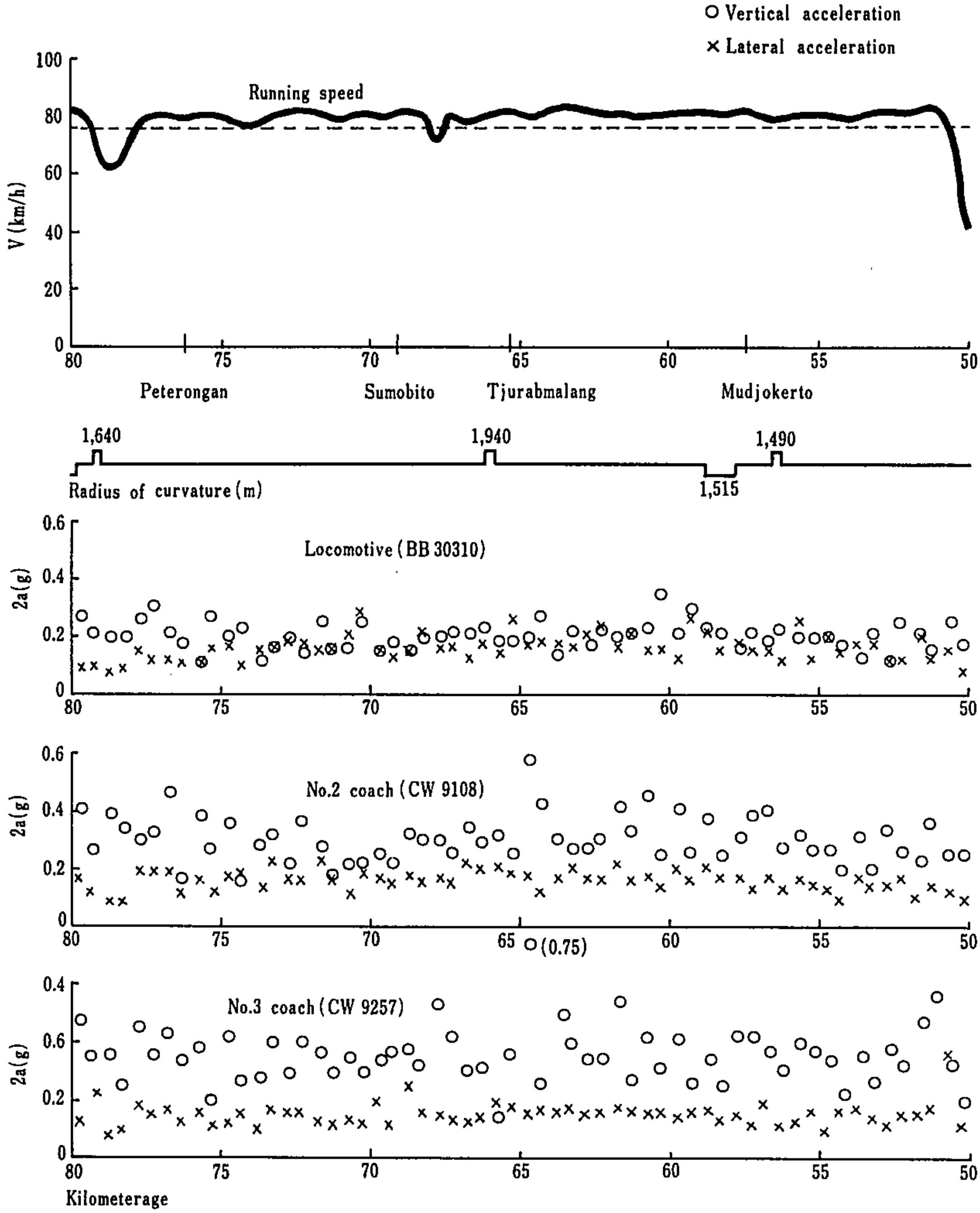


Fig. 10-10 Values of vibration acceleration (long-distance test)

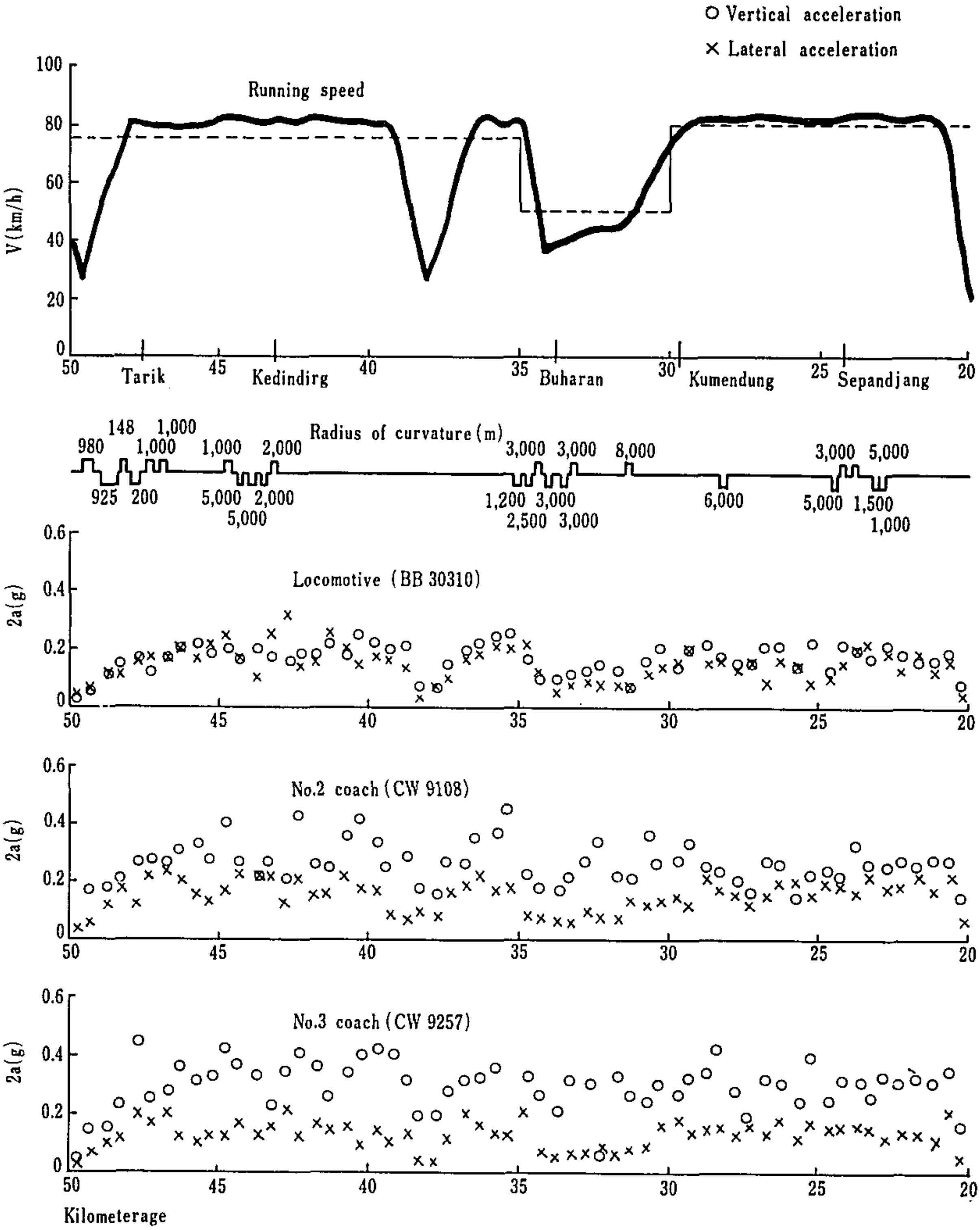
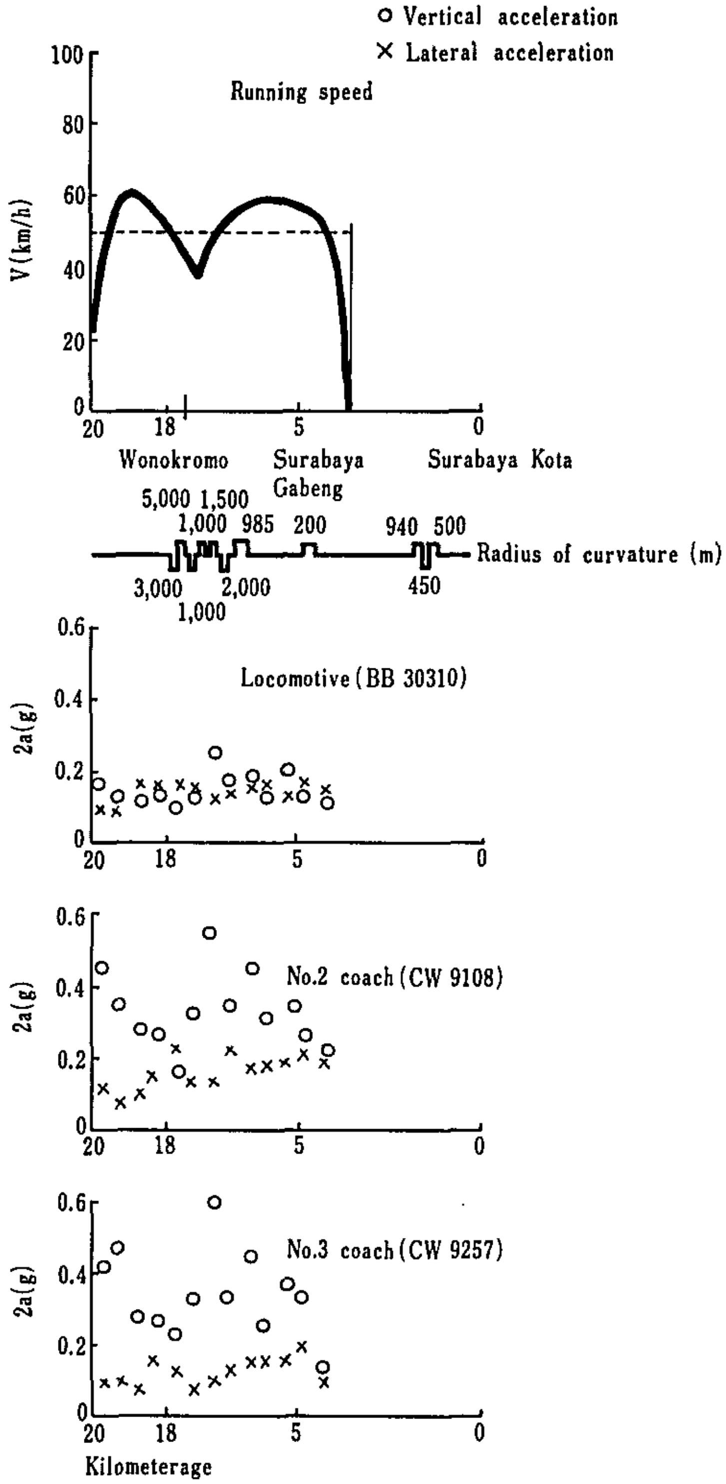


Fig.10-11 Values of vibration acceleration (long-distance test)



acceleration measurement and the track inspection conducted in the long-distance test. The test train ran through these sections at the speed 10 to 20 km/h higher than that for the commercial train (express passenger train) and vibration accelerations were measured in the same manner as in the case of the long-distance test. Fig. 11 shows the maximum vibration accelerations for every 500 m section, together with the corresponding train speeds.

The increase of vibration acceleration is comparatively small on each of the test sections I, II, III and IV, and the maximum is only 0.4 g. But, a tendency is seen that the vibration acceleration over 0.2 g appears more often.

Lateral vibration

Comparing the case where the test train is run at 80 km/h on the schedule-speed-of-70 km/h section with the case where the test train is run at 90 km/h on the schedule-speed-of-80 km/h section, the rate of increase in the vibration acceleration is larger in the latter case. But the vibration acceleration is below 0.4 g for each car on both sections, indicating that the planned increase of train speed can cause no large problem in respect to lateral vibration.

Vertical vibration

The maximum vibration acceleration of each car on every section is below 0.4 g, indicating that, on these sections, the maximum value of vibration may not go beyond 0.5 g against 100 km/h operation, not to speak of the planned train speed-up to 90 km/h. The locomotive shows the least vibration and No. 3 coach the largest. This is the same tendency as that found in the long-distance test.

5.3.4 Vibration analysis

(1) Frequency analysis of vibration acceleration wave form

Fig. 12 shows the frequency analysis, being made in the following way:

(a) Taken up are the frequency and full amplitude of the constant wave, which is considered to appear successively three times at least, out of the measured waves of vibration acceleration.

(b) Circles are drawn in radius of curvature in proportion to 2a, the above full amplitude. In this case, their centres are obtained from the abscissa of velosity and the ordinate of the above frequency.

No. 1 coach was put to the short-distance test only, and so analysis was made based only on this test. But other test cars were analyzed including the results of the long-distance test, too.

Fig. 11-1 Values of vibration acceleration (high-speed test)

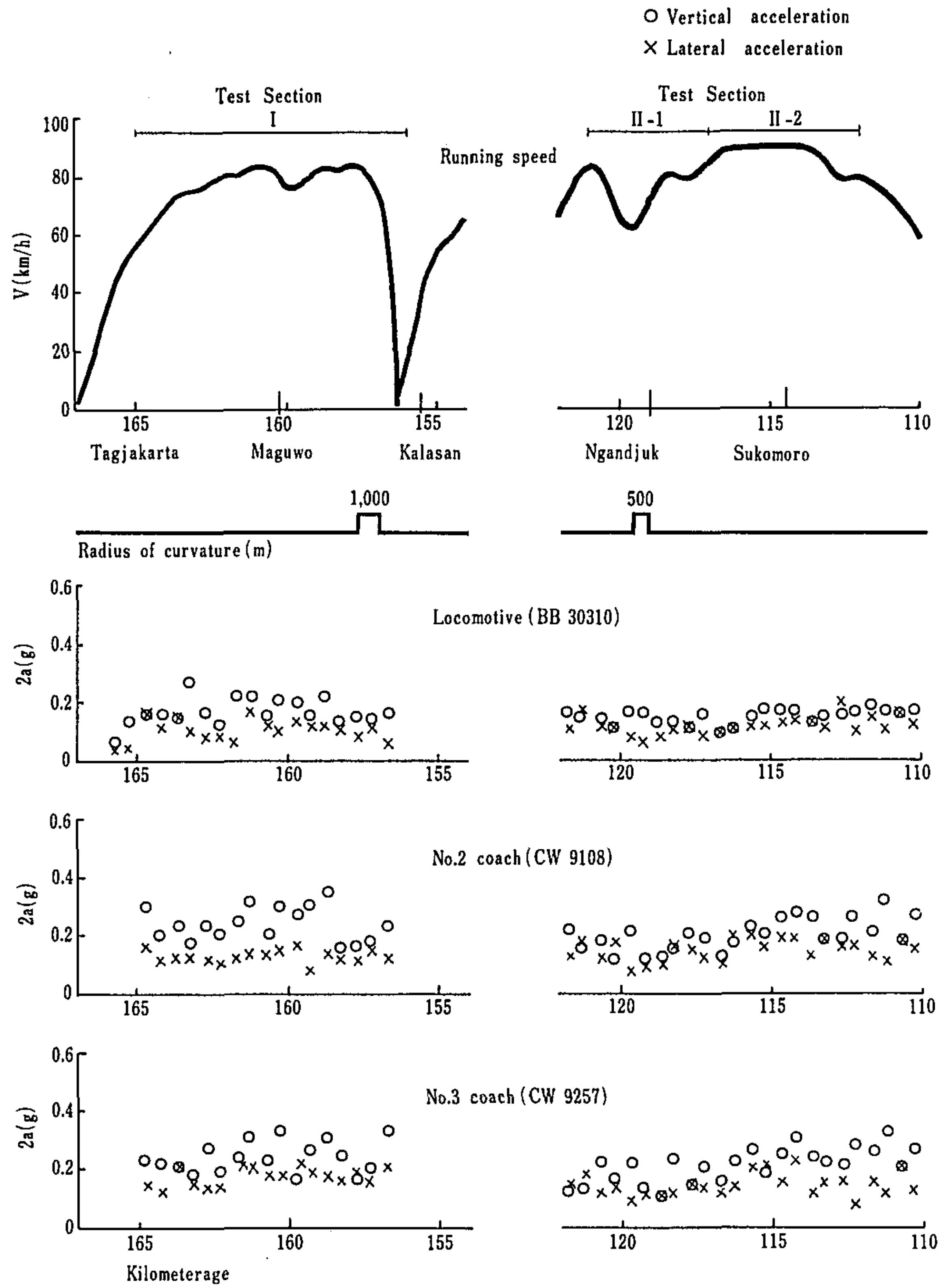


Fig. 11-2 Values of vibration acceleration (high speed test)

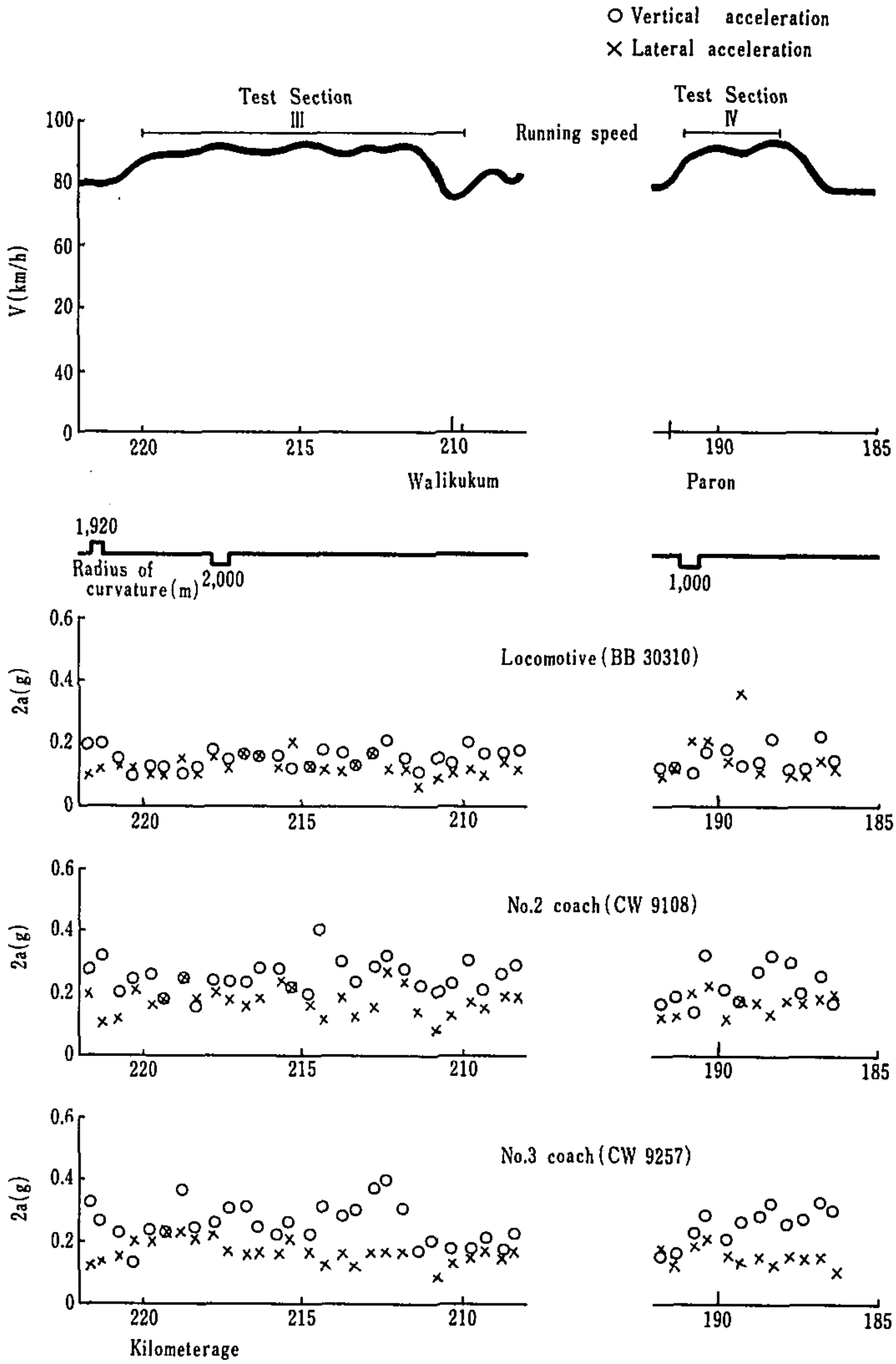


Fig. 12-1 Frequency analysis of No. 1 coach
(CW 9055, Ferrostaal bogie)

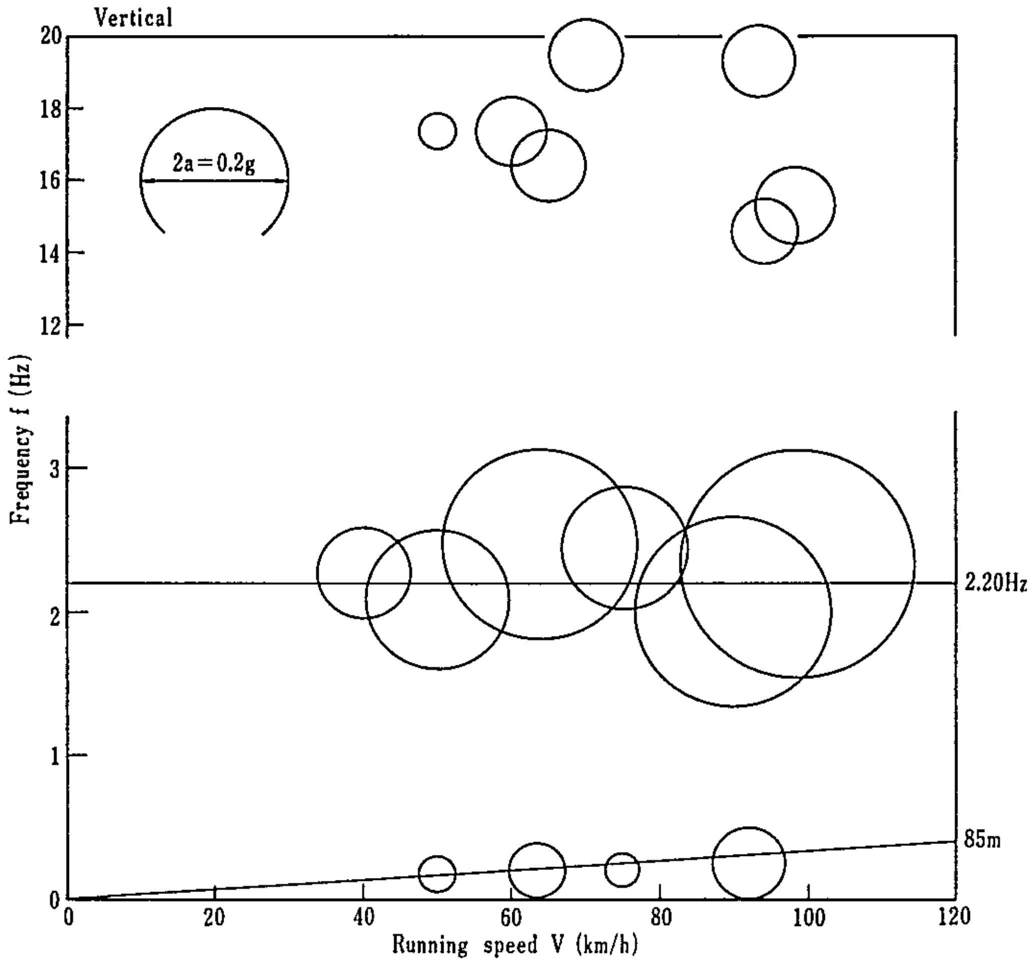


Fig. 12-2 Frequency analysis of No. 1 coach
(CW 9055, Ferrostaal bogie)

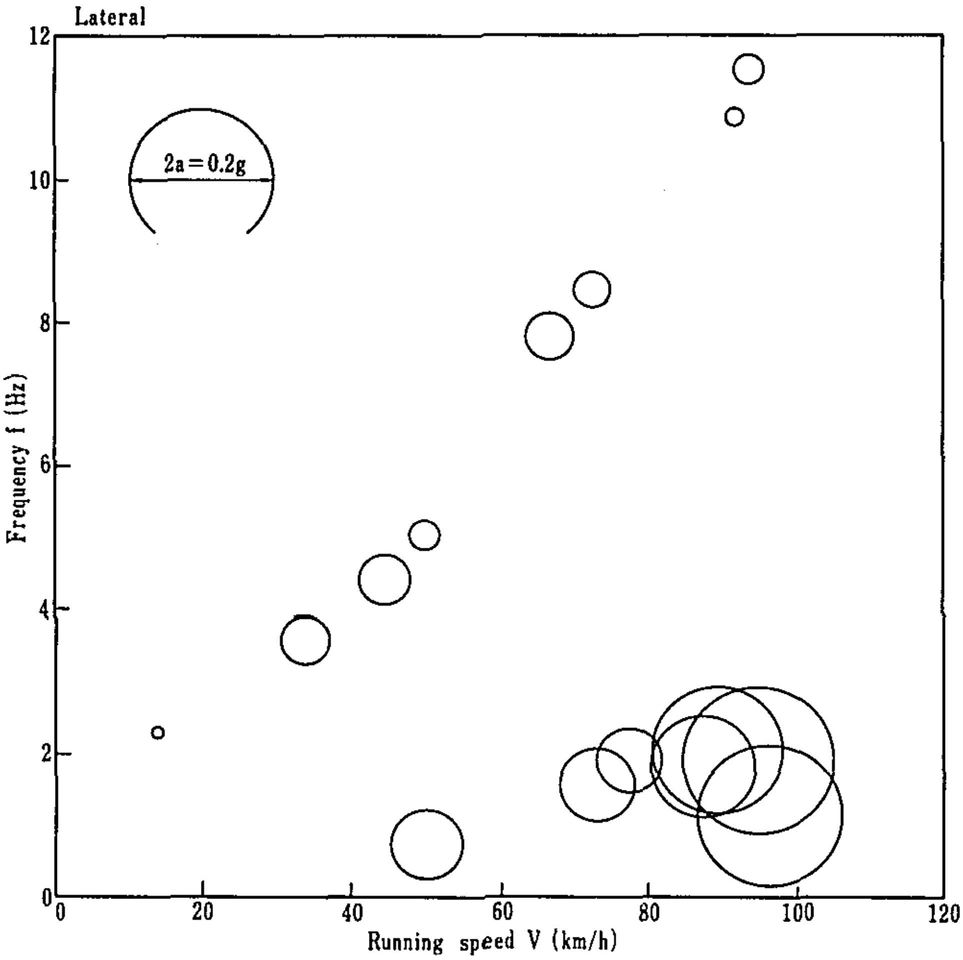


Fig.12-3 Frequency analysis of No.2 coach
(CW 9108, NT-11 bogie)

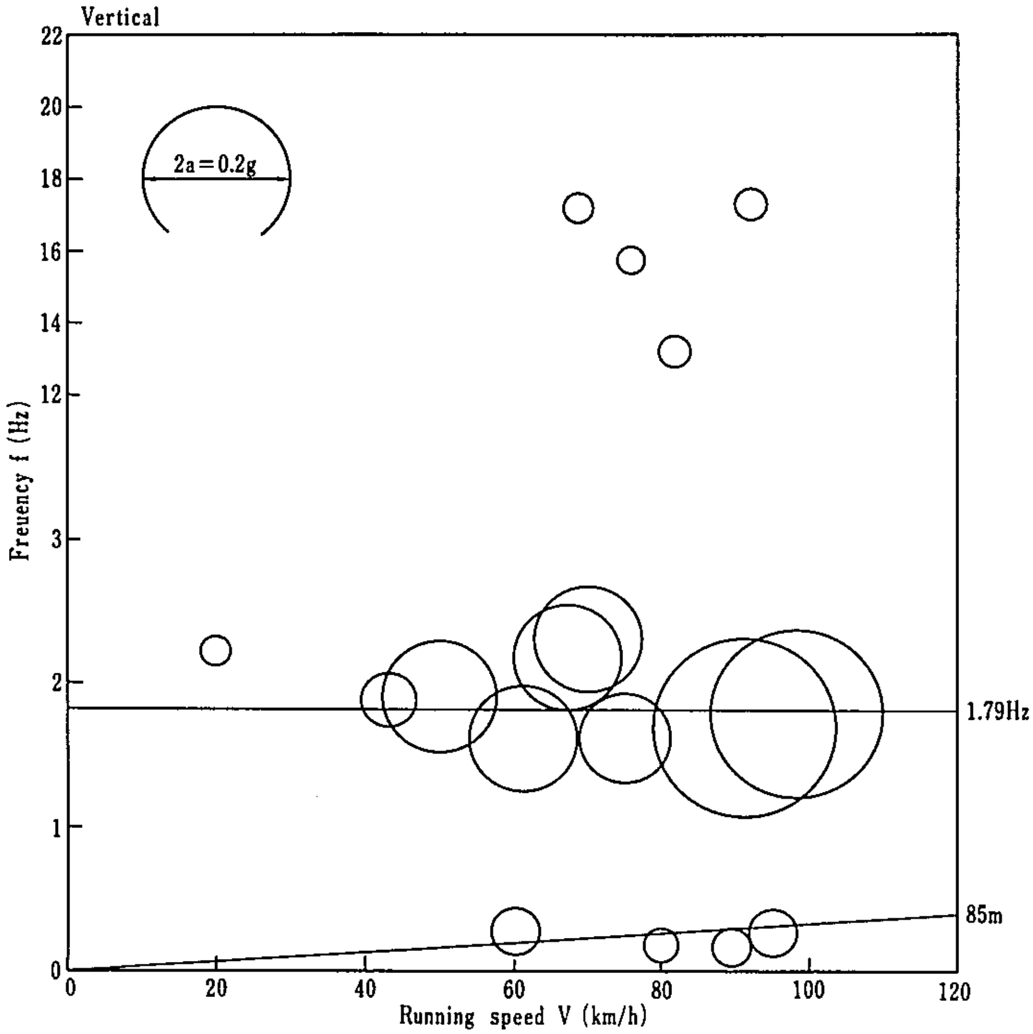


Fig.12-4 Frequency analysis of No.2 coach

(CW 9108,NT-11 bogie)

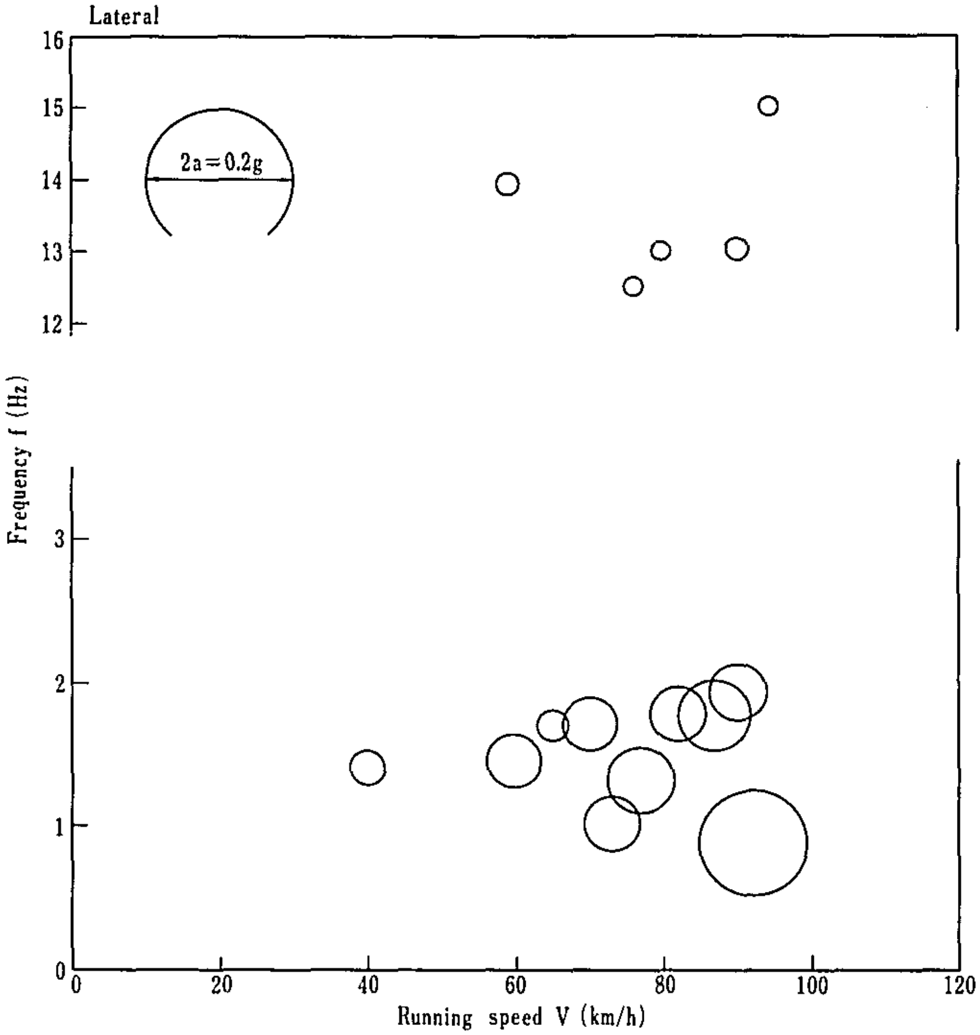


Fig. 12-5 Frequency analysis of No.3 coach (CW 9257, Görlitz bogie)

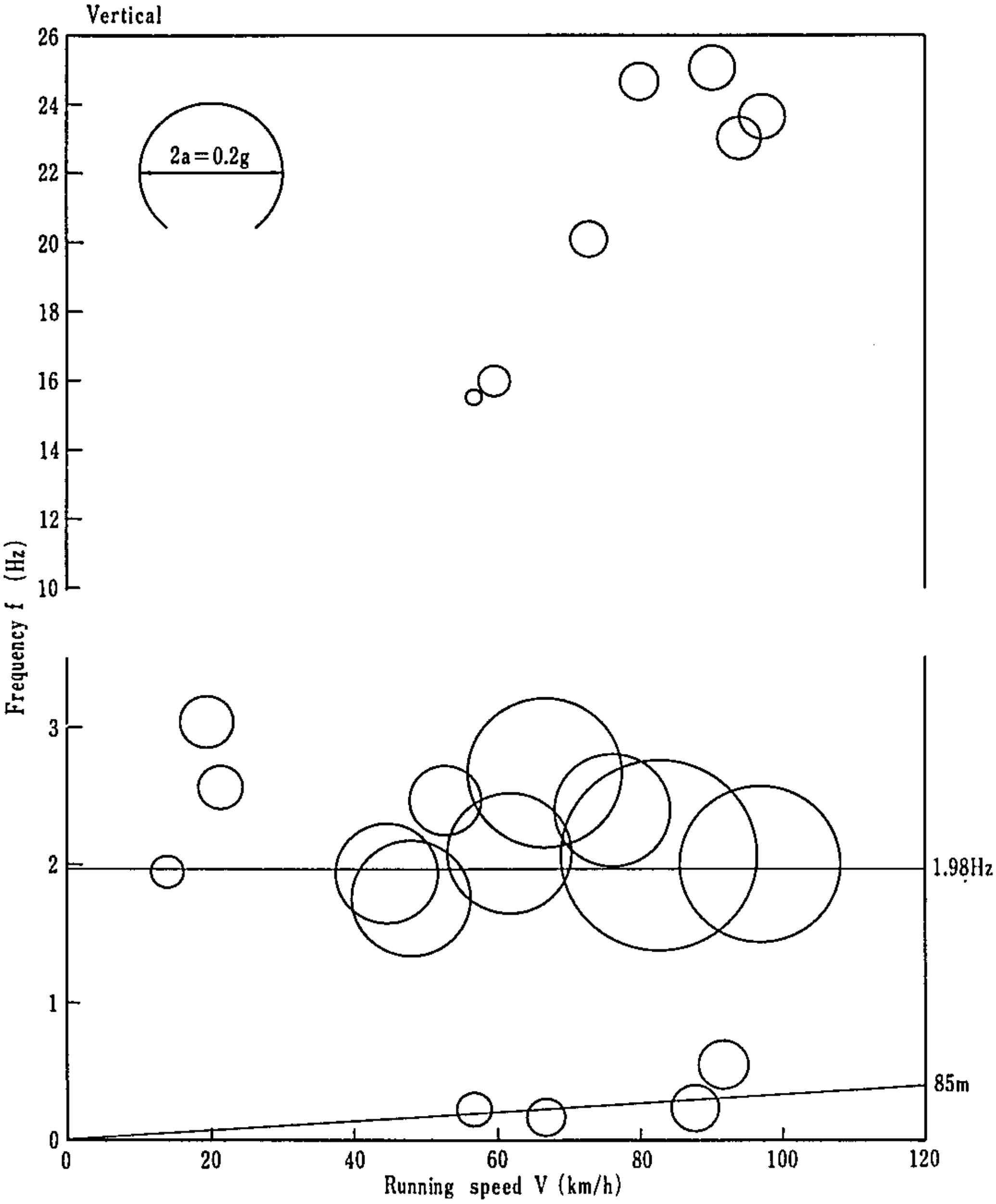
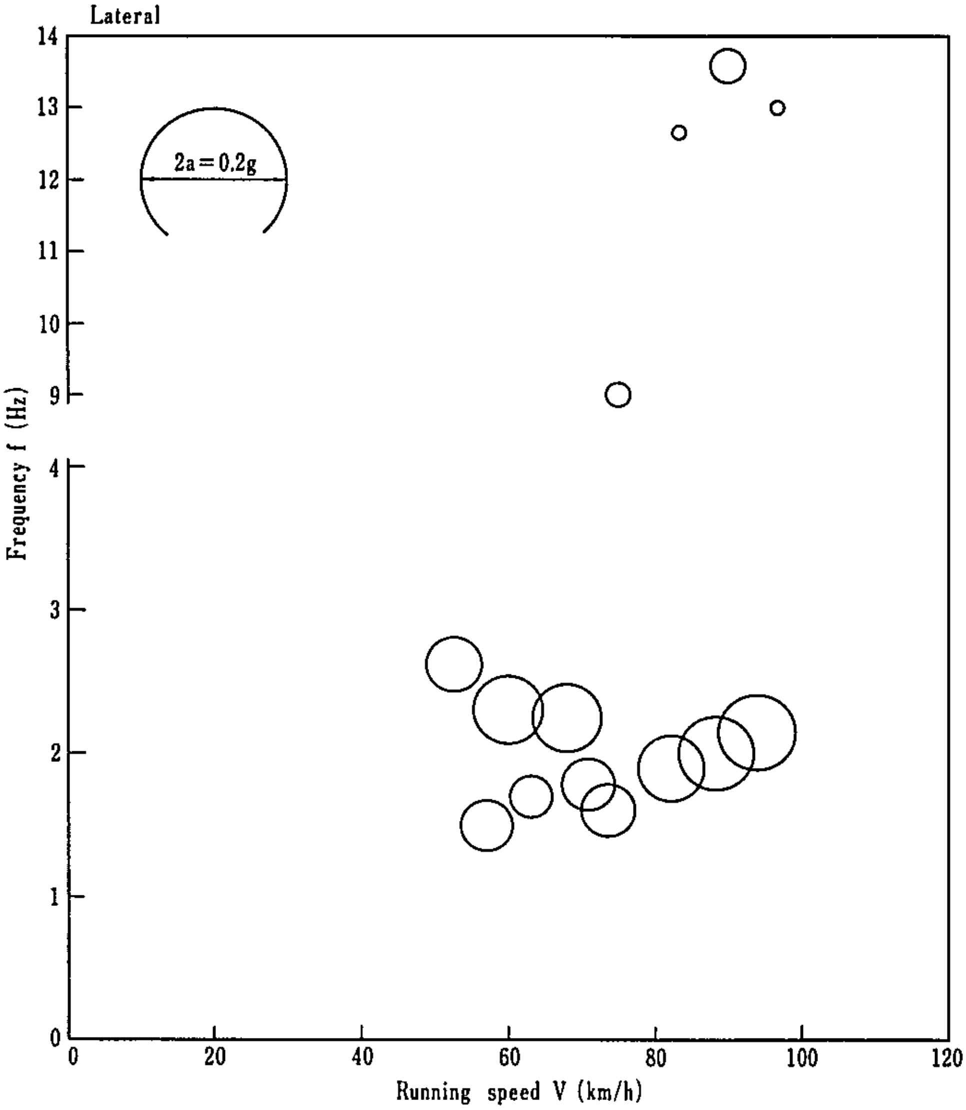


Fig. 12-6 Frequency analysis of No.3 coach

(CW 9257, Görlitz bogie)



In each of these figures, when circles are located in line on the oblique line going through the origin, it shows that that vibration is generated forced by an external force of a certain constant wave-length, and the wave length can be obtained from the inclination against the axis of abscissa. When circles are located in line parallel to the axis of abscissa, it indicates that there is generated the natural vibration of the rolling stock. Therefore, the natural frequency can be obtained from the figure (for detail, see APPENDIX IV).

Lateral vibration

High-frequency vibrations are being generated in No. 1 coach, proportional to the rotation of its wheels. Few other high-frequency vibrations of constant nature are noticed. In addition, their acceleration amplitudes are so small as to cause no trouble.

In each car, vibrations due to external force of 13–14 m are noticed occurring. This is considered to be due to the snake motion of truck caused by external force. The natural frequency in the lateral direction due to the carbody supporting device (spring and swing bolster device of the truck) is 1.2 – 2.0 Hz. In the case of No. 1 coach, the carbody is nearly in a resonance condition at speeds near 90 – 100 km/h, while in the case of No. 2 and No. 3 coaches, such a resonance condition is not observed.

Vertical vibration

Forced vibration of wave-length about 85 m (equivalent to the length of one piece of rail) is noticed in each passenger car. In addition, forced vibration of wave length 1 – 2 m is superposed in No. 3 coach. The generation of these vibrations are considered to be due to the settlement of rail joints for the former vibration and due to the eccentricity of wheels for the latter.

The vertical natural frequency of the carbody due to the spring device of the truck is about 2.2 Hz for No. 1 coach, about 1.7 Hz for No. 2 coach and about 2 Hz for No. 3 coach, and these cars are in resonance condition at 80 km/h – 100 km/h. For each car, high-frequency vibration hardly occurs constantly, and, if there is, it is small in amplitude.

(2) Cumulative frequency of vibration accelerations

Fig. 13 shows the relation between the cumulative number of vibration accelerations (per km) and the magnitude of vibration accelerations, which is obtained by taking, out of the vibration accelerations measured in the long-distance test, the maximum value of vibration accelerations for each 500 m section of the section run at

Fig.13-1 Number of occurrence of diesel locomotive (BB 30310)

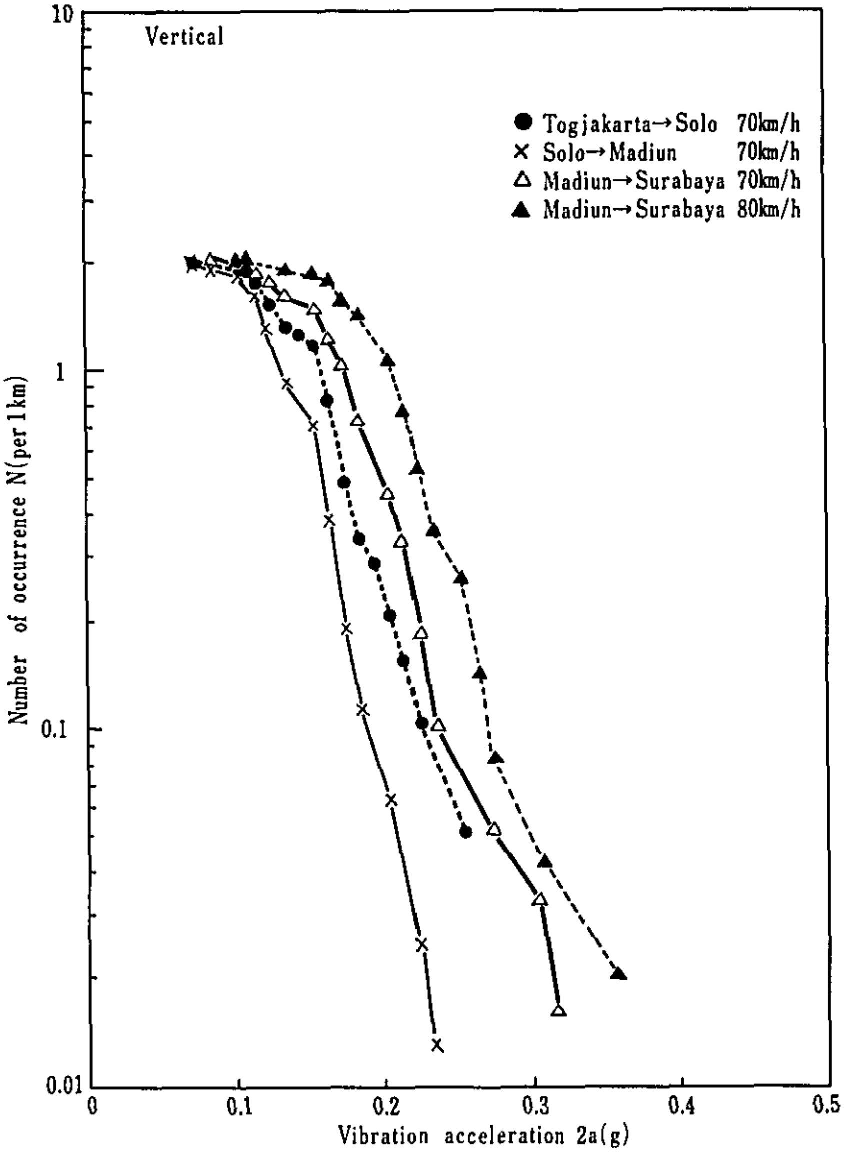


Fig.13-2 Number of occurrence of diesel locomotive (BB 30310)

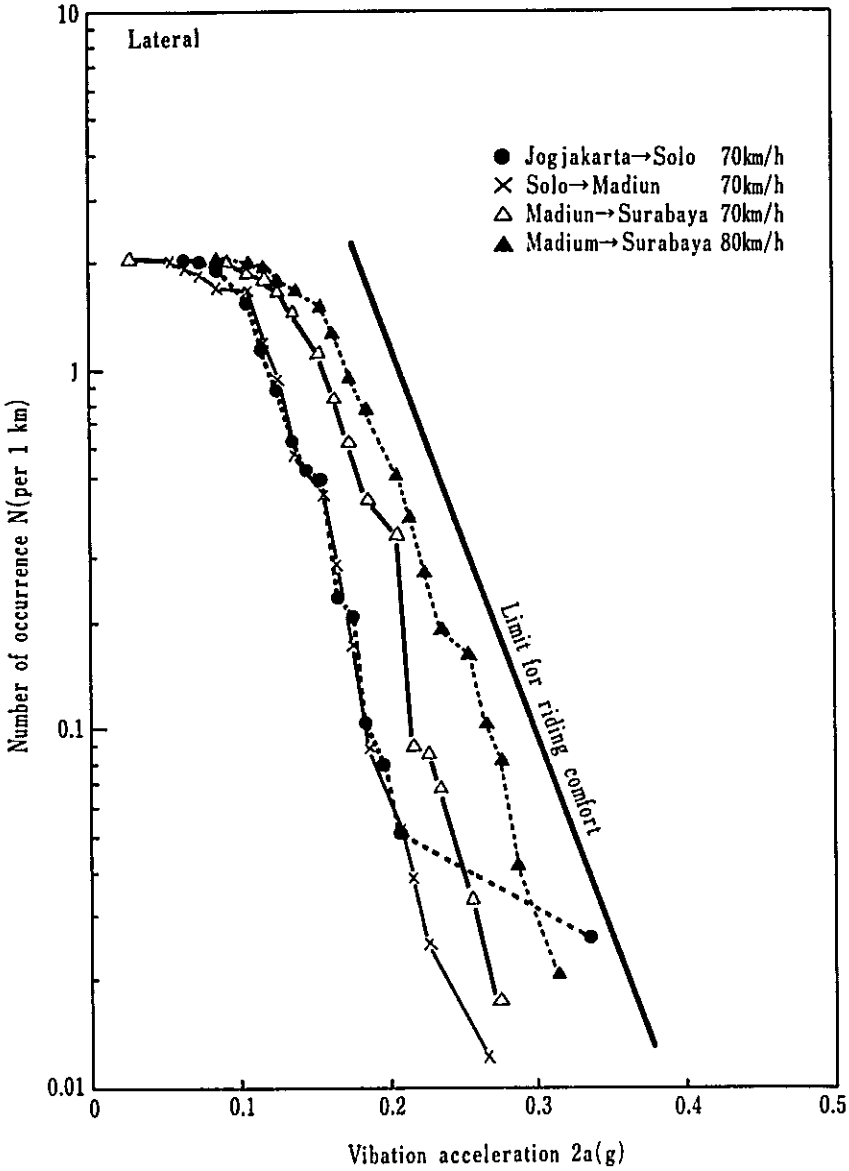


Fig. 13-3 Number of occurrence of No.2 coach
(CW 9108, NT-11 bogie)

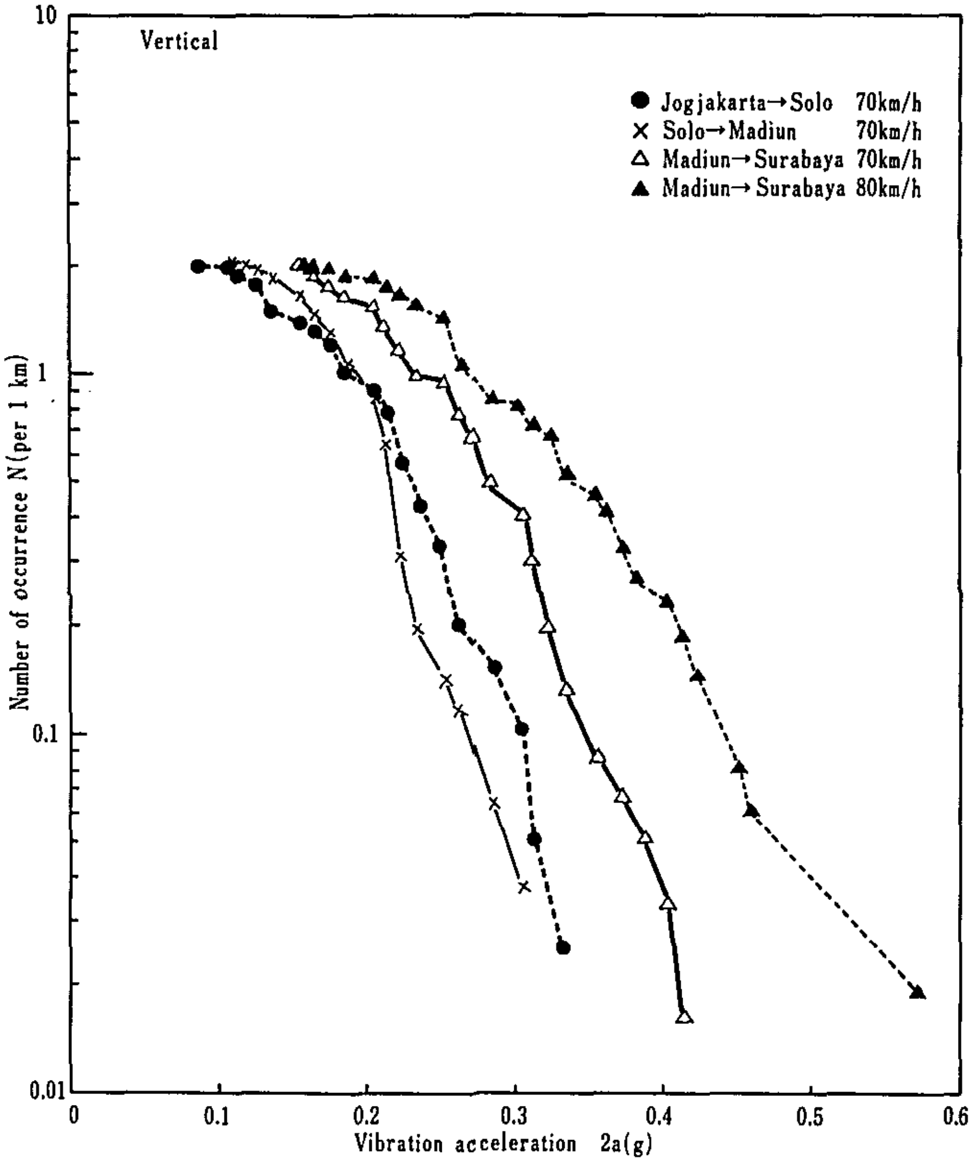


Fig. 13-4 Number of occurrence of No.2 coach (CW 9108, NT-11 bogie)

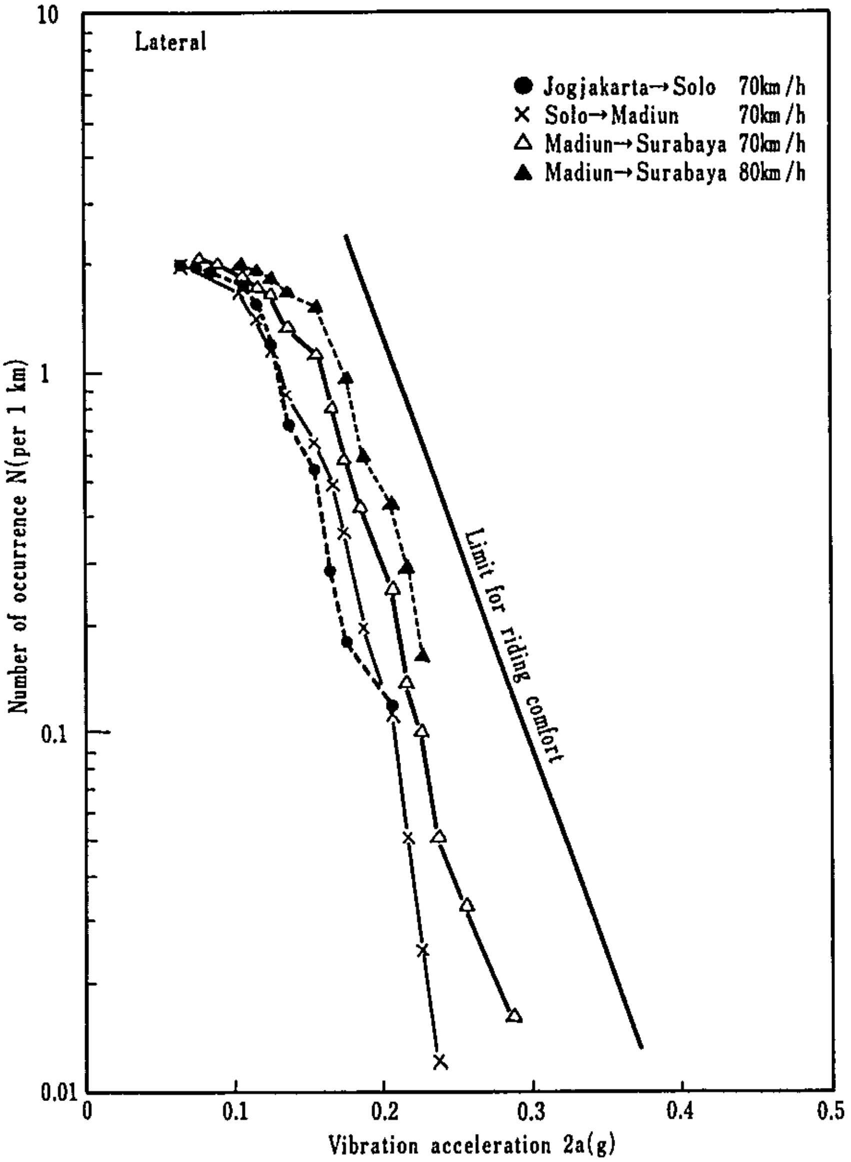


Fig.13-5 Number of occurrence of No.3 coach
(CW 9257, Görlitz bogie)

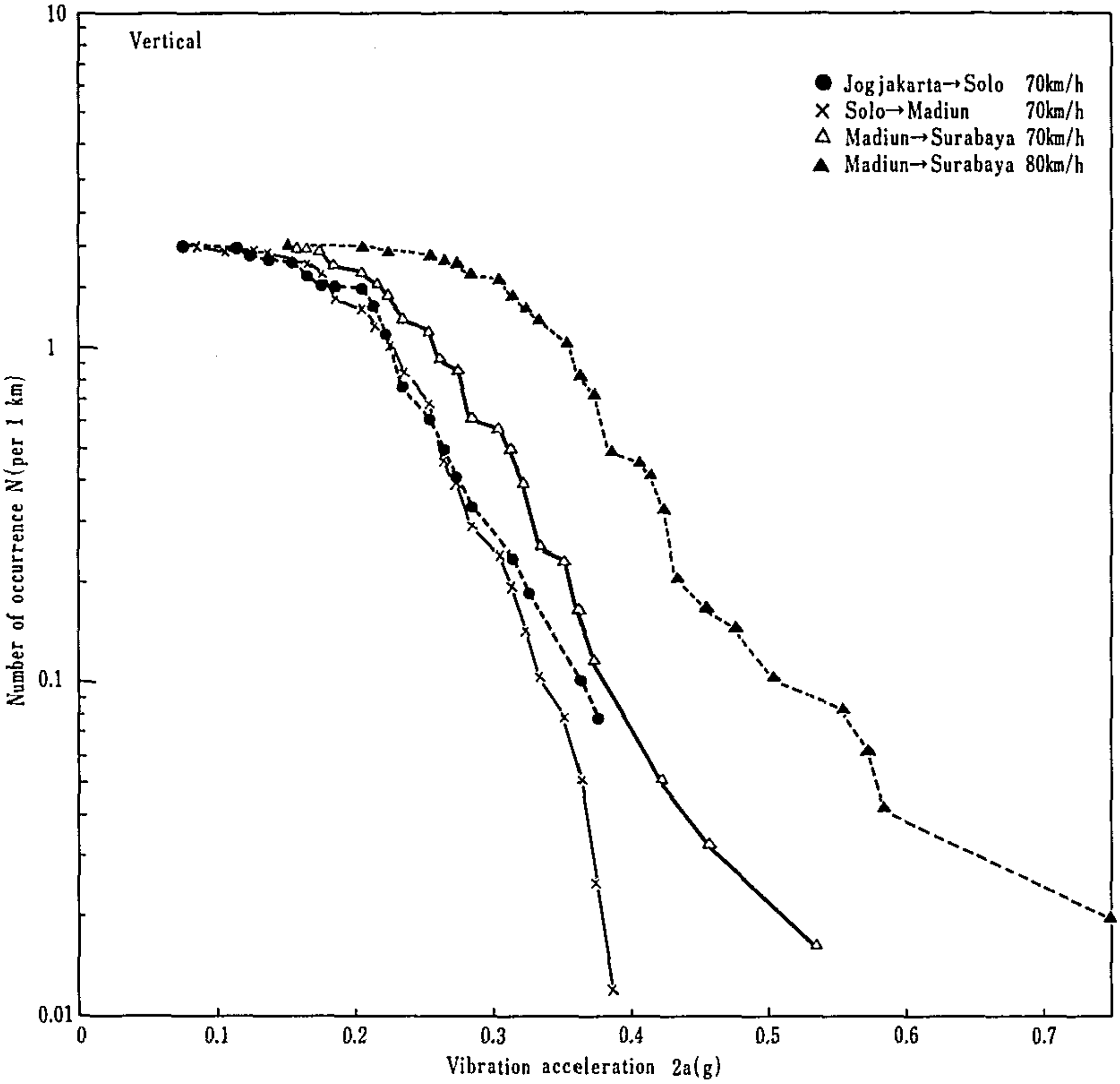
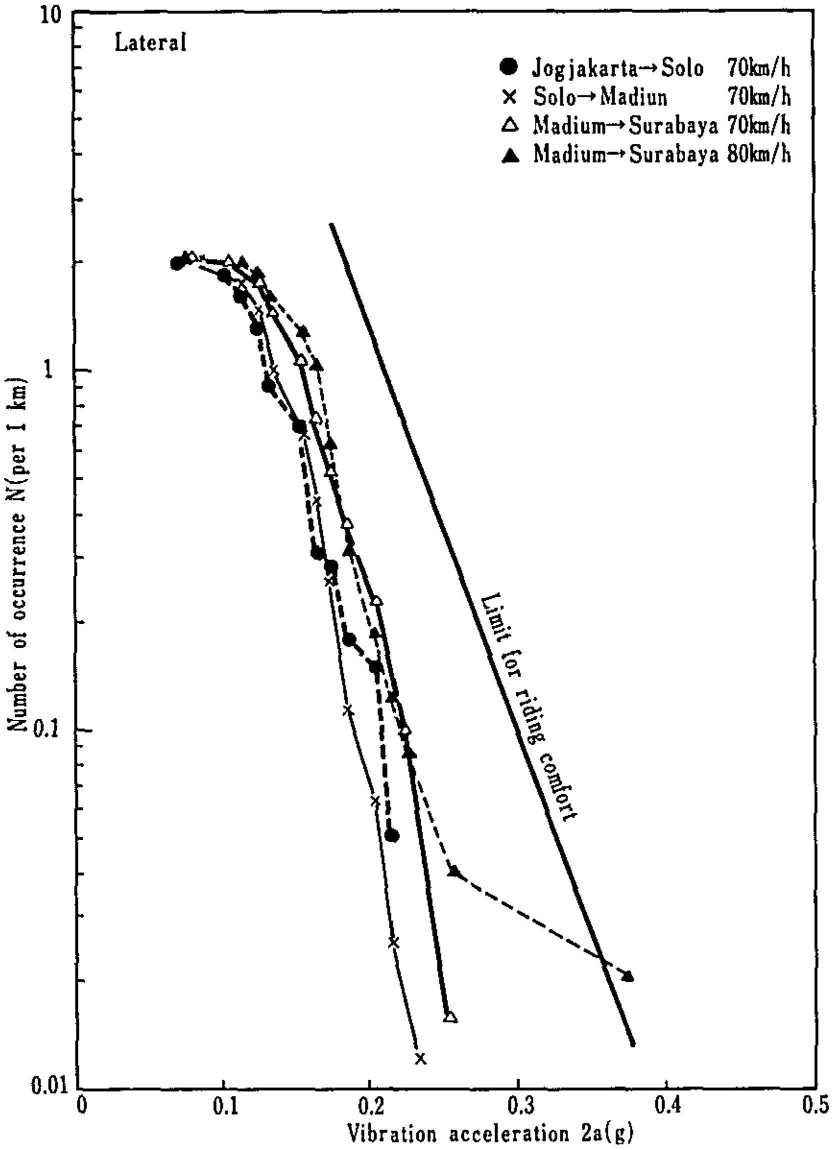


Fig.13-6 Number of occurrence of
No.3 coach (CW 9257, Görlitz bogie)



70 km/h–84 km/h, and classifying the maximum values by magnitude (0.01 g order) to obtain the number of vibration accelerations by each magnitude. For each car, the test section is classified into the Jogjakarta–Solo section, the Solo–Madiun section and the Madiun–Surabaya section, and, especially, for the Madiun–Surabaya section, the case run at 70 km/h and the case run at 80 km/h are shown to make it easy to compare with the vibration occurring condition in the other sections.

Lateral vibration

Any of the test cars did not exceed the riding comfort limit fixed by JNR. In the comparison by test section, the Solo–Madiun section is best and the Madiun–Surabaya section worst. As for the difference between at running speeds of 70 km/h and 80 km/h, No. 3 coach shows the least difference.

Viewing from these results, it can be said that any problem does not exist in increasing the train speed as to lateral vibration.

Vertical vibration

The locomotive is comparatively free from this type of vibration. But No. 2 and No. 3 coaches show generation of vibration acceleration exceeding 0.5 g when the test train ran at 80 km/h between Madiun and Surabaya, indicating that even the existing scheduled maximum speed of 80 km/h for this section seems problematical. As can be seen from Fig. 13, it is preferable to lower the train speed for this section to below 80 km/h. To realize 100 km/h train operation on this section, it seems indispensable, first of all, to better the present track maintenance condition, judging from the results of the short-distance test and the high-speed test.

(3) Riding comfort

By reading the amplitudes and frequencies of vibration at the points where vibrations are occurring conspicuously and plotting the frequencies on the abscissa and the amplitudes (half amplitude in this case) on the ordinate, the kinds of vibrations mostly occurring on the car can be known, and, also from this figure, it becomes possible to express the riding comfort of the car quantitatively (for detail, see APPENDIX IV).

Fig. 14 shows the results of the short-distance and long-distance tests at speeds over 70 km/h.

Each of the polygonal lines representing the riding-comfort indices 1, 2, 3 given in the figure shows the same level of riding comfort. In JNR, the following riding-comfort index is established to judge the riding comfort of the coaches on the SHINKANSEN:

Fig.14-1 Riding comfort of diesel locomotive (BB 30310)

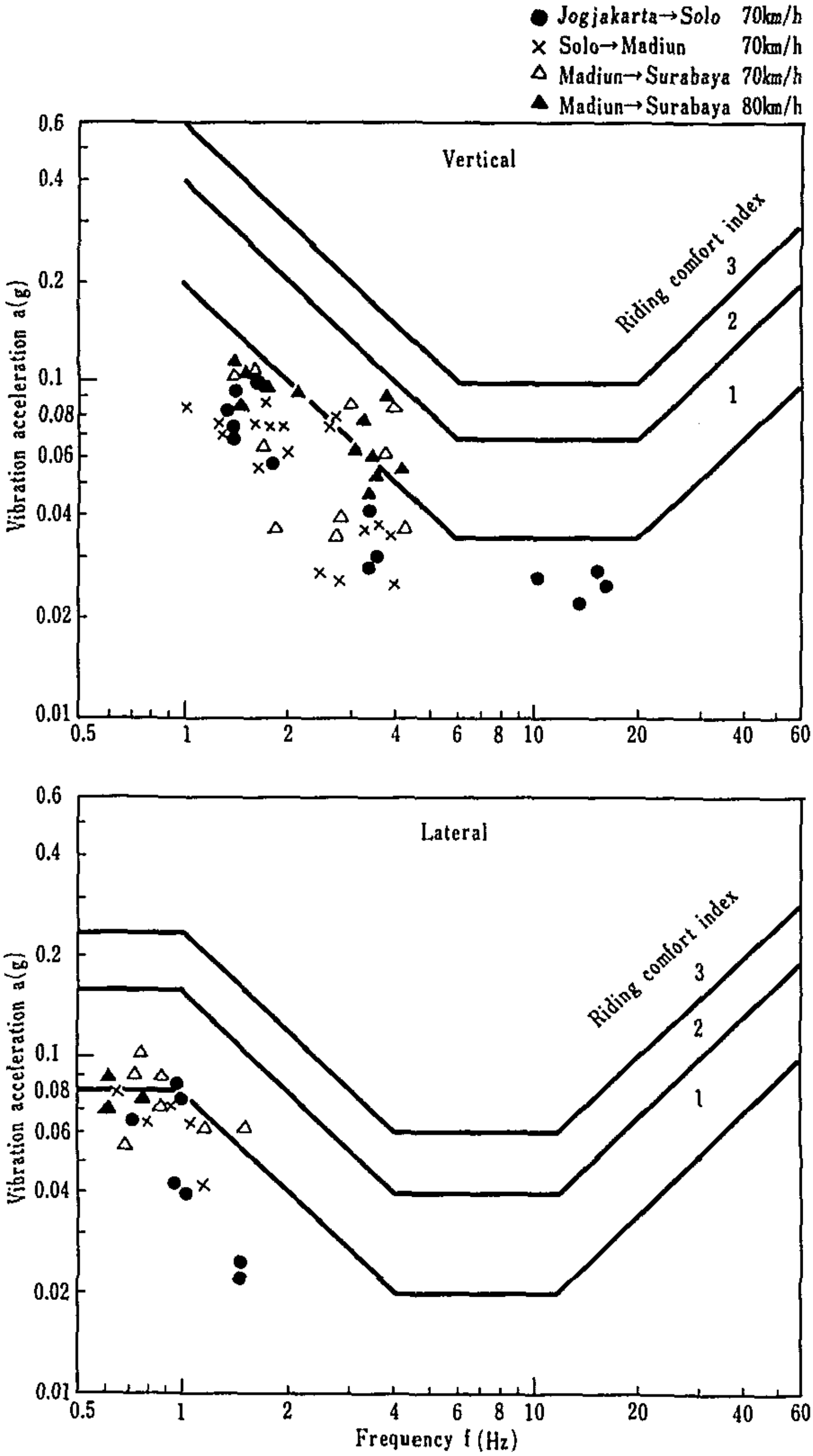


Fig. 14-2 Riding comfort of No. 1 coach
(CW 9055, Ferrostaal bogie)

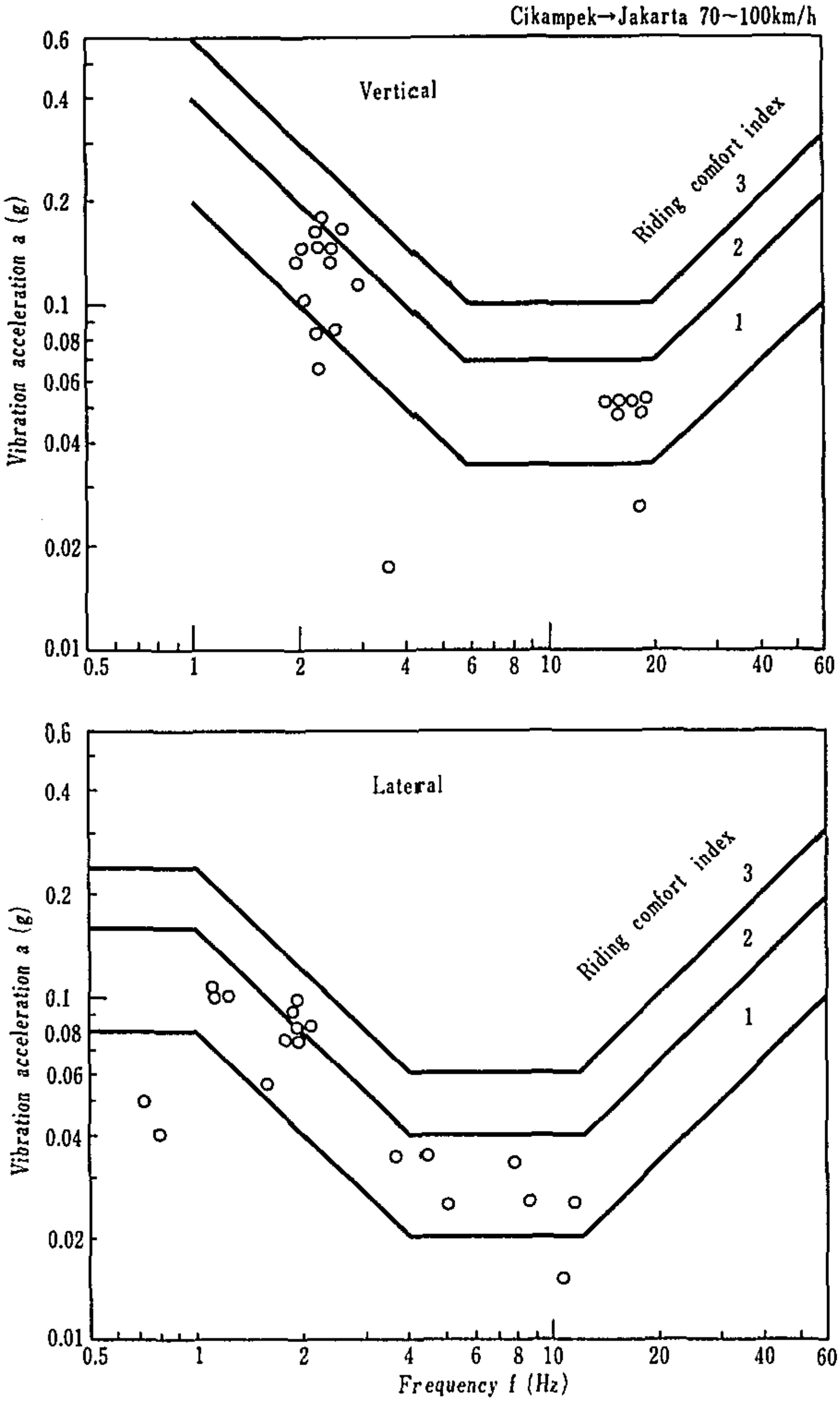


Fig.14-3 Riding comfort of No.2 coach
(CW 9108,NT-11 bogie)

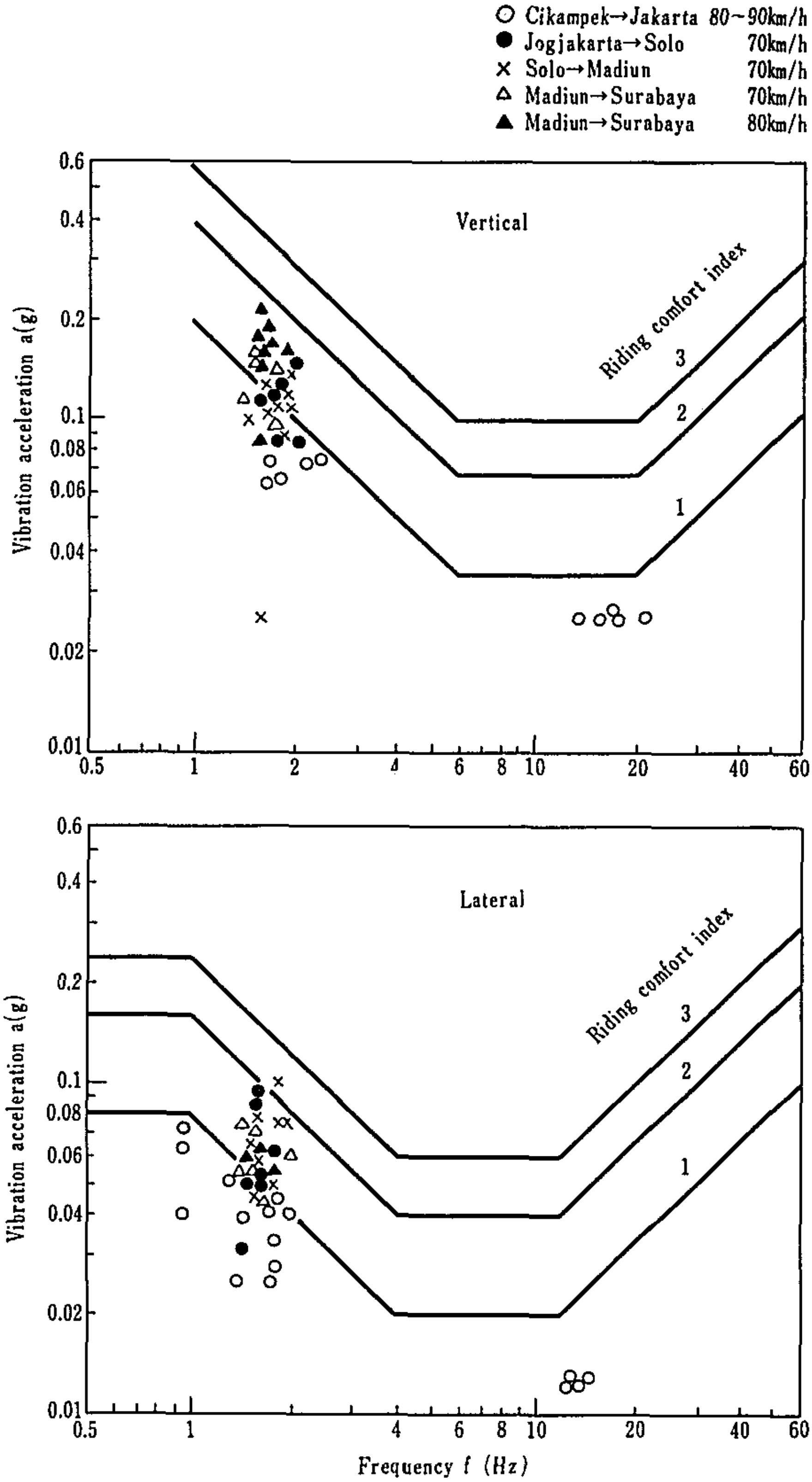
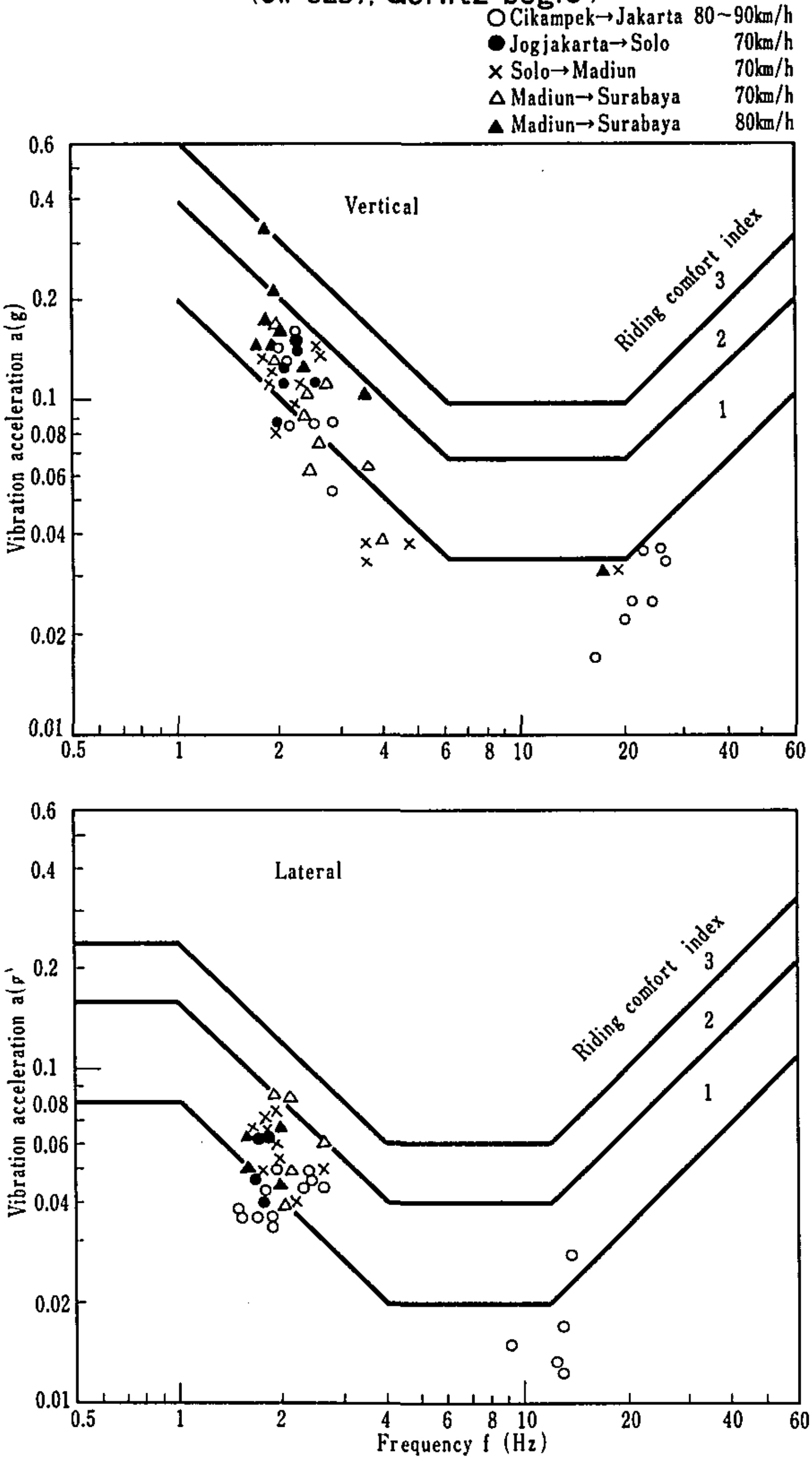


Fig.14-4 Riding comfort of No.3 coach
(CW 9257, Görlitz bogie)



Below 1	Very good
1 – 1.5	Good
1.5 – 2	Ordinary
2 – 3	Bad
Over 3	Very bad

Lateral vibration

For each passenger car, the riding-comfort index in lateral vibration is "2 or less", though a few cases of "over 2" are seen. Among all the test cars, the locomotive is best and No. 1 coach worst. High-frequency vibrations are not so problematical as to worsen riding comfort.

Vertical vibration

The riding-comfort index is below 2 for all cars at the train speed of 70 km/h, but it becomes over 2, sometimes reaching 3, for No. 2 and No. 3 coaches at the train speed of 80 km/h indicating that riding comfort seems to deteriorate at higher speeds. High-frequency vibrations are not so problematical as to worsen riding comfort.

(4) Riding comforts specified by PJKA

The riding comfort is expressed as follows in PJKA:

Read is the average value of the full amplitudes of the constant waves of 1–2 Hz which occur constantly for about 200 m, the riding-comfort index is calculated by the following formula (Koffman's formula) and judgement is made according to the standard given in Table 4.

$$R_H \text{ or } R_V = 0.896 \sqrt[16]{\frac{a^3}{f} F(f)}$$

where:

R_H = Lateral riding-comfort index

R_V = Vertical riding-comfort index

a = Lateral or vertical vibration acceleration amplitude (cm/sec^2)

f = Frequency (Hz)

$F(f)$ = $0.80\ f^2$ for lateral, $0.325\ f^2$ for vertical

The regulation of PJKA says that the most preferable value of R_H or R_V shall be below 3.0–3.25 for passenger cars and below 4.0–4.25 for freight cars.

Table 4 Riding-comfort index of PJKA (for passenger car)

Index	Signification
1	Very good
1.5	Almost very good
2	Good
2.5	Almost good
3	Satisfactory
3.5	Just satisfactory
4	Tolerable
4.5	Not tolerable
5	Dangerous in service

Table 5 shows the riding-comfort index calculated according to PJKA system, from the values of vibration accelerations measured in the short-distance test and the long-distance test.

Table 5 Riding-comfort index by PJKA method

	Section	Running speed (km/h)	Vertical			Lateral		
			max.	min.	mean	max.	min.	mean
No. 1 coach (CW 9055)	Cikampek → Jakarta	80–89	4.10	3.68	3.88	3.54	3.35	3.42
		90–98	4.08	3.70	3.94	3.68	3.48	3.56
Locomotive (BB 30310)	Jogjakarta → Solo	70–78	3.30	2.62	3.00	3.18	2.25	2.81
	Solo → Madiun	70–79	3.20	2.85	3.05	2.90	2.48	2.74
	Madiun → Surabaya	70–79	3.36	1.98	2.98	3.90	3.08	3.25
		80–84	3.40	3.10	3.25	3.17	–	–
No. 2 coach (CW 9108)	Cikampek → Jakarta	80–89	3.18	2.97	3.04	2.98	2.88	2.93
		90–98	3.53	3.40	3.47	3.17	2.60	2.88
	Jogjakarta → Solo	70–79	3.83	3.10	3.46	3.52	2.54	3.07
	Solo → Madiun	70–79	3.68	2.18	3.29	3.65	2.91	3.20
	Madiun → Surabaya	70–79	3.74	3.18	3.47	4.05	3.15	3.66
		80–83	4.15	3.15	3.66	3.21	2.91	3.06
No. 3 coach (CW 9257)	Cikampek → Jakarta	80–89	3.98	3.71	3.84	3.00	3.00	3.00
		90–98	3.52	3.28	3.36	3.06	3.00	3.03
	Jogjakarta → Solo	70–79	3.86	3.25	3.63	3.20	2.76	3.01
	Solo → Madiun	70–79	3.89	3.41	3.64	3.38	2.83	3.13
	Madiun → Surabaya	70–79	4.26	3.04	3.58	3.51	2.81	3.22
		80–83	4.80	2.37	3.80	3.24	2.91	3.05

When the test train was run between Madiun–Surabaya at over 80 km/h, the maximum value of the vertical riding-comfort index of No. 2 and No. 3 coaches sometimes exceeded 4. But on the average, the value was between 3–4 for all test cars, a value somewhat disagreeable for passenger cars.

(5) Expression of rolling stock vibration by class

Among the methods to classify rolling stock vibrations against their running speed in a long-distance train operation, statistically by size of vibration, there is one method by dividing the whole operating section into 100 m sections, reading out the average train speed and the maximum value of the full amplitudes of vibration accelerations for each 100 m subsection, obtaining arithmetic average of the amplitudes for each speed from the frequency at each 5 km/h stage and 0.025 amplitude stage and obtaining the vibration acceleration characteristics curve by plotting the average amplitudes of vibration acceleration against the center value for each train speed (for instance, 82.5 km/h at 80-85 km/h stage). In this case, it is preferable that there is a total running distance of about 180 km. However, for convenience's sake, there is given in Fig. 15, the vibration acceleration between Cikampek and Jakarta which is classified by class in the method given above. In Fig. 15, Class A-1 represents the best condition of vibration and Class C the worst. To improve the vibration performance, it is said that it is preferable to improve rolling stock vibration to Class A (refer to APPENDIX IV).

From Fig. 15, it can be seen that all of the three coaches, No. 1 to No. 3, belong to Class A-2 for lateral vibration, having the same good performance and almost trouble-free, but that, as for vertical vibration, No. 1 and No. 3 coaches belong to Class B and No. 2 coach to just between Class A and Class B, indicating that they are in need of improvement concerning vertical vibration.

As for the increasing condition of vibration acceleration as speed goes up, a trend is seen of monotonously increasing for all cars, but the rate of increasing of vertical acceleration is larger than that of lateral in general and especially so for No. 3 coach.

6. Observation

Consolidating the results of the investigations and tests made this time, the following several problematical points in increasing the maximum train speed-up to 100 km/h are noticed for the track and rolling stock from the view point of riding comfort and train operation safety:

Fig. 15-1 Car body vibration of No. 1 coach
(CW 9055, Ferrostaal bogie)

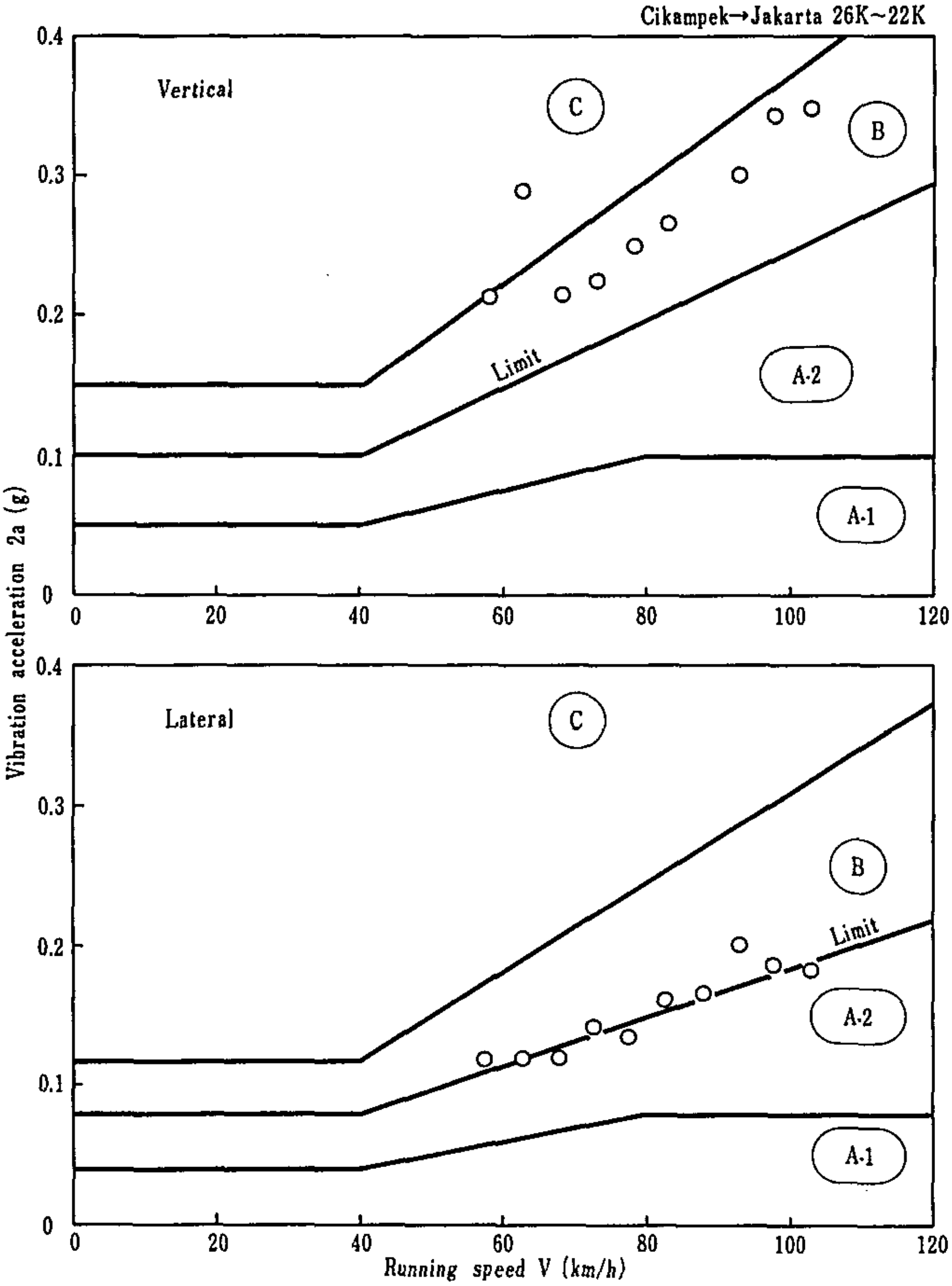


Fig.15-2 Car body vibration of No.2 coach
(CW 9108, NT-11 bogie)

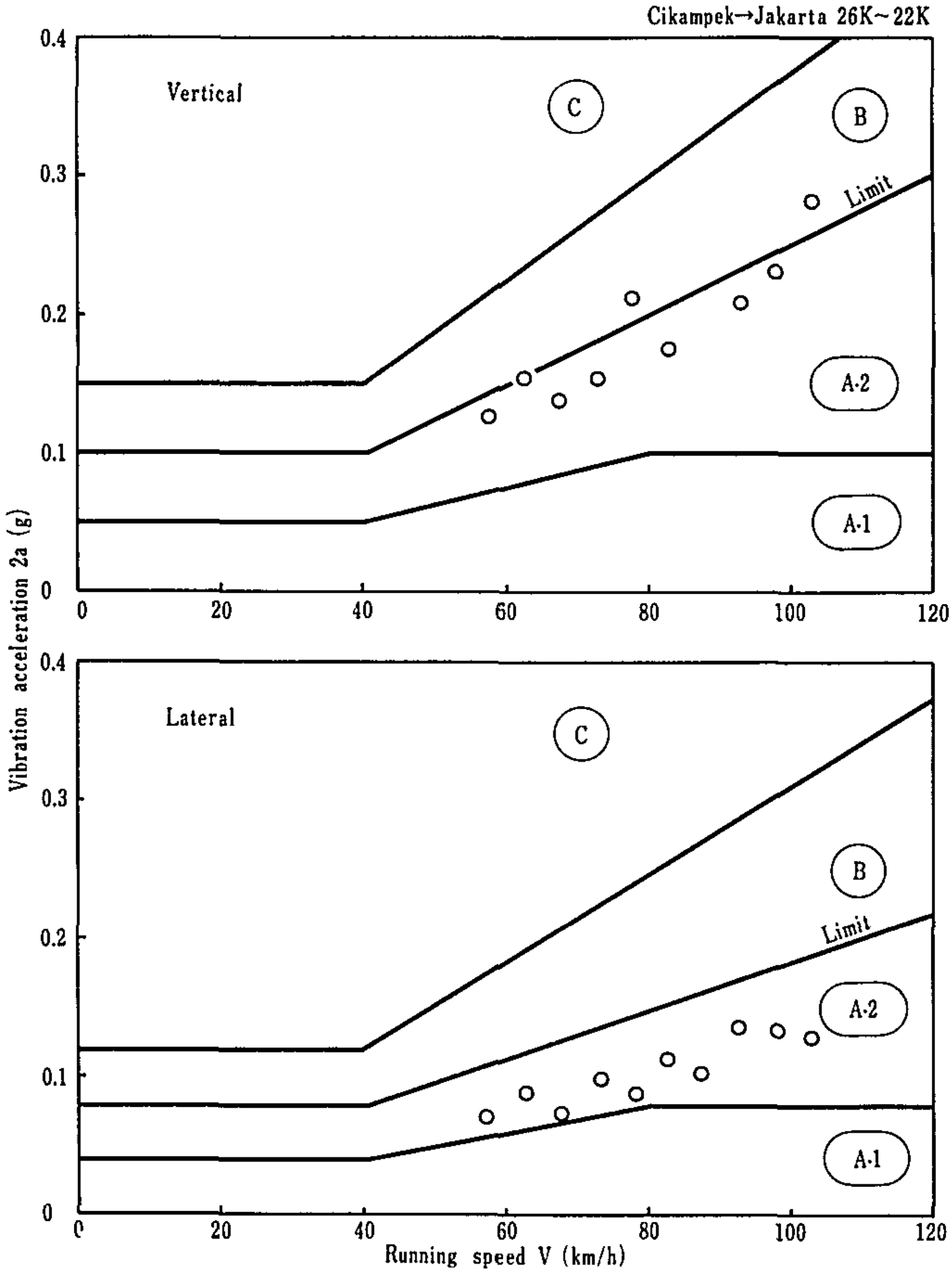
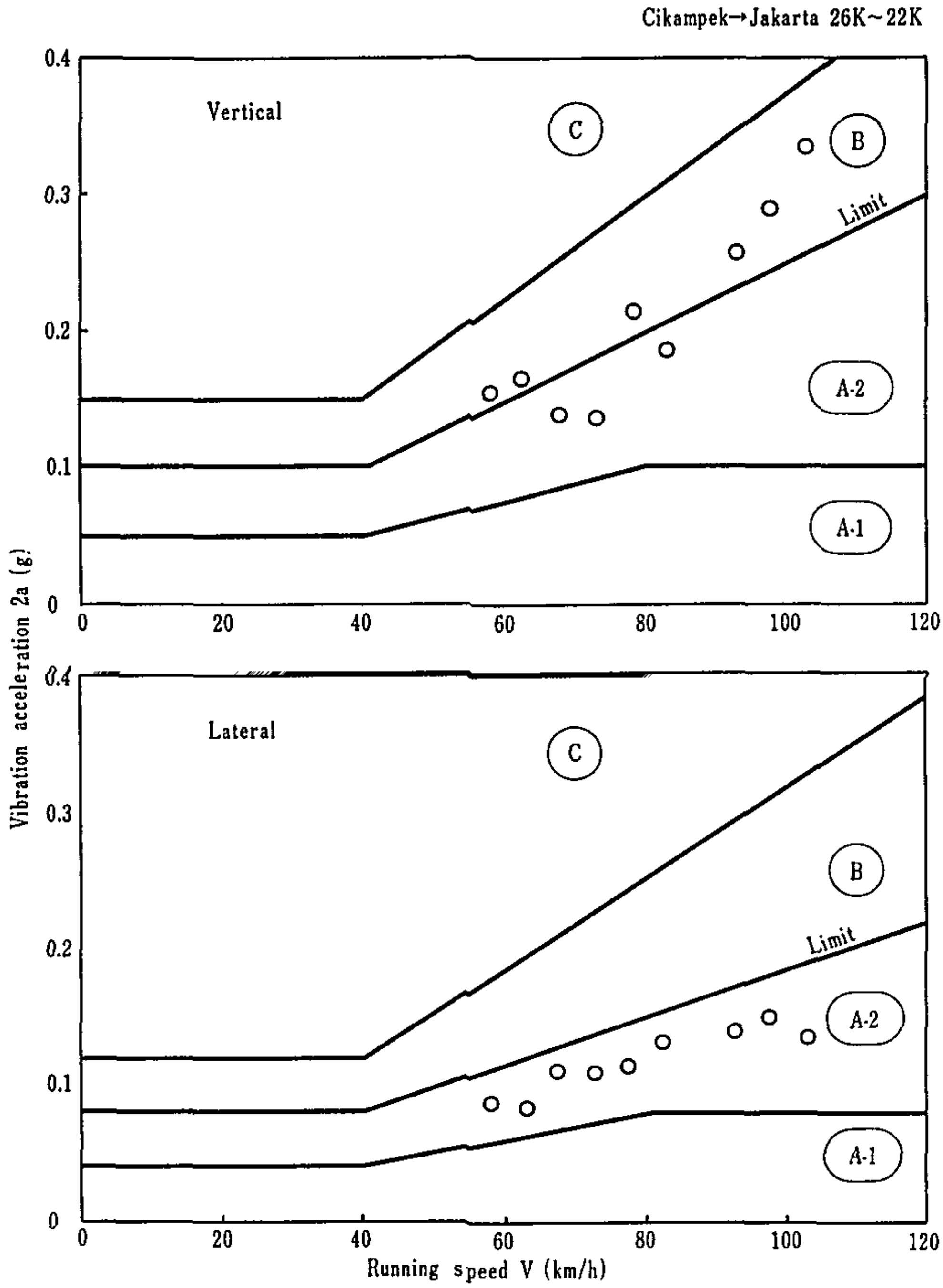


Fig. 15-3 Car body vibration of No.3 coach
(CW 9257, Görlitz bogie)



6.1 Track

(1) It is necessary to replace 15P rail with heavier rail (R14A) and secure a ballast thickness of at least 20 cm, from the viewpoint of bearing capacity of track, for 100 km/h train operation (refer to Par. 2.1.2 in APPENDIX II).

When train speed is increased, track irregularities will progress more rapidly, and therefore, it becomes necessary to strengthen so much either track or its maintenance force.

If it is intended to keep the present amount of track maintenance work even after speed-up of trains, it is recommended to strengthen the track by shortening the distance between sleepers to the range 62 to 65 cm (refer to Par. 4.3 in APPENDIX II).

As part of the strengthening of track structure, it is preferable to improve the rail-joint structure.

(2) As for track irregularities, it is necessary to better the maintenance in longitudinal level with the allowance limit of the level made smaller. Special attention should be paid to such places as rail joints, level crossings and bridges.

The progress of track irregularities depends largely on train speed besides intrinsic and geographical conditions of track. Therefore, it is necessary to make track inspections as often as possible in order to grasp the track irregularities conditions correctly, to keep the irregularities and the rolling and pitching of rolling stock within the allowable limits.

(3) As for track maintenance, it is preferable to improve the maintenance work system together with maintenance equipments and tools to raise the efficiency of track maintenance work. For the ballast, it is necessary to raise the track by overall tamping of ballast and to maintain the fixed thickness of ballast under sleeper by screening of ballast.

(4) It is important to always grasp the actual status of deterioration of track materials, especially rail, rail joints, rail fastening devices and sleepers in order to establish the maintenance plan for high-speed train operation.

(5) It is preferable to stop letting oil drop from the running rolling stock on the track, especially on the rail surface, for better track maintenance.

(6) As for deteriorated bridges (those having problems in the strength of bridge girder, abutment or pier) and soft-roadbed sections, it is necessary to make measurements of stress, deflection and vibration for the former and bearing capacity, settlement and vibration for the latter, and to consider their improvement or

allowable speeds prior to proposed train speed-up.

6.2 Rolling stock

(1) As for the rolling stock to be used for 100 km/h train operation, it seems necessary to pay attention to reducing the vertical vibration of their carbodies. For this, it is necessary to use vertical dampers and let them keep their normal characteristics.

(2) For high-speed train operation, brake performance becomes an important factor to obtain a fixed brake distance. Further, the structure of coupling device of rolling stock influences the riding comfort of rolling stock to a great extent. From these points of view, the cars of the same types as No. 1, No. 2 and No. 3 coaches are usable for the proposed 100 km/h train operation.

(3) The lateral vibration accelerations of carbodies are comparatively small in value.

This indicates that the cars are in high stability against lateral vibration and that the track is comparatively free from irregularity in alignment.

Even at 100 km/h running, generation of snake motion or hunting was not noticed. Stability against snake motion depends on the gradient of wheel tread. In general, as the train speed rises, stability against snake motion decreases, and at over a certain speed, the car becomes unstable against snake motion, and snake motion occurs. This speed is the "Limit speed of snake motion" and also the allowable speed limit of train from the point of view of snake motion. The "Limit speed of snake motion" decreases as the gradient of wheel tread increases.

As the wheel tread wears, it becomes to have a certain radius of curvature. Then, the gradient of wheel tread becomes larger equivalently. (refer to APPENDIX IV) Therefore, for high-speed train operation, it is necessary to pay attention to the wear of wheel tread.

Attention should also be paid to the wear of wheel flange, which increases the lateral movable gap of wheel axle. This makes, when the lateral vibration of wheel axle occurs, its amplitude larger, making the vibration of carbody larger, too.

(4) According to the results of the short-distance test and the long-distance test, it has become clear that lateral vibration accelerations are small for passenger cars, and that it is sufficient to make a study placing emphasis on vertical vibrations only, for the proposed train speed-up.

By comparing the result of the long-distance test with that of the high-speed test, it is understood that the carbody vertical vibration acceleration changes according to the train speed. In the high-speed test, the vertical vibration acceleration is 0.25 g → 0.3 g for No. 2 coach at the train speed 70 km/h → 80 km/h on Test Section I. On Test Section II-1, the same is 0.15 g → 0.2 g for both No. 2 and No. 3 coaches at the train speed 70 km/h → 80 km. On Test Section II-2, the same is 0.1 g → 0.25 g for No. 2 coach and 0.2 g → 0.25 g for No. 3 coach at the train speed 70 km/h → 90 km/h. On Test Section III, the same is 0.15 g → 0.25 g for No. 2 coach and 0.2 g → 0.3 g for No. 3 coach at the train speed 75 km/h → 90 km/h. On Test Section IV, the same is 0.2 g → 0.3 g for No. 2 coach and 0.15 g → 0.3 g for No. 3 coach at the train speed 75 km/h → 90 km/h. Averaging these, it can be concluded that the vertical vibration acceleration increases by 0.06 g for each speed-up of 10 km/h.

Taking the limit of vertical vibration acceleration as 0.4 g (refer to APPENDIX IV) from the viewpoint of riding comfort and providing that the track is maintained always in the present condition, the maximum allowable speed under the present condition in case where No. 2 and No. 3 coaches are used is expected to become, from the result of the long-distance test, as follows:

Jogjakarta

Kilometerage	Maximum speed
165 km – 145 km	85 km/h
145 – 127	70
127 – 112	100
112 – 260	60
260 – 215	90
215 – 200	100
200 – 184	90
184 – 180	80
180 – 166	85

Madiun

Kilometerage	Maximum speed
166 km – 138 km	90 km/h
138 – 127	50
127 – 80	80
80 – 35	75
35 – 30	50
30 – 20	80
20 – 3	50

Surabaya

The above are indicated in broken line in Fig. 10.

(5) It is necessary to prepare locomotives capable of running at 100 km/h

7. Improvement for proposed train speed-up

7.1 Improvement of track between Jogjakarta and Surabaya 322 km

(1) All 15P or R3 rails must be replaced by R14A rails (42.59 kg/m). Also, part of old R14 rails having batter at rail ends must be replaced. Total distance (track km): 90 km

(2) 15P or R3 rail turnouts must also be replaced with R14A rail turnouts, as in the case of rails in ordinary sections. Number of sets to be replaced: 52

(3) Rail joints are desirable to be improved. Total track km: 90 km

(4) The number of sleepers must be increased and part of older sleepers must be replaced with new ones. Sleeper pitching is desirable to be 62 cm. Sleepers to be prepared: 1,620 pieces/km for 68 km, 600 pieces/km for 140 km and 300 pieces/km for 114 km

(5) To increase the ballast depth under sleeper to 20 cm, crushed stone must be additionally supplied to level up the track. Distance of improvement: 26 km of 5 cm level-up section, 132 km of 10 cm level-up section and 164 km of 15 cm level-up section

(6) Machines and equipments must be prepared for the improvement works.

In this connection, Table 6 shows an example of the cost estimate for the above track improvement.

Table 6 An example of cost estimate for track improvement, Jogjakarta-Surabaya 322 km

	Foreign currency (US$)	Equivalent amount in Rp (1US$ = 415Rp)	Domestic currency	
			Clearing handling (20%) (Rp)	Material & execution (Rp)
Rail 14A & fastening	4,187,000	1,737,605,000	347,521,000	
Turnout	520,000	215,800,000	43,160,000	
Machinery	250,000	103,750,000	20,750,000	
Sleeper				457,200,000
Ballast				347,700,000
Welding				42,400,000
Work execution				688,300,000
Total	4,957,000	2,057,155,000	411,431,000	1,535,600,000
Contingency	247,850	102,857,750	20,571,550	76,780,000
GRAND TOTAL	5,204,850	2,160,012,750	432,002,550	1,612,380,000
			2,044,382,550	

Remarks: These figures are as of October 1973.

7.2 Improvement of rolling stock

As the result of measurement of rolling stock vibrations, it has been found that, to realize max. 100 km/h running of commercial trains, it is necessary to remodel or newly manufacture trucks and newly manufacture rolling stock.

First, let us pick up the number of express passenger trains that may be required to run between important cities, from the existing train time-table, and the following has been obtained:

Section	No. of trains
Cirebon – Surabaya	2
Bandung – Surabaya	2 (6 sleeping cars included)
Jakarta – Bandung	4
Jakarta – Solo	2 (3 sleeping cars included)
Jakarta – Cirebon	1
Jakarta – Surabaya (via Solo)	2 (6 sleeping cars included)
Jakarta – Surabaya (via Semarang)	2 (6 sleeping cars included)
Total	15 (21 sleeping cars included)

Taking the number of cars per train as 6 cars and the allowance ratio of rolling stock for repair, etc. as 15%, then the number of rolling stock required becomes 104 cars (including 24 sleeping cars).

Remodelling can be made by replacing of trucks or installing oil dampers. Adding to it new manufacture of rolling stock, we will estimate their numbers as follows:

(1) Installation of oil damper: 28 cars (4 trains)

(2) Replacement of trucks: 21 cars (3 trains)

(3) New manufacture of rolling stock: 21 cars including 7 sleeping cars (3 trains)

As for locomotives, taking their allowance ratio as 10%, the number required becomes 17. It is necessary to newly manufacture 4 locomotives for 3 trains.

Table 7 shows an example of the cost estimate for the above remodelling and new construction of rolling stock.

Table 7 An example of rolling stock cost estimate

Item	Foreign currency (US$)	Equivalent amount in Rp (1 US$ = 415 Rp)	Domestic currency Clearing handling (10–20%) (Rp)
Oil damper equip.	27,000	11,205,000	2,241,000
Bogie change	490,000	203,350,000	40,670,000
Passenger car	2,749,000	1,140,835,000	114,083,500
Sleeping car	1,634,000	678,110,000	67,811,000
Locomotive (1800 Ps)	2,267,000	940,805,000	94,080,500
Total	7,167,000	2,974,305,000	318,886,000
Contengency (5%)	358,350	148,715,250	15,944,300
GRAND TOTAL	7,525,350	3,123,020,250	334,830,300

Remarks: 1. Maintenance cost will increase by 56% annually for every car for 100 km/h operation.

2. These figures are as of October 1973.

In this connection, if diesel-railcar trains are adopted, the maintenance work will be complicated compared with the case of traction by diesel locomotive, as diesel engines are dispersed, while such works will become easier as parting of a train, combining of trains, turning back of trains, and so forth.

As for the expense for new construction of rolling stock, these both cases are deemed almost same in the unit of a train.

APPENDIX I

Items to be investigated

for increase of train speeds

Contents

Items to be investigated for increase of train speeds

1. Introduction

Technical investigation is made in how high train speed can be raised on straight and curved tracks or at turnouts in relation to running performance of rolling stock, and also in how much the maximum speed can be raised by improving the track and the rolling stock.

2. Factors restricting train speeds

The train speed is restricted by the mutual relation between the rolling stock and the track. The maximum available speed is determined after the running safety is confirmed by executing a running test and also after fully investigating maintenance work affairs.

The main factors to restrict the speed are as follows:

2.1 Running performance and structure of rolling stock

(a) Q/P; where Q denotes the side force of a wheel and P wheel load

(b) Riding comfort by vibration acceleration in the vertical and the lateral directions.

(c) Available maximum r.p.m. of motive engine

(d) Stress of parts of rolling stock

(e) Temperature rise of parts of rolling stock

(f) Brake performance

The parts to be investigated are engine gears (engines, convertors, etc.), running gears, brake gears and bodies of rolling stock.

2.2 Factors concerning the relation between track and rolling stock structure

(a) Derailment by abnormal sway of rolling stock or by large wheel side force on running

(b) Derailment and overturn of rolling stock by large centrifugal force on a curved track

(c) Uncomfortableness of passengers by sway of rolling stock body

(d) Progress of track irregularity and increase of labour for maintenance work caused by speed-up

(e) Destruction of track materials by vibration, shock and side force of wheels which arises during rolling stock running

2.3 Others

(a) Strength of construction (bridge beams)

(b) Wind pressure in the lateral direction

(c) Countermeasure for railway crossing accident

(d) Lateral sway of rolling stock and construction gauge (limitation of lateral play of car body on passing platform, etc.)

(e) Public nuisance (noise, vibration)

3. Factors concerning track

3.1 Straight section

(a) Bearing capacity of track (strength of track structure)

Rail bending stress and roadbed perssure regarding passing tonnage and track structure.

(b) Track deterioration (progress of track irregularity)

Investigation of the growth of track irregularity by fall down and subsidence of roadbed and of the labour of maintenance work; those are determined by load factor of train (car factor x axle load x average speed of train).

(c) Standard of track equipment

(d) Sway and riding comfort

(e) Derailment and safety

Alignment of track or hunting of rolling stock

(f) Strength of track against lateral force

Tie plates used or not, rail spikes pulled out or not.

3.2 Curved section

(a) Overturn

Speed, radius of curvature, maximum cant, allowable shortage of cant, etc.

(b) Derailment

Value of Q/P

$Q = Q_{SC}$ (side force by excess centrifugal force and bogie turning force)

$+ Q_V$ (side force by hunting motion and lateral vibration)

(c) Strength of track against side force of wheels

Irregularity of alignment of track, destruction of rail spikes by pull out, tie plates exist or not.

(d) Sway of car body and riding comfort

Riding comfort, time variation of cant, time variation of cant shortage (the latter two are related to the length of a transition curve).

3.3 Turnout area

(a) Standard side of turnout

(i) Deterioration of track (especially at the heel of turnout)

(ii) Riding comfort by sway

(iii) Strength of track against lateral force

(b) Turnout side

(i) Derailment (cant shortage)

(ii) Strength of track against lateral force by car vibration

(iii) Riding comfort by sway

4. Financial aspect in speed-up

For train speed-up, it is necessary to examine a plan of improving rolling stock and track to realize a scheduled shortened operation time with a minimum cost.

(1) Investigation of the relation between train speed and train-operation time.

(2) Calculation of the cost for increasing the speed.

(3) Comparison between train-operation time saying and cost.

(4) Determination of the speed-up financial plan of the minimum cost which satisfies a planned train-operation time.

(5) Establishment of an optimum plan in consideration of a long-term plan and other existing conditions.

APPENDIX II

Problematical points in track for proposed train speed-up

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Table List

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Problematical points in track for proposed train speed-up

1. *Introduction*

When planning speed-up of trains on a given railway line or section, it is necessary to improve track structure strong enough to withstand the proposed maximum train speed or to measure the track always to keep it in a specified standard of maintenance. The answer to the question asking how to improve the track for train speed-up varies depending on the existing condition of the track and rolling stock, policy of improvement, availability of the fund required, social environmental conditions, etc. This paper provides one example of standard method of calculation for some of the possible answers to this question.

2. Bearing capacity of track (strength of track structure)

Track receives train load via wheels. This train-load is classifiable into "wheel load" working in the vertical direction and "lateral pressure" working in the lateral direction. When trying to raise the train speed, it is necessary first of all to check the bearing capacity of each part of track to bear this load. In general, the strength or the bearing capacity of track is comparatively sufficient for sleepers and ballast to bear the wheel load. Therefore, we can judge by calculating the bearing capacity of the track mostly for rail bending stress and roadbed pressure.

2.1 Rail bending stress and roadbed pressure

The bearing capacity of track is calculated on the preconditions that rails are supported on the sleepers with a specified pitching and that ballast and roadbed supporting the sleepers have elasticity. In this case, the deformation of rails due to wheel load is almost independent between the right and the left. So, this problem may be considered only for one side of the rails of track.

2.1.1 Method of calculation

(1) Rail bending stress

The maximum bending moment of rail is generated, just below the wheel load and at the center between two adjacent sleepers. Its calculation formula is simplified as follows:

$$M = \Sigma F(\gamma)\ a\ W$$

where:

M : Maximum bending moment of rail (kg-cm)

$F(\gamma)$: Bending moment coefficient of rail, function of γ

a : Sleeper pitching (cm)

W : Wheel load (kg)

$$\gamma = \frac{B}{D}$$

B : Center load required for rail to be deformed a unit displacement (cm) at the center of its span taking the rail for a simple beam with a span of 2a (kg/cm)

D : Rail pressure required for sleeper to generate settlement of a unit displacement (cm) at the bottom of the rail (kg/cm)

$$B = \frac{6Es\ Ix}{a^3}$$

Es : Elastic modulus of rail steel (kg/cm^2)

Ix : Moment of inertia of rail (cm^4)

$$D = \frac{Cb}{K[\eta]}$$

C : Ballast coefficient (kg/cm^3)

b : Width of sleeper (cm)

$$K = \sqrt[4]{\frac{Cb}{4EoI'x}}$$

Eo : Elastic modulus of sleeper (kg/cm^2)

I'x : Moment of inertia of sleeper (cm^4)

$[\eta]$: Coefficient and function of $K\ell$, $K\Upsilon$

ℓ : 1/2 of the length of sleeper (cm)

Υ : 1/2 of center distance of the pair of rails (cm)

(2) Rail pressure

The maximum rail pressure is generated just below the wheel load and just above the sleeper, and its calculation formula is simplified as follows:

$$P = \Sigma f(\gamma)\ W$$

where:

P : Maximum rail pressure (kg)

$f(\gamma)$: Settlement coefficient of rail supporting body, function of γ

(3) Roadbed pressure

The ballast pressure on the bottom face of the sleeper just below the rail is:

$$Ps = \frac{KP[\eta]}{b}$$

where:

Ps : Ballast pressure on the bottom face of the sleeper just below rail (kg/cm^2)

The roadbed pressure is decided by the ballast pressure and ballast thickness.

Its value is:

$$pr = PoPs$$

where:

pr : Pressure on the surface of roadbed (kg/cm^2

Po : Coefficient and $\frac{50}{10 + h^{1.35}}$ according to JNR's empirical formula.

h : Ballast thickness (cm)

2.1.2 Calculation of values

Calculation of values is made considering the following items:

(1) In the Jogjakarta-Surabaya section of PJKA, to be investigated and tested this time, there exist various track structures. But, among them, only the representative track structure is taken up (refer to Table II-1 ~ 3).

(2) As for the bearing capacity of track to be used in the calculation, the cases of ballast coefficients 5 kg/cm^3 and 13 kg/cm^3 are compared and checked and the larger one is taken up.

(3) The train-speed is presumed as 100 km/h.

When a rolling stock runs on track, wheel load increases dynamically due to effects of irregularity on the rail surface, rolling and pitching of rolling stock, flat of the wheels, etc. The speed impact coefficient represents this phenomenon, expressed in relation with the speed of the rolling stock.

In JNR, this speed impact coefficient is:

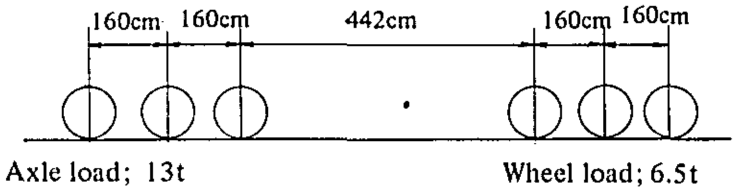
Against the bending moment of rail;

$$i = 1 + \frac{V}{100}$$

Against the pressure of rail-supporting body;

$$i = 1 + \frac{0.6\ V}{100}$$

(4) Out of the locomotives now held by PJKA, BB201 series is considered to give the maximum effect on the bearing capacity of track, under the proposed 100 km/h operation. So, this locomotive will be taken up for calculation. The wheel arrangement and wheel load of BB201 are as follows:



The result of the calculation of values is given in Table II-3 As can be seen from the table, in the case of R14A or R14 rail and sleeper pitching 68 cm, and in the case of 15P rail and sleeper pitching 65 cm, a ballast thickness of 20 cm is required from the view point of the bearing capacity of track.

3. Track irregularities and carbody vibration acceleration

3.1 Vertical vibration acceleration and irregularity in the longitudinal level of track

The following formula has been obtained, in JNR, by checking the interrelation between the maximum value of vertical vibration acceleration for each km and irregularity at corresponding point measured by JNR's track inspection car.

$$\alpha = 1.465 + 0.254X + 0.0419V$$

where:

α : Maximum value of vertical acceleration in 1 km (in 0.01 g)
The value is measured by track inspection car.

X : Amount of irregularity in longitudinal level of the track at the point of generation of vibration (mm)
The value is measured by track inspection car, and is larger than the static one obtained by the ground measurement.

V : Train speed (km/h)

From this formula, it can be known that, within the speed-up range of by 10 – 20%, increase of vertical vibration of car body accompanying speed-up by about 6 km/h is almost equal to that accompanying additional irregularity of longitudinal level by 1 mm. In general, to raise train speed without worsening riding comfort, it is necessary to mitigate the level of track irregularities by the amount equivalent to the intended speed-up.

3.2 Lateral vibration acceleration and irregularity in alignment of track

The interrelation between the lateral vibration and irregularity in alignment of the track is not so clear as in the case of the vertical vibration. It seems very difficult to represent this relation in a certain formula, as it is affected by the relation among the radius of track curvature, the cant, the train speed, etc., on curved section, and by the track irregularities, wear of wheel flanges, etc. on straight section or curved section with a fairly long radius of curvature. But one guidance can be given as follows: According to the experiment made by JNR using the track inspection car, it has been found, in the train speed range of 70 – 100 km/h, that the amount of increase or decrease in the lateral vibration due to difference in the irregularity by 1 mm is equal to that due to difference in train speed by 2 to 5 km/h.

4. Track destruction

4.1 Track destruction and structure coefficient

The track is a structure subject, at all times, to destruction such as settlement of ballast, deformation and wear of various parts of the track due to repeated passage of train. The term, "Destruction", used here in a narrow sence denotes irregularities progressively generated in the track due to settlement of the ballast by the combined action of train load and vibration induced by passing of trains. Repairing work against this destruction occupies a larger part of track maintenance work, and so it seems little error to the conclusion to observe track structure mostly from this point of view.

Factors directly related to track destruction are, among others, train load, track structure and condition of track materials.

According to the result of experiments, settlement of ballast increases in proportion to the magnitude of load and also to the frequency of repetitions of load. Therefore, as for the load condition, it is necessary to consider the multiplication of the magnitude and the frequency of repetitions of the load, or "passing tonnage".

At the same time, the settlement of ballast increases in proportion to the vibration acceleration occurring in ballast. Therefore, after all, settlement becomes to increase in proportion to train speed. Furthermore, ballast vibration receives different effects depending on the structure of rolling stock composing the train. Therefore, it is necessary to consider rolling stock coefficient as a quantitative measure of this effect.

To be more precise, the magnitude of track destruction due to train load can be expressed by "Load coefficient" which is the multiplication of passing tonnage, train speed, and rolling stock coefficient.

For the same track structure, track destruction is proportional to the load coefficient. But, for different track structures, track destruction differs even when the load coefficient is the same. As the measure to express the level of strength of the track against vertical destruction, "Structure coefficient" M is defined including track structure.

Ballast settlement is proportional to the multiplication of ballast pressure and ballast acceleration. Therefore, let us find out difference in it due to difference in track structure. Let us consider the ballast vibration acceleration, classifying it into the acceleration generated when the wheel load is let drip from a specified height and the magnitude of impact generated when the rolling stock is run on the track. Then:

Structure coefficient M = (Ballast pressure) x (Ballast acceleration by wheel impact of a specified magnitude) x (Impact coefficient)

As structure coefficient is a relative value, it is convenient to express it by the ratio against the value of a standard track construction. The smaller value indicates the smaller track destruction.

Even in the case of the same track structure, the degree of track destruction differs depending on the condition of materials of track, for instance, age of rails, rate of deterioration of sleepers, rate of soil mixing into the ballast, number and condition of rail joints, etc. Therefore a coefficient that can be called "Condition coefficient" may be considered. But, it is still very hard to grasp it quantitatively. So, let us disregard it in the present study.

4.2 Calculation of structure coefficient

4.2.1 Method of calculation

Structure coefficient M is represented as follows:

$M = Pb \cdot \ddot{y} \cdot S / (Pb \cdot \ddot{y} \cdot S$: standard track)

where:

Pb: Ballast pressure

$\ddot{y}$: Ballast vibration acceleration

S: Impact coefficient

The pressure on the ballast, Pb can be found by the conventional calculating method for track stresses and here the maximum value of pressure which arises just below the wheel load is assumed.

If upon tracks of different structures, wheels and axles are dropped from a fixed height and the maximum values of acceleration which arise on the ballast are calculated, it is found that the ballast acceleration is chiefly related to both the spring constant of sleeper compression k_1 and the mass of support m, and is little related to other values.

$$\ddot{y} \propto \sqrt{k_1} \cdot \frac{1}{\sqrt{m}}$$

The impact on the track arises when the wheel hits upon the rail due to existence of a low spot at a rail joint. The amount of the shock is ascertained through experiments as follows.

$$S = \frac{1}{EIx\beta^2}$$

where:

EIx : vertical bending rigidity of rail

$$\beta = \sqrt[4]{\frac{k}{EIx}}$$

k : spring constant of rail support (sleeper and roadbed)

Note: Yutaka Satoh: "The Load-Factor, Structure-Factor and Condition-Factor of a Railway Track", Permanent Way No. 7 June 1960.

4.2.2 Calculation of values

Table II-4 and Fig. II-1 show the results of calculation of structure coefficient M made for representative track structures of PJKA. Table II-5 and Fig. II-2 show the result of calculation of structure coefficient made when the sleeper pitching and ballast thickness are changed using R14A rail.

4.3 Volume of track maintenance work and track construction

As was already explained, when planning to raise train speeds, it is preferable to minimize the magnitude of track irregularities. On the other hand, the higher the train speed and the larger the traffic volume, the quicker the speed of track destruction. All of them can serve as factors to increase the volume of track maintenance work required.

On the other hand, strengthening the track lowers the speed of track destruction. Therefore, when planning to raise the train speeds, it is usual to strengthen the track to suppress the resultant increase in the volume of track maintenance work required.

Suppose we intend to raise the maximum train speed from the existing 80 km/h to 100 km/h or 1.25 times, and the average speed of all trains operating on the line concerned 1.1 times. Then, as the volume of track maintenance work accompanying reduction of the allowable value of track irregularities is considered to increase almost at the same rate as the rate of increase of the maximum train speed, and, also, as the speed of track destruction increases almost at the same ratio as the ratio of increase in the average value of speed of trains operating on the section in question, the volume of track maintenance work required to maintain the track in the existing maintenance condition becomes about $1.25 \times 1.1 = 1.38$ times.

This means that, if we want to keep the track maintenance work volume at the same volume as heretofore, it becomes necessary to strengthen the track structure so that the structure coefficient of the track becomes 1/1.38 of that under the existing track structure. For instance, if the existing track structure is R14 rail wooden sleeper with 68 cm pitching, crushed stone ballast of 5 – 10 cm thick, or the structure coefficient 1.62 – 1.40, then it is necessary to make the track with the structure coefficient 1.17 – 1.01 or thereabout, to keep the track maintenance work volume almost unincreased. To be more plain, it may be sufficient to fix the distance between sleepers at 58 to 66 cm, practically at 62 to 65 cm in the case of the track in which R14A rail is used and the ballast thickness under sleeper is 20 cm.

However, sometimes it may be more advantageous to increase track maintenance workers or to mechanize track maintenance work than to strengthen the track structure, depending on the labor supply situation of the railways concerned. So, to choose a method it is necessary to consider economic advantageousness, too.

5. Strength of track against lateral pressure

What call for our attention relative to track lateral pressure are, among others, sharp generation of irregularity in the alignment, expansion of the track gauge due to pushing out of rail spike by rail and decrease in the rail fastening force due to being-pulled-up of the rail spike by tilting of the rail. For these three items, there exist limits something like the "yielding point", above which, if the lateral pressure works, the part concerned may deform excessively and may not be able to restore to its original shape.

No measurement was made, this time, on the lateral pressure performance of the locomotive, and, so, concrete study cannot be made. But let us imagine it from the lateral pressure performance of the comparable locomotive of JNR and compare it with the track structure of PJKA. Then, it is judged that the track structure of PJKA studied has full lateral-pressure capacity against the operating locomotives.

Table II-1 Particulars of rail

Item \ Kind of rail		R14A	R14	15P	R3
Rail weight	(kg/m)	42.59	41.52	38.00	33.40
Rail head width	(mm)	68.5	68.0	64.0	58.0
Rail bottom width	(mm)	110	110	110	105
Rail height	(mm)	138	138	134	134
Height of neutral axis (from bottom)	(mm)	68.3	69.5	66.5	67.3
Section modulus Wx	(cm^3)	200	196	177	154
Section modulus Wy	(cm^3)		43		28.7
Moment of inertia Ix	(cm^4)	1,360	1,369	1,180	1,036.6
Moment of inertia Iy	(cm^4)		235	205	150.7
Cross-sectional area	(cm^2)	54.26	58.1	48.64	42.5
Rail length	(m)	17.0	17.0	14.0	13.6

Table II-2 Particulars of sleeper

Item \ Kind of sleeper		Wooden sleeper	Steel sleeper
W x T x L	(cm)	22 x 12 x 200	23.2 x 7.5 x 200
Moment of inertia Ix	(cm^4)	3,168	162.5
Weight	(kg)	48	47
Elastic modulus	(kg/cm^2)	100,000	2,100,000

Table II-3 Calculation of bearing capacity of track

Kind of rail		Unit	R14A		R14		15P	
Vertical bending rigidity of rail Es Ix		(kg/cm^2)	2,874.9 x 10^6		2,856.0 x 10^6		2,478.0 x 10^6	
Section modulus of rail Dx		(cm^2)	200		196		177	
Kind of sleeper			Wooden		Wooden		Wooden	
Width of sleeper b		(cm)	22		22		22	
1/2 of length of sleeper ℓ		(cm)	100		100		100	
Sleeper pitching a		(cm)	68		68		68	
Vertical bending rigidity of sleeper Eo I'x		(kg/cm^2)	3,168 x 10^5		3,168 x 10^5		3,168 x 10^5	
1/2 of rail center-to-center distance γ		(cm)	56.78		56.78		56.55	
Ballast coefficient C		(kg/cm^3)	5	13	5	13	5	13
Rail bending moment		(kg/cm)	126,157	112,948	126,044	112,855	121,849	109,249
Rail stress		(kg/cm^2)	(Note 1) (631)	565	(643)	576	(688)	617
Rail pressure		(kg)	3,112	(3,188)	3,112	(3,189)	2,967	(3,054)
Values increased depending on speed (Note 2)	Rail stress	(kg/cm^2)	1,262		1,286		1,376	
	Rail pressure	(kg)	5,100		5,102		4,886	
	Ballast pressure	(kg/cm^2)	2.32		2.32		2.22	
Roadbed pressure (kg/cm^2)	Ballast thickness (cm)	10	3.57		3.57		3.42	
		15	2.37		2.37		2.26	
		20	1.74		1.74		1.67	

Note: 1. The bearing capacity of the track was calculated for Ballast coefficient 5 kg/cm^3 and 13 kg/cm^3, and the one of the larger value was taken up for final calculation.

2. "Speed" was pressumed to be 100 km/h.

3. The allowable bearing capacity of the track at annual passing tonnage about 10 million ton and train speed about 100 km/h is about:

Rail bending stress: 1,550 kg/cm^2

Ballast pressure : 3.3 "

Roadbed pressure : 2.2 kg/cm^2

Therefore, for any of the three track structures given in the table, if ballast thickness is made 20 cm, its conditions are satisfied.

Table II-4 Calculation of structure coefficient

Kind of rail		R14A	R14	15P	40N (Note 1)
Vertical bending rigidity of rail Es Ix ($kg\text{-}cm^2$)		$2,874.9 \times 10^6$	$2,856.0 \times 10^6$	$2,478.0 \times 10^6$	$2,894.0 \times 10^6$
Kind of sleeper		Wooden	Wooden	Wooden	Wooden
Width of sleeper b (cm)		22	22	22	20
1/2 of length of sleeper ℓ (cm)		100	100	100	105
Vertical bending rigidity of sleeper Eo Ix ($km\text{-}cm^2$)		$3,168 \times 10^5$	$3,168 \times 10^5$	$3,168 \times 10^5$	$4,573 \times 10^5$
Sleeper pitching a (cm)		68	68	65	66
1/2 of rail center-to-center distance r (cm)		56.78	56.75	56.55	56.6
Compression coefficient of sleeper D' (kg/cm)		100,000	100,000	100,000	100,000
Supporting weight m (kg)	Ballast thick-nessed (cm)				
	5	218	318	218	
	10	233	233	233	
	15	247	247	247	
	20	274	274	274	273
	25	316	316	316	
Structure coefficient M	Ballast thick-nessed (cm)				
	5	0.95×10^{-5}	0.96×10^{-5}	1.01×10^{-5}	
	10	0.83×10^{-5}	0.83×10^{-5}	0.87×10^{-5}	
	15	0.77×10^{-5}	0.77×10^{-5}	0.81×10^{-5}	
	20	0.72×10^{-5}	0.72×10^{-5}	0.76×10^{-5}	0.59×10^{-5}
	25	0.67×10^{-5}	0.67×10^{-5}	0.70×10^{-5}	
When 40N is made the standard structure (Note 2)	5	1.61	1.62	1.69	
	10	1.39	1.40	1.48	
	15	1.30	1.31	1.38	
	20	1.21	1.22	1.28	1
	25	1.13	1.13	1.18	

Note: 1. Under the heading "40N", are given values for the standard track structure of JNR, i.e. 40N rail with rubber pad, tie plate, wooden sleeper, sleeper pitching 66 cm crushed stone ballast x 20 cm, for annual passing tonnage 6.5–10 million ton at max. speed 105 km/h.

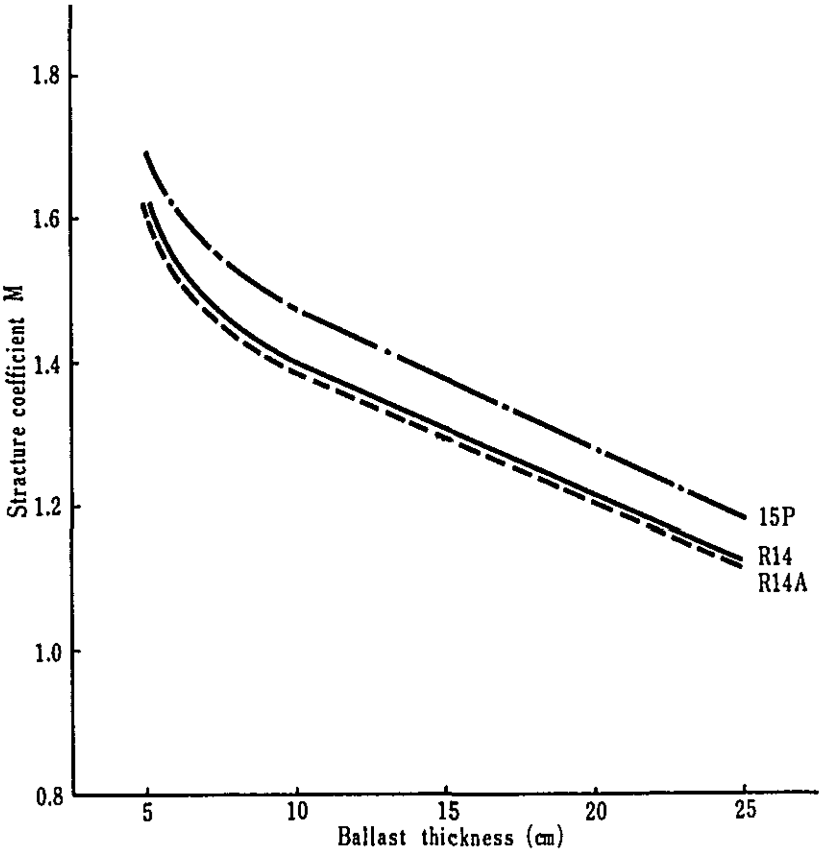
2. The structure coefficient is a relative one, and abovementioned 40N rail track structure is made 1 (one) and other track structure are expressed in the ratio against the standard track structure.

Table II-5 Structure coefficient of track using R14A rail

Sleeper pitching \ Ballast thickness	20 cm	25 cm
68 cm	0.72×10^{-5} (1.21)	0.67×10^{-5} (1.13)
65	0.68×10^{-5} (1.15)	0.63×10^{-5} (1.07)
62	0.64×10^{-5} (1.09)	0.60×10^{-5} (1.01)
59	0.61×10^{-5} (1.03)	0.56×10^{-5} (0.95)
56	0.57×10^{-5} (0.97)	0.53×10^{-5} (0.90)
53	0.53×10^{-5} (0.90)	0.50×10^{-5} (0.84)

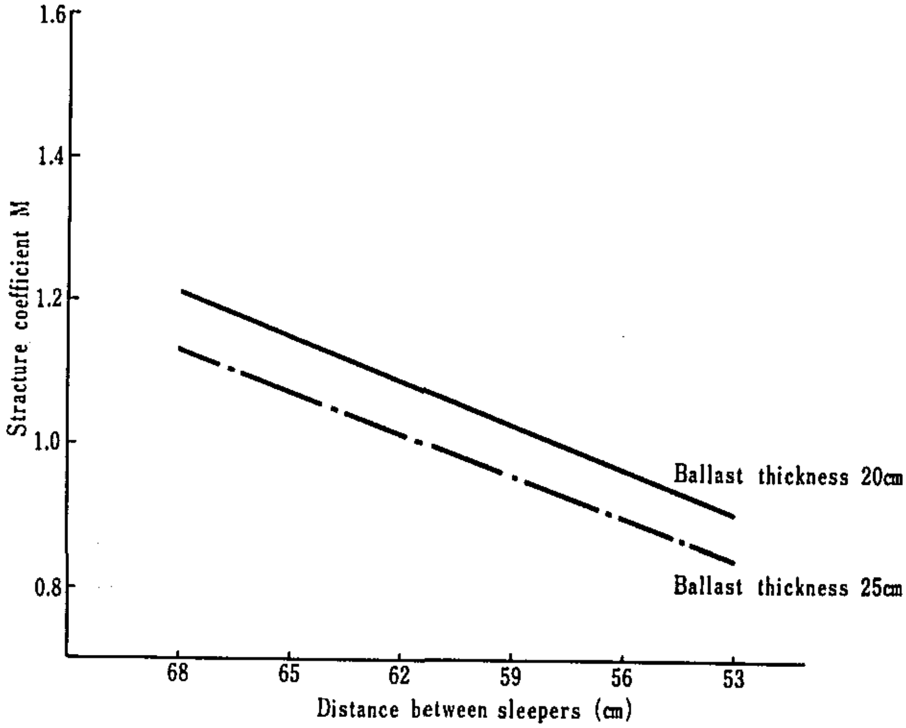
Note: Figures in ( ) show the value of the structure coefficient expressed in the ratio against that of standard track structure using 40N rail, taking it as 1.

Fig. II - I Stracture coefficient



Note: Stracture coefficient is based on "$40N\frac{W_{TPD}}{39}\cdot\frac{C}{200}$".

Fig. II-2 Stracture coefficient (rail : R14A)



Note: Stracture coefficient is based on "$40N\frac{W_{TPD}}{39} \cdot \frac{C}{200}$".

APPENDIX III

Calculation formula for and values of carbody vertical vibration acceleration

Contents

Calculation formula for and values of carbody vertical vibration acceleration

1. Introduction

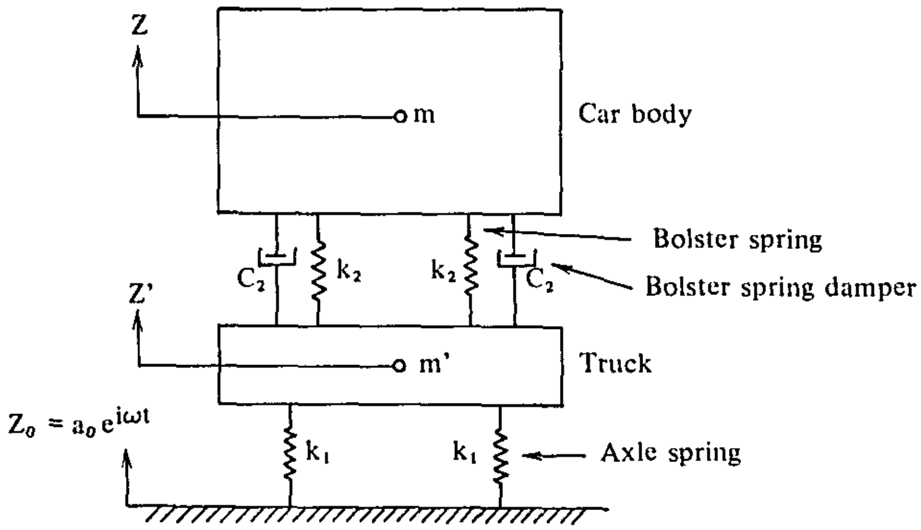
When considering the carbody vertical vibration of a bogie car, let us make the following simplification to make it easy to grasp its outline and simplify calculation:

(1) As for the carbody, its front half (or rear half) only will be considered.

(2) As for the trucks, only one truck will be taken up, and the mass of its center part between the bolster spring and axle spring will be considered.

(3) The wheels and axle are supposed to move along the track.

As for the bogie passenger cars being tested this time, their vertical dampers are being considered for the bolster spring only. Therefore, the vibration system is considered as shown in the figure below:



2. Nomenclature

Symbol	Unit, calculation formula	Description
a	m	Aplitude of carbody displacement
a_0	m	Amplitude of forced vibration
c_2	kg s/m	Damping coefficient of bolster spring damper (1/2 of truck)
g	$= 9.8 \, m/s^2$	Acceleration of gravity
k_1	kg/m	Spring constant of axle spring (1/2 of truck)
k_2	kg/m	Spring constant of bolster spring (1/2 of truck)
m	$kg \, s^2/m$	Mass of carbody (The part above bolster spring, 1/2 of carbody)
m'	$kg \, s^2/m$	Mass of truck (The part between bolster spring and axle spring, per truck)
m_W	$kg \, s^2/m$	Mass of wheels and axle (The part below axle spring, per truck)
Z	m	Vertical displacement of carbody
Z_0	m	Forced displacement given by rail or wheels
ϵ_2	$= 2C_2/(m\nu_1)$	Dimensionless amount expressing damping coefficient of damper
κ	$= k_2/k_1$	Ratio between spring constants of bolster spring and axle spring
μ	$= m'/m$	Ratio of masses of truck and carbody
ν_1	$= \sqrt{2k_1/m}$: rad/s	Vertical natural angular frequency when carbody is supported by axle spring only
η	$= \omega/\nu_1$	Dimensionless amount expressing frequency of forced vibration
ω	rad/s	Angular frequency of forced vibration

3. Calculation formula for carbody vertical vibration acceleration

3.1 Calculation formula for acceleration

The carbody vertical vibration acceleration is expressed by $a\omega^2$(m/s^2). But let us make it dimensionless and express it as $a\omega^2/(a_0\nu_1{}^2)$. The calculation formula for it is:

$$\frac{a\omega^2}{a_0\nu_1{}^2} = \sqrt{\frac{(\kappa^2 + \epsilon_2^2\,\eta^2)\eta^4}{\Delta_B}}$$

$$\Delta_B = [\mu\eta^4 - \{1 + (1+\mu)\kappa\}\,\eta^2 + \kappa]^2 + \epsilon_2^2\eta^2\,\{1-(1+\mu)\eta^2\}^2$$

Let us make η a variable value, and make calculation changing η one after another and the characteristic curve can be obtained.

3.2 Calculation of resonance frequency when attenuation coefficients of damper are 0 and ∞.

That the attenuation coefficients of the damper are 0 and ∞ means that ϵ_2 are 0 and ∞. The resonance frequency as at that moment becomes, when that is considered to be the point when the amplitude of acceleration is ∞, as follows:

(1) When $\epsilon_2 = 0$

$$\frac{a\omega^2}{a_0\nu_1{}^2} = \frac{\kappa\eta^2}{|\mu\eta^4 - \{1+(1+\mu)\kappa\}\,\eta^2 + \kappa|}$$

Placing 0 on the denominator, and taking η at that time as η RO, then:

$$\mu\eta_{RO}^4 - \{1+(1+\mu)\kappa\}\,\eta^2 + \kappa = 0$$

$$\eta_{RO}^2 = \frac{1}{2\mu}\,[1+(1+\mu)\kappa \mp \sqrt{\{1+(1+\mu)\kappa\}^2 - 4\mu\kappa}\,]$$

as η is positive:

$$\eta_{RO} = \sqrt{\frac{1}{2\mu}\,[1+(1+\mu)\kappa \mp \sqrt{\{1+(1+\mu)\kappa\}^2 - 4\mu\kappa}\,]}$$

(2) When $\epsilon_2 = \infty$:

$$\frac{a\omega^2}{a_0\nu_1{}^2} = \frac{\eta^2}{|1-(1+\mu)\eta^2|}$$

Placing 0 on the denominator, and taking η at that time as $\eta_{R\infty}$, then:

$$1-(1+\mu)\eta^2 = 0$$

$$\eta_{R\infty} = \sqrt{\frac{1}{1+\mu}}$$

Note that, at that time, the acceleration when $\eta \to \infty$ is taken is:

$$\left(\frac{a\omega^2}{a_0\nu_1^2}\right)_{\substack{\varepsilon_2=\infty \\ \eta=\infty}} = \frac{1}{1+\mu}$$

4. Points unchanging against damping coefficient of damper

Looking at Fig. 7, we find three points unchanging even when ϵ_2 is changed to various values. Let us call them, from the one with η of the smaller value, as P_1, Q, Q_2, and the value of η of each point can be expressed by the following formula:

$$\eta_{P_1} = \sqrt{\frac{1}{2\mu}\left[1+2(1+\mu)\kappa - \sqrt{\{1+2(1+\mu)\kappa\}^2 - 8\mu\kappa}\right]}$$

$$\eta_Q = \sqrt{\frac{1}{\mu}}$$

$$\eta_{P_2} = \sqrt{\frac{1}{2\mu}\left[1+2(1+\mu)\kappa + \sqrt{\{1+2(1+\mu)\kappa\}^2 - 8\mu\kappa}\right]}$$

And, the value of acceleration at each of these points is:

$$\left(\frac{a\omega^2}{a_0\nu_1^2}\right)_{P_1} = \frac{\eta_{P_1}^2}{1-(1+\mu)\eta_{P_1}^2}$$

$$\left(\frac{a\omega^2}{a_0\nu_1^2}\right)_{Q} = 1$$

$$\left(\frac{a\omega^2}{a_0\nu_1^2}\right)_{P_2} = \frac{\eta_{P_2}^2}{(1+\mu)\eta_{P_2}^2 - 1}$$

5. Value table

The following values were used for the calculations made this time:

Item	No.1 coach (empty)	No.2 coach (empty)	No.3 coach (empty)	Locomotive
$(m + m' + m_W) \times g$	11,950	14,600	14,500	21,400
$(m' + m_W) \times g$	5,500	5,500	5,500	7,500
$m \times g$	6,450	9,100	9,000	13,900
$m_W \times g$	3,000	3,000	3,000	4,000
$m' \times g$	2,500	2,500	2,500	3,500
$\mu = m'/m$	0.388	0.275	0.278	0.252
k_1 (per half truck)	121,200	166,400	268,400	246,600
k_2 (per half truck)	181,300*	97,000	99,200	92,600
$\kappa = k_2/k_1$	1.50	0.583	0.370	0.376
ν_1	19.19	18.93	24.18	18.65
	(3.05 Hz)	(3.01 Hz)	(3.85 Hz)	(2.97 Hz)
ϵ_2	0	0	0	0
(4 kinds)	0.5	0.5	0.5	0.5
	1.0	1.0	1.0	1.0
	∞	∞	∞	∞
c_2/ϵ_2	6,320	8,790	11,100	13,200
(per half truck)				
η_{R01}	0.722	0.595	0.514	0.518
η_{R02}	2.72	2.45	2.24	2.36
$\eta_{R\infty}$	0.849	0.886	0.885	0.894
	13.85	11.26	12.43	9.66
	(2.20 Hz)	(1.79 Hz)	(1.98 Hz)	(1.54 Hz)
$\left(\frac{a\omega^2}{a_0\nu_1^2}\right)_{\substack{\epsilon_2=\infty \\ \eta=\infty}}$	0.720	0.784	0.775	0.799
η_{P_1}	0.780	0.704	0.635	0.640
η_Q	1.61	1.91	1.90	1.99
η_{P_2}	3.56	2.92	2.57	2.70
$\left(\frac{a\omega^2}{a_0\nu_1^2}\right)_{P_1}$	3.93	1.35	0.834	0.838

Item	No.1 coach (empty)	No.2 coach (empty)	No.3 coach (empty)	Locomotive
$(\frac{a\omega^2}{a_0\nu_1^2})Q$	1.0	1.0	1.0	1.0
$(\frac{a\omega^2}{a_0\nu_1^2})P_2$	0.764	0.864	0.888	0.897

Note: * For a laminated spring, an equivalent spring constant (ke) is used, which is calculated by the following formula:

$$ke = k_2 \left[1 + \frac{\bar{\mu}\delta}{a + 0.01\delta}\right]$$

where $\delta = P_S/k_2$ (61.0 mm)

P_S : static load (4,475 kg)

$\bar{\mu}$: frictional coefficient (0.14)

a : amplitude (5.2mm; 2.2 Hz, 0.1 g)

k_2 : (73,400 kg/m)

APPENDIX Ⅳ

Running safety and vibration problems of railway rolling stock

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Running safety and vibration problems of railway rolling stock

1. Preface

Railway Technical Research Institute of JNR has for some decades been making research on running safety and vibrational characteristics of rolling stock. Especially, during the last twenty years theoretical and experimental studies on these subjects have been actively advanced. On the other hand, as great many tests were performed on running safety and riding comfort of many rolling stock, a considerable empirical knowledge in this field has been acquired. The results have been fully utilized in the design of the railcar and the track of the SHINKANSEN, and they have become the foundation to make the high speed railway of great safety and good riding comfort.

A great many pages will be necessary to describe the details of these studies and tests, so this report is primarily intended to give an account of the results only. It will be mainly bogie cars that are dealt with in this paper, while the two-axle freight cars and the locomotives attended by conditions of driving system are laid aside as they have their own special problems.

2. Running safety

2.1 Overturn

Overturn of railway vehicles can be caused by the centrifugal force working on the car during the run on the curved track, or by the wind pressure working on the side of the body.

Actions of the centrifugal force and the gravitational force on a car running on the curved track are shown in Fig. IV-1. The car overturns when the resultant of the two forces at the center of gravity of the car works in a direction outside the lines which connect the gravity center with the wheel-and-rail contact points. In order to prevent the overturn, elevation of the outer rail of the curved track must be increased. Too much canting, however, makes passengers uncomfortable for the big inclination of the car body when the train stops at the curve. From the experimental studies the permissible limit of the lateral static acceleration in regard to the riding comfort is said to be 0.1 to 0.13 g. Therefore the limit of the angle of the cant $\alpha = C/G$ is 1/10 to 1/8. The relations between the overturn limits of the running speed and the curve radius of the track with the cant angle of 0.1 is shown in Fig. IV-1. From this figure it is clear that the safety against overturn is higher when the height of the center of gravity of the car above the rail level is lower and when the track gauge is wider.

As mentioned above, passengers feel uncomfortable when the steady state acceleration of 0.1 g acts laterally. Therefore, when the resultant of the components of gravitational force and centrifugal force in the direction parallel to the car body floor exceeds 1/10 of the gravitational force, the riding comfort is impaired. Fig. IV-1 indicates also the relation between the running speed and the radius of curve at this critical condition when the cant angle is 0.1. It is clear that the comfort limit takes a lower position than the overturn limit at the same radius of curvature. Therefore, the train speed is restricted by the comfort limit of the passengers.

In the foregoing observation, the effects of the lateral displacement and the inclination of the car body due to its suspension systems (the spring system and the swing hanger system of the truck) are disregarded. When these effects are taken into consideration, the lateral displacement of the center of gravity of the car body lowers the overturn speed limit and the inclination of the car body lowers the comfort speed limit, but these effects upon the speed limit are generally only about 10%.

When a vehicle receives a lateral wind pressure on the side of the car body while passing a curve, it is liable to overturn. Fig. IV-2 shows, as an example of the results of calculation, the relation between the critical wind velocity for overturn and the running speed in the case of the SHINKANSEN railcar, considering the lateral wind pressure, the centrifugal force, the gravitational force and the inertia force caused by the vibration of car body.

As is seen in the figure, when the car runs through the curve below its balancing speed, an overturn toward the inside of curve is more liable to occur, and its critical wind velocity is minimum at the running speed about 80 km/h. While at speeds above the balancing speed, the overturn toward outside is more liable to occur than the overturn toward inside, and as the running speed becomes higher the critical wind velocity becomes smaller, approaching to zero at the overturn speed limit mentioned in the foregoing passages.

The figure tells that the most dangerous condition against the overturn occurs during the run on the curved track at a speed fairly lower than the balancing speed, except the case when the vehicle runs at a very high speed.

2.2 Derailment

Force working between a wheel and a rail are shown in Fig. IV-3, where Q represents the lateral force and P the vertical. It is clear, theoretically and experimentally, that the larger is Q and the smaller is P, the easier for the wheel to override the rail. The ratio of Q to P is interpreted as an index of the possibility of derailment, so it is called "derailment quotient".

If the condition shown in Fig. IV-3 is to be sustained, forces in the figure must be balanced. Thus the following equation is derived from the conditions of equilibrium of forces.

$$\frac{Q}{P}=\frac{\tan\alpha-\mu}{1+\mu\tan\alpha} \qquad (1)$$

where, μ is the coefficient of friction between the wheel and the rail at their contact point, α is the angle of the tangent at the contact point measured from the horizontal line, which is nearly equal to the flange angle (inclination of the inside surface of the flange). Fig. IV-3 shows the balanced condition and if Q becomes larger or P becomes smaller, the wheel flange begins to climb the rail, eventually riding on the rail top and then falling over. Therefore the derailment quotient calculated by the equation (1) gives the critical value.

Although the critical value of derailment quotient for the static side thrust and static wheel load working between wheel and rail is given by the equation (1), the value for the impulsive side thrust which often appears in high speed run and for the sudden drop-off of the wheel load can not be given by this equation. When the impulsive side thrust acts on a wheel, the "jump-up derailment" occurs. When a theoretical analysis is made of the derailment of this kind and a formula giving the critical value of the derailment quotient is introduced, it is found that the duration of the impulse has much to do with the occurrence of derailment.

When the relative angle between the wheel and the rail in respect of running direction (attack angle or incidence angle) is taken into consideration, both the equation (1) which gives the critical value of climbing derailment and the equation which gives that of jump-up derailment must be modified. Nevertheless, the equation (1), which is very simple and gives the safety side value, may be conveniently used as an expendient for determining the possibility of derailment.

The criterion to determine the running safety against derailment is shown in Fig. IV-4. Depending upon the duration of side thrust, the figure is divided into two parts, climbing derailment ($t_1 > 1/20$ sec) and jump-up derailment ($t_1 < 1/20$ sec), and the criterion is shown respectively.

2.3 Causes of increase of side thrust

As mentioned above, the chance of derailment is greater with the larger side thrust and the smaller wheel load.

The causes of increase of side thrust may be divided into the following five categories:

(a) Vibration of vehicle, mainly hunting

When the car body vibrates in lateral direction, side thrust reacting for the inertia force of vibration arises. Therefore the magnitude of side thrust is proportional to the inertia force due to the vibration of car body. An example of the wave form of this kind of side thrust is shown in Fig. IV-5ⓐ.

Hunting of the truck (Hunting will be specially dealt with in later pages.) also causes the side thrust. The side thrust of this kind is proportional to the lateral vibration acceleration of the truck. The wave form is shown in Fig. IV-5 ⓑ

(b) Swivelling of car in passing a curve

When the car passes a curve, side thrust arises against the outer and inner rails. An example is shown in Fig. IV-5ⓒ. The side thrust against the outer rail is obviously larger than that to the inner rail. The side thrust relates to the frictional coefficient between the wheel and rail. It depends upon the radius of the curve, the flange clearance, the tyre conicity, the wheel diameter and the wheel base of the truck. Moreover it is influenced by the gap and the elasticity existing between the axle and the truck or the car body.

(c) Local irregularity of the track

Very large side thrust is produced when the wheel flange hits against the rail which has irregularities, especially local irregularity in alignment.

An example is shown in Fig. IV-5ⓓ. The product of the side thrust and the duration of its working is in many cases nearly constant at a certain place irrespective of the running speeds.

(d) Static force such as the lateral component of the gravitational force caused by the cant, the centrifugal force or the wind pressure.

(e) Buckling of the train

When a heavy axial force such as abrupt braking works on a train, buckling occurs in some cases. Forces transmitted from front and rear draft gears make a car body rotate around the vertical axis as shown in Fig. IV-6(a), or make it displace sideways as shown in Fig. IV-6(b). In either case, the side thrust at the wheels is produced.

2.4 Causes of drop-off of wheel load

Causes of the reduction of the wheel load, or the wheel load drop-off, are broadly classified into the following four categories.

(a) Car vibration, mainly rolling.

(b) Static force, such as the lateral component of the gravitational force caused by the cant, the centrifugal force or the wind pressure.

In the cases (a) and (b), total wheel load of one side of a car decreases vibrationally or statically.

(c) Eccentricity of the center of gravity of car body or uneven distribution of passengers and cargo.

When the center of gravity of car body shifts longitudinally, a difference is made to the wheel load between the front and rear trucks. If it shifts laterally, a difference is made to the wheel load between the both sides of the car. Thus the wheel load of the one side of the car is reduced statically.

(d) Twist of the track or initial distortion of car body

The track is twisted in the section of the transition curve where the elevation of the outer rail is gradually changed along the curve and also in the section where the irregularity of the cross level is not uniform. For another case, the car body has more or less an initial distortion. The relative position of track and vehicle in such cases is shown in Fig. IV-7. In such cases the summation of the diagonal wheel loads of a vehicle is smaller than the summation of another diagonal wheel loads. In a bogie car the summation of the wheel loads on one side of front truck and the wheel loads of the other side of rear truck is different from the summation of the remaining four wheel loads. Thus the drop-off of the wheel load occurs at the wheels in the diagonal position.

These are the phenomena which have been recently made clear through theoretical and experimental studies, as the problem of the drop-off of wheel load caused by the track twist or the car body distortion. The tank cars (when empty) with a long wheel base and with stiff springs have high possibility to derail on the transition curve because of the drop-off of wheel load.

In cases where the spring is the laminated leaf spring and the friction of the leaves is too large, or where the height of the springs or the spring characteristics are not uniform, the same result as in the case (d) is brought about.

3. Vibration of rolling stock

3.1 Classification of the car vibration

Car vibrations are classified as the vibration of sprung mass and that of unsprung mass. The vibration of sprung mass is the vibration of car body and may be broadly divided into the two kinds. One is a car body oscillation in which a body oscillates as a rigid body, and the other is a body shake in which a body vibrates wholly or locally as an elastic body.

The car body oscillations are generally classified into next six kinds, depending upon the mode of vibrations.

Translational motions	Longitudinal oscillation Lateral oscillation Vertical oscillation
Rotational motions	Rolling around longitudinal axis Pitching around lateral axis Yawing around vertical axis

Among these vibrations pure lateral vibration generally does not come out separately, but combined with rolling, and results in the primary rolling (rotational vibration whose center is situated at a lower position) and the secondary rolling (center is situated at an upper position).

Predominant vibrations of usual bogie passenger car or electric motored car in running are as follows;

The *vertical oscillations comprise pure vertical vibration and pitching* with a frequency range of 1.2–2.5 Hz, and they are excited by the irregularity of the track longitudinal level or uneven sinking of the rail.

The lateral oscillations comprise the primary rolling with the frequency 0.6–0.8 Hz which appears generally in the running speed range less than 50–60 km/h, and the yawing with the frequency 1.0–2.0 Hz which appears in the higher speed range. These lateral oscillations are excited by the irregularities of cross level or alignment or by the hunting of rolling stock.

As for the longitudinal vibration, especially in the case of electric motored cars, it sometimes occurs heavily with the frequency range of 5–8 Hz at running speeds more than about 70 km/h. This vibration is exicted violently near the rail joint where the local sinking is very large.

The shaking comprises primary or secondary mode of elastic car body vibration (vertical or lateral bending vibration or torsional vibration) with the frequency range of 6–15 Hz, local vibrations of car body occurring numerously in the frequency range more than about 15 Hz, truck vibrations (mainly truck pitching) with the frequency 5–8 Hz and unsprung mass vibrations scattered in the frequency range of 20 –60 Hz. All these vibrations, which appear as a mixture, are excited by the eccentricity of the wheel, corrugation of the rail surface, local abrasion of tyre or rail surface (skidding and slipping abrasion) or impulse of the rail joint.

The magnitude of these vibrations differs much depending upon the quality of the construction of car body and truck.

3.2 Forced vibration arising from the unevenness of rail surface

The vibration system shown in Fig. IV-8 represents a half of a bogie car equipped with axle springs and bolster springs. As for the bogie car, the radius of gyration of car body around the lateral axis passing through its center of gravity is nearly equal to half the length between the centers of front and rear trucks. Therefore, when the vertical vibration is dealt with, it will be sufficient if observation is made of the vibration system of one truck on which a half of the car body is supported as shown in the figure.

Now consider a case where the vertical sinusoidal displacement is given to the axle by the unevenness of the rail surface. When the amplitude of forced displacement is uniform and frequency is changing, the resonance curves of vibration acceleration of the car body are calculated to be as shown in Fig. IV-8. Each curve in the figure corresponds to a value of damping force of the damper which is equipped in parallel with the bolster spring.

From the figure and also considering the results of the theoretical analysis, the following matters are clear.

(1) The vertical forced vibration of bogie car has two resonant points. At the first resonant point the car body and the truck vibrate in the same phase, where the amplitude of the vibration of car body is larger. At the second resonant point they vibrate in anti-phase, where the amplitude of the vibration of car body is very small and that of truck is very large.

(2) There are three fixed points P_1, Q, P_2 where all resonance curves, regardless of the damping force of the damper, pass through. Therefore the peak of the resonance curve cannot be made lower than these fixed point no matter what damping force may be selected.

(3) When the damping force is increased gradually from zero, the peak of the first resonant point is remarkably reduced. At a certain damping, the peak coincides with the P_1 and the height becomes minimum. Further increase of the damping makes the peak higher again, until the peak becomes infinite because of the resonance by the axle spring only where the bolster spring does not work.

(4) The vibration acceleration at the secondary resonant point is not much influenced

by the damping force. The vibration acceleration at the point Q depends theoretically only upon the stiffness of axle spring, and therefore the peak can be lowered by decreasing the stiffness of the axle spring. Point P_2 can be left out of consideration as it is always lower than the point Q.

(5) In order to make the maximum height of the resonance curve as low as possible, the damping coefficient must be selected in such a way that the resonance curve becomes maximum at either the point P_1 or Q whichever is higher. If it is allowed to choose a suitable stiffness ratio of the bolster spring to the axle spring, the stiffness ratio κ and damping coefficient C_2 can be determined so that the height of the point P_1 and Q becomes equal and that the resonance curve becomes maximum at both points P_1 and Q.

However, making the height of P_1 and Q equal is only a standard for the selection of the stiffness ratio κ. It is not that this standard is the best. The value must be appropriately decided in accordance with the condition of the track and vehicle.

3.3 Hunting

When a set of an axle and wheels with conical tyres rolls on the rails, as shown in Fig. IV-9, it moves along the sinusoidal passage which consists of lateral translational motion and yawing. This motion is called hunting, and its wave length is given by the following formulas:

$$S = 2\pi \sqrt{br/\gamma e} \qquad (2)$$

$$\gamma e = \gamma/(1-\rho_R/\rho_T) \qquad (3)$$

where, 2b is the distance between the contact points of both wheels to the rails, r is the radius of the wheel at the contact point to the rail, γe is the effective conicity of the tyre, γ is its geometrical conicity, and ρ_R and ρ_T are the radii of the respective curvatures of the rail and tyre at the contact point, so γ is equal to the inclination of the tangent at the same point. When the standard tyre of JNR is new and correct in shape, $\rho_T = \infty$ and $\gamma = 1/20$, therefore $\gamma e = 1/20$. However, when it wears, ρ_T becomes small and gradually approaches to ρ_R. Thus the γe sometimes becomes larger than 1/10, the wave length of the hunting becomes short commensurately with the wear.

When the relative displacement of the two axles is restricted both in longitudinal and lateral directions as in the case of a truck, the hunting of a wheelset of two wheels and one axle does not occur, but the hunting of a truck as a whole can arise. The theoretical formula of the wave length in this case is expressed as follows:

$$S = 2\pi \sqrt{\frac{br}{\gamma}\left(1 + \frac{a^2}{b^2}\right)} \qquad (4)$$

where 2a represents the wheel base.

In the calculation, the wave length of a truck hunting (given by the formula (4)) becomes much larger than that of a wheelset given by the formula (2) (usually three times as large). However, in actual cases there exists a certain amount of clearance between the axle and the truck frame. When the clearance is large the restriction on the relative displacement becomes imperfect, so the wheelset starts hunting.

There are two kinds of hunting: self-excited hunting and forced hunting. The former is more important than the latter, and it becomes unstable at a certain speed range even if the track is completely level and straight. Once it becomes unstable, as shown in Fig. IV-9, the amplitude of the wheelset increases successively and it finally comes to be limited within the range of flange clearance accompanied by the collision of wheel flanges against rails. The energy to increase the amplitude is supplied from the forward movement of wheelset whose kinetic energy is partly converted in lateral direction by the medium of the creep forces acting between wheelset, truck and car body, the self-excited hunting occurs usually in a certain critical speed range, being attended by the severe lateral oscillation of car body. There are two kinds of the critical speed; one exists in a low speed range and another in a high speed range. The hunting in the former is called the primary hunting or the body hunting, where the oscillation of car body is more remarkable. Therefore, the magnitude of damping adopted in the suspension system of the car body has a great bearing on the prevention of this vibration. The hunting at the critical speed in the high speed range, is called the secondary hunting or the truck hunting. The main mode of this hunting is the rotational vibration of the truck. For the prevention of vibration of this kind, the magnitude of frictional resistance produced by the relative rotation between the car body and truck and the stiffness of the spring acting in series with this friction must be taken properly.

The phenomenon of self-exicted hunting varies widely with the change of shape of tyre tread due to wear, the change of the supporting stiffness of axle in longitudinal and lateral directions and the magnitude of gaps in the supporting part.

The forced hunting is induced by the laterally bent deformation, or irregularity in alignment, of the rail. Its amplitude becomes very large because of resonance when the wave length of the rail deformation coincides with the natural wave length of the hunting. As the lateral deformation of the rail is usually not regular and continuous, the forced hunting is generally non-steady.

4. Diagnosis of vibration

Diagnosis of the rolling stock vibration means a series of devices beginning with the examination of the condition of car vibration when it occurs and, through the detection of cause, ending in the setting up of a measure towards the solution of the problem. Just like the health examination, examining in general in pursuit of a method of prevention of possible problems even if the vibration does not matter at present, and obtaining the reference data of vibration when it occurs, area also implied by the diagnosis.

Therefore the contents of the series of devices are classified into the following: (1) Investigation on the condition of vibration (2) examination of natural vibration characteristics (3) studies on the source of vibration (4) inference of the cause of vibration (5) decision of the methods to prevent the vibration.

Investigation of the condition corresponds to the examination of the disease. It is important to check the condition of vibration minutely and to make clear the special features. The natural vibration characteristics mean the natural features of vibration of the car and imply the natural frequency, damping characteristics and mode of vibration, which can be obtained by theoretical calculation or by the experiments. The studies on the source of vibration mean finding out all existing sources of vibration and source of energy including the source which is generated in the rolling stock itself. When the characteristics of natural vibration and the sources of vibration are found out, the response of the rolling stock to the work of these sources can be made clear. Thus, by comparing the response with the special features of the vibration in question, causes of the vibration can be inferred and methods of prevention of vibraiton can be decided.

The vibrometer most commonly used for measurement of vibration of rolling stock is the vibration accelerometer. So in this paper methods to diagnose the vibration by the wave forms of the vibration acceleration of the car body are dealt with, supposing that such wave forms are measured by the accelerometer.

4.1 Diagnosis by the nature of wave form of vibration

When the wave form of the vibration is known, the diagnosis is often made fairly well. Some of the typical wave forms are shown in Fig. IV-10.

When a steady state wave form as shown in Fig. IV-10(a) is obtained on the rolling stock, it is quite probable that the source of the vibration is in the rotational system. The wave form of vibration arising from the wheel eccentricity, traction motor, diesel engine and auxiliary machines are all like this. Sometimes the vibration of this type is caused by the hunting or the regularly located rail joints. The cause of the composite wave form as shown in Fig. IV-10(b) is generally not single, except the causes ascribable to the diesel engine, from which the exciting forces of the frequency of one, two, three or six times or half its rotational frequency can be produced. Wave form Fig. IV-10(c) is sinusoidal, which is often caused by the rotary machine or the hunting. Wave form Fig. IV-10(d) is the frictional, which is often produced by the laminated leaf spring that has too much friction or by the relative motion of axle box against the axle box guide lacking clearance in between. It must be noted in this connection that a similar wave form is produced when the pressure of the writing pen of the accelerometer upon the recording paper is too large. Wave form Fig. IV-10(e) represents the shock. If this wave is due to the vibration in lateral direction, it is probably attributable to the collision between the wheel and rail, between axle box and axle box guide, or between swing bolster and side sill of the truck. If the wave comes from the vibration in vertical direction, the full-stroke of the axle spring or the bolster spring is suspectable. The full stroke of the draft gear, full stroke of the shock absorber, collision of the driving force transmission parts in the truck or car body of electric locomotive or motive railcar, often cause the shock wave in longitudinal direction. Wave form Fig. IV-10(f) represents the beat which is caused by the difference between the wheel diameters of the front and rear trucks, difference of the rotational speeds of the driving mechanism or difference between the vibrational characteristics of front and rear tracks.

4.2 Diagnosis by the frequency analysis

The purpose of the frequency analysis is to find out what frequency of vibration with what amplitude is contained in a measured wave form. The amplitude and the wave form of measured vibration vary with the running speeds of the rolling stock. The wave form is by no means stationary. However, when the wave form is checked locally, it sometimes happens that the stationary wave is sustained for a while. To put it concretely, the vibration is regarded as stationary when the wave form appears stationarily or when more than three or four continuous waves with nearly equal period and amplitude come out. In these cases it is easy to read out the frequency and amplitude of the vibration.

Now, the values are plotted in graph whose axis of abscissa corresponds to the running speed V of the rolling stock and axis of ordinate corresponds to the frequency f as shown in Fig. IV-11. Then the circle whose center is located at the plotted point is drawn with the radius which is proportional to the acceleration amplitude 2a. Repetition of drawing of circle at each step of running speed eventually makes the figure as shown in Fig.IV-11.

When the centers of the circles line up on the oblique straight line in the figure, it shows that the forced vibration whose frequency is proportional to the running speed is excited. From the inclination of this oblique straight line to the axis of abscissa the wave length of the exciting force is obtained (wave length S = V/f). As a matter of course, the longer the wave length, the smaller is the inclination of the straight line. The shorter the wave length, the larger is the angle of the straight line.

When the centers of the circles line up along a lateral straight line, it shows that a natural frequency of the car body or the truck exists nearby. Therefore, the circle having grown large shows that the resonance occurs by the coincidence of forced frequency and natural frequency. With this figure it is clear that what kind of vibration occurs against the running speed and at what speed the vibration comes into resonance.

As the spring system of usual truck has considerable friction and the car body has internal damping, the natural frequency is somewhat influenced by the amplitude; as the amplitude becomes larger, the frequency becomes smaller. Therefore the centers of the circles in the figure seldom line up just along the same horizontal line, and the frequency becomes smaller as the circle becomes larger. In this case even if the frequency differs to some extent the group of the circles can be regarded as representing the same natural vibration.

Wheel base, peripheral length of the wheel, wave length of hunting of the wheelset or the truck, center distance between the front and rear trucks and length of a rail are often given as the wave length of the exciting force. Calculating and drawing the oblique line in the figure prior to the analysis are very convenient to obtain the expectation of the analysis.

4.3 *Allowable limit of vibration for riding comfort and running safety*

According to the results of experiments conducted by foreign countries and our laboratory the standard of the riding comfort which has the frequency characteristics of determined as shown in Fig. IV-12.

The standard shown in this figure is led from the statistically summarized results of human body experiment in which many subjects were made to suffer various kinds of vibration on shaking tables and evaluate the comfort by their own sensitivity. As the criterion of the riding comfort, this standard is applied to all kinds of rolling stock including the SHINKANSEN railcars.

In this figure, the riding comfort index 1 is the standard, or a sort of datum line, derived from the riding comfort experiment and the indices 2 and 3 mean that the amplitudes are twice and three times, respectively, as large as the amplitude of index 1. Further, ① – ⑤ are the symbols of evaluation of the riding comfort. The rating ① of the riding comfort is very good, ② good, ③ average, ④ bad and ⑤ is very bad.

Evaluating the riding comfort by the magnitude of acceleration, as seen in the figure, the vibration in any direction with the frequency 5 – 15 Hz affects the riding comfort most. With the same magnitude of vibration, the effect is greatest in the lateral direction, and smallest in the vertical direction, the vibration in longitudinal direction being in between. In the case of lateral vibration, the boundary of ratings of riding comfort is constant in frequency range less than 1 Hz. It means that the lateral acceleration impairs the riding comfort even if it works stationarily. So it is seen that the stationary lateral acceleration due to the excessive centrifugal force produced by the high speed running of the rolling stock on the curved section must be held down under this boundary.

When the large vibration occurs in running, the vibration can be estimated by reading the frequency and amplitude and plotting them in this figure. In case the vibration comes into the zone ④ or ⑤ , a measure must be taken from the viewpoint of riding comfort.

Though convenient for the experiment on a short section or for the determination of the riding comfort in relation to steady vibration, this standard is not easy to utilize for long distance test and in the determination of the riding comfort relative to unsteady vibration. Therefore, in regard to the lateral vibration which is especially significant in riding comfort, another standard shown in Fig. IV-13 is established, based on the distribution of numbers of occurrence of vibration acceleration measured on many rolling stock. In the same figure, the standard of lateral vibration acceleration as viewed from fatigue strength of axle is also shown. As the latter standard has an ample allowance for safety, excess thereover does not necessarily leads to immediate danger. But it may constitute the upper limit of the number of occurrence of vibration acceleration in commercial runs. In regard to the vertical vibration, assuming that the predominant frequency is 2 Hz or less, the full amplitude of acceleration of 0.4 g (or a = 0.2 g) is taken for a limit, as this value is nearly equal to the riding comfort index 2 at 2 Hz or 1 at 1 Hz.

For the determination of the safety against derailment there is no other method than checking the derailment quotient derived from the measured values of side thrust and vertical load at the wheel. The safety can not be determined by the magnitude of acceleration measured on the car body. But in case the track condition is good, the cause of derailment is attributable almost to the hunting, and therefore the safety can be determined by the measurement of vibration acceleration. In this case, in regard to the car body oscillation, the allowable limits have been provisionally determined at 0.4 g in total amplitude in lateral direction and 0.5 g in total amplitude in vertical direction. Properly speaking, these values must be decided after investigating the relation between the vibration acceleration and the side thrust and vertical load at the wheel.

4.4 Classified expression of vibration of rolling stock

In order to statistically obtain the representative value of the vibration against the running speeds by processing the continuous wave form measured over such a long distance as between Tokyo and Osaka, the following method is usually adopted.

After the continuous wave form is divided by small sections of 100 m in running distance, the mean running speed and the maximum amplitude of vibration acceleration for each section is calculated. This is done as to all the sections.

Then the numbers of occurrence of the vibration of the amplitude range of 1 mm and in the speed range of 5 km/h are entered in the form as shown in Fig. IV-14. After the form is filled in with all read out values, the mean value of the amplitude in each speed range is calculated. Mean value multiplied by the sensitivity of the meter will be the mean amplitude of vibration acceleration.

Finally this mean amplitude is plotted against the middle value of each speed range (for example the middle value is 102.5 km/h in the range of 100 – 105 km/h). The plotted points being connected, a characteristic curve of running vibration against the running speed of the rolling stock is obtained.

When such a process is followed, the criterion for the class of vibration of rolling stock is already given as shown in Fig. IV-15. Therefore the characteristic curve as compared with the criterion, shows whether the vibration of this rolling stock belongs to the class A-1, or A-2, B, or C.

For the improvement of its vibration characteristics, a measure must be taken until it comes to the class A. When the vibration characteristic in a certain direction is especially worse than the others, naturally effort must be made so as to improve the vibration in question.

100 km or so is enough for the length of the test section to know the statistical character of this kind of vibration.

4.5 Table of the diagnosis of rolling stock vibration

The relation between the causes of the vibration of rolling stock and the vibration phenomena resulting from it is made clear through many theoretical analysis and through the experiences derived from the tests with actual rolling stock.

Table of correspondence in the typical cases is shown in Table IV-1.

By use of this table causes of vibration are determined by comparing the findings with these vibration characteristics, and countermeasures are found accordingly.

5. Consideration of vibration prevention in designing rolling stock

The discussion so far made is concerned with the measures to improve the symptoms of vibration in case any occurs to rolling stock. However, there are some rolling stock whose performances are not good already at the time of purchase. Improvement in the vibration of the rolling stock of this kind by later adjustment in part are usually almost impossible as it involves too many fundamental problems. In order to get rolling stock of good vibration characteristics, the prevention of vibration must be considered at the designing stage. Items involved in the vibration prevention are very many, and some of them can not be adopted because of the prohibitive condition existing aside from the vibration prevention. On the contrary, some items are very important for the vibration and in some cases they must be adopted in preference to many other conditions. In this paper only such important items will be dealt with.

5.1 Prevention of hunting

(1) Rotational resistance of the truck

For the prevention of hunting it has been found effective to provide an appropriate resistance to the relative rotation of the truck against the car body.

Too large rotational resistance make large side thrust at the wheel on the curved track. An appropriate magnitude of the resisting moment is empirically given by the following formula,

$$W/20 = T \text{ kg-m} \qquad (5)$$

where T is the resisting moment, W is the weight (in kg) of the car body per truck. As this is an experimental formula, the dimensions on both side of the equation are not equal.

Suppose the weight of the car body is partially supported by the side bearer, the ratio of the partial weight on the side bearer is denoted as κ, distance between the side bearers as 2b (meter), and the coefficient of friction on the side bearer as μ, then,

$$T = \mu W \kappa b \qquad (6)$$

Therefore, in combination with equation (5)

$$\mu \kappa b = 0.05 \text{ (meter)} \qquad (7)$$

For example, if $\kappa = 1$, which means that all the weight of the car body is supported on the side bearer, and a material with $\mu = 0.1$ is used, then b = 0.5 m from the equation (7).

In order to give the frictional resistance it is also effective to use a center plate with a large diameter, apart from the method of making the side bearer support part or whole of the weight of the car body.

The rotational resistance T must be a little smaller than W/20 of equation (5) in the case of the rolling stock which have to pass many curves. On the contrary it must be a little larger with the rolling stock running at an especially high speed. So, the allowable range seems to be W/30 – W/10.

(2) Elasticity of the axle support

In order to prevent the secondary hunting which becomes unstable at high speed, it is effective to support the axle on the truck without gaps in both longitudinal and lateral direction and with appropriate elasticity. The optimum value of the supporting elasticity are 2t/mm per axle in longitudinal direction and 1t/mm in lateral direction.

(3) Elasticity for bolster anchor

In order to connect the upper bolster with the truck frame in longitudinal direction, bolster anchors fitted with rubber bushes at both ends are preferred.

The elasticity of the bolster anchor given by the rubber bushes, makes a pair for the right and left sides of the truck, and constitutes a rotational spring which acts in series with the frictional resistance due to side bearer previously mentioned, against the relative rotational movement between the car body and the truck around the center pin. When the relative distance between the anchors on both sides of the truck is 2 – 2.5 m, the optimum value of the elasticity given to each anchor is 500 kg/mm.

5.2 Prevention of vibration

(1) Spring system in vertical direction

The axle and bolster springs had better be soft. The stiffness ratio of bolster spring to that of axle spring must be within a range of 0.5 – 0.6 when the oil damper is attached to the bolster spring, and 1.1 – 1.5 when it is attached to the axle spring. But, when the air spring is used for the bolster spring, the stiffness ratio may be made 0.2 – 0.4. In this case oil damper can be omitted if a throttle is used in the air spring device.

When the coil spring is used as the axle spring or the bolster spring, a rubber pad must be inserted in the spring base. This is effective to isolate the high frequency vibration (surging) of spring and to prevent the car body from shaking.

(2) Supporting system in lateral direction

When the swing hanger mechanism is adopted, the length of the hanger must be as long as possible and the angle of the hanger must be taken at 10 degree or so.

In the case where the air spring is used for bolster spring instead of the swing hanger mechanism and its lateral elasticity is utilized to reduce the lateral vibration, the quotient of the weight on the air spring divided by the spring constant in lateral direction (it is called equivalent lateral deflection) must be about as long as the swing hanger. These have the effect of preventing yawing and rolling.

It is better to adopted the oil damper in lateral direction which acts in parallel with the swing hanger or with the lateral movement of air spring. This oil damper works to prevent the transitional sway at the entrance or the exit of the curve. Moreover it is effective to prevent the primary hunting.

5.3 Prevention of the inclination of the car body

It is accepted that the passengers begin to feel uncomfortable when the inclination of the car body exceeds 0.02 radian by the action of the lateral acceleration 0.1 g at the center of gravity of the car body. Therefore, if possible, it is preferable to restrict the inclination to less than 0.01 rad.

To this end, the distance between the right hand and left hand of the spring or that of oil damper on both sides must be taken as large as possible. It is preferable to locate the bolster spring as high as possible. If the calculated value of the inclination of the car body exceeds the above mentioned limit, torsionbar stabilizer must be adopted. Especially in case that the air spring is used as the bolster spring, it is often necessary to use the stabilizer in parallel.

These methods are effective to prevent not only the excess stationary inclination but also the rolling.

5.4 Prevention of the off loading of the wheel load

The countermeasures to prevent the off loading of the wheel load are contradictory to the prevention methods of car body inclination except the problem of height of the bolster spring. Therefore the magnitude of the restoring force to prevent the excess inclination has not only the lower limit but also the upper limit.

As already described in section 2.4, in the case that the laminated leaf spring is used as bolster spring of the truck, it is especially important to prevent that the friction between the leaves becomes too large.

When the torsional rigidity of the car body is small, the countermeasure to prevent the off loading due to the truck twist is often unnecessary, as the off loading is generally small in this case.

6. Criterion for the speed up seen from the vehicle side

Judgement whether the running speed of the rolling stock is adequate or not or whether the speed up is possible or not must be given considering both of running safety and riding comfort.

In case of passenger cars the restriction of the riding comfort is generally more strict than that of the running safety. The criterion of the riding comfort is given in Fig. IV-12 and Fig. IV-13 as described in section 4.3.

In some cases, as the SHINKANSEN rail cars, allowable speed limit determined by the running safety is lower than that by the riding comfort. In other cases the speed up is required without regarding to the riding comfort. In these cases the allowable speed limit is decided by the running safety.

The most serious problem related to running safety is the derailment. In order to judge the limit of running safety to prevent derailment, it is necessary to check the derailment quotient derived from the measurements of side thrust and wheel load.

Fig. IV-4 in section 2.2 is the criterion for the allowable limit of derailment quotient generally used in our country. However, for the climbing-up derailment ($t > 1/20$ sec), frequency of occurrence being taken into consideration, the values of quotient 0.9 or 1.0 are temporarily allowed corresponding to the occurrence being only one time in every running distance 100 km respectively.

When the measurement of side thrust and wheel load is impossible, the measurement of vibration acceleration on car body will be substituted although the later is imperfect for the check of safety.

The criterion in this case is that the total amplitude of acceleration contained in the car body oscillation must be less than 0.4 g or 0.5 in lateral or vertical direction respectively.

Table IV-1 Diagnosis of vibrations of rolling stock

Cause of vibration		Source of vibration	Characteristics of induced vibration	Countermeasure
Track	Rail alignment is bad	Lateral distortion of rail (irregular alignment)	Rolling in low speed range and yawing in high speed range, accompanied with hunting of wheelset or truck, lateral vibration with large amplitude frequently.	Mending of the rail alignment. Exchange of worn rails.
	Soft foundation, sinking, insufficient ballast, imperfect tamping	Unevenness of the rail surface (irregularity of the longitudinal level)	Irregular vertical vibration	Ballst supplying Tamping, especially making effort to tamp at rail joint parts
		Slanting of rail (*irregularity of* cross level)	Same as the lateral deformation *of the rail*	
		Local sinking of the rail joint part	Comparatively steady state vertical vibration. Sometimes it becomes a large amplitude resonantly. Shorter the rail length is, the effect appears more strongly. Some kind of trucks induce pitching, and at the same time uncomfortable longitudinal vibration with the frequency 4 – 6 Hz occurs in the car body	
	Looseness of rail joint	Shock at the rail joint	Noise at the rail joint becomes louder and the car body shake occurs at every joint. At this time wheel vibratates transiently at 20 – 60 Hz in vertical direction. Several kinds of elastic vibration with high frequency are induced in the car body.	Fastening of fish-plate bolts

Cause of vibration		Source of vibration	Characteristics of induced vibration	Countermeasure
Track	Rail corrugation	Unevenness of the rail surface	Continuous severe car shake and noise are induced. Wheels vibrate at 20 – 60 Hz. On the car body several kinds of elastic vibration are induced mixingly. In some cases they are compariatively steady. They often occur near the station and on the overhead bridge in electrified section. Wave length is 0.30 – 0.80 m on the track of JNR and 0.06 – 0.1 m on street car tracks.	Exchange of rails
	Entrance and exit of curved track (transient curve is not good)	Sudden change of running direction	Sudden lateral vibration occurs at the entrance and exit of the curved track. Sometimes it brings an occasion to begin hunting.	Insert the transient curve or adjustment
Roll-ing stock	Wear of wheel type	Hunting of a wheelset or truck	Severe steady state lateral vibration is induced. It is most uncomfortable vibration. So called crescent-shaped wear gives the worst influence.	Turning of the tread
	Clearance on the part of axle box	Hunting of a wheelset	This is the main cause of the lateral vibration next to the tread wear. Clearance in longitudinal direction specially gives the worst influence.	Taking away the clearance between axle box and box guard
	Eccentricity of wheels	Eccentricity of wheels	Vertical vibration with a frequency near 2 Hz . is induced at the low speed of about 30 km/h and rough vertical and longitudinal vibrations with 7 – 8 Hz. are induced at high speed. They appear continuously.	Turning of the tread (Centering of the car wheel lathe must be checked.)

Cause of vibration		Source of vibration	Characteristics of induced vibration	Countermeasure
Rolling stock	Too stiff spring	Unevenness of rail, sinking of rail joint, shock at rail joint	Vertical vibration and car shake increase generally and the riding comfort becomes worse. In the case that axle spring is too stiff, frequency and amplitude of vertical vibration increase. Also the car body shake increases generally. When the bolster spring is too stiff, damping for the vertical vibrations becomes insufficient, so the vertical vibration continues rather long and sometimes it will be with large amplitude resonantly.	Exchange of the spring with the softer one
	Full stroke of spring (fatigue deformation of spring or overload)	Same as above	Car body shake becomes severe. When the axle spring sticks, shake becomes especially severe. When the bolster spring sticks, vertical vibration can not be damped until it becomes dangerous.	Exchange of the spring
	Insufficient rigidity of car body	Shock at rail joints, rail corrugation and others	Vertical and lateral bending vibrations of car body occur. Wave form is comparatively sinusoidal and regular with a frequency 6 – 12 Hz. We feel shake and are hard to read news-paper.	Increase of the car body rigidity
	Weakness near the attaching part of the upper center plate of car body	Same as above	Very severe shake of car body occurs. Its wave form is rather irregular and stationary frequency is hardly found.	Strengthen the cross beam, especially the member near the upper center plate
	Local weakness of car body	Same as above	Local resonance at the floor, ceiling, side wall and glass of window. Rattling, chattering vibration and noise.	Strengthen locally and take away clearance

Cause of vibration		Source of vibration	Characteristics of induced vibration	Countermeasures
	Bad coupling of gear of main motor	Shock of gear coupling	Large noise. Car body shake and special noise are induced just after departure.	Exchange of the gear
	Imperfect action of shock absorber in draft gear	Longitudinal shock	Severe longitudinal shock is felt at departure, stop and speed change in running.	Exchange of the shock absorber

Fig. Ⅳ-1 Speed limit on curved track

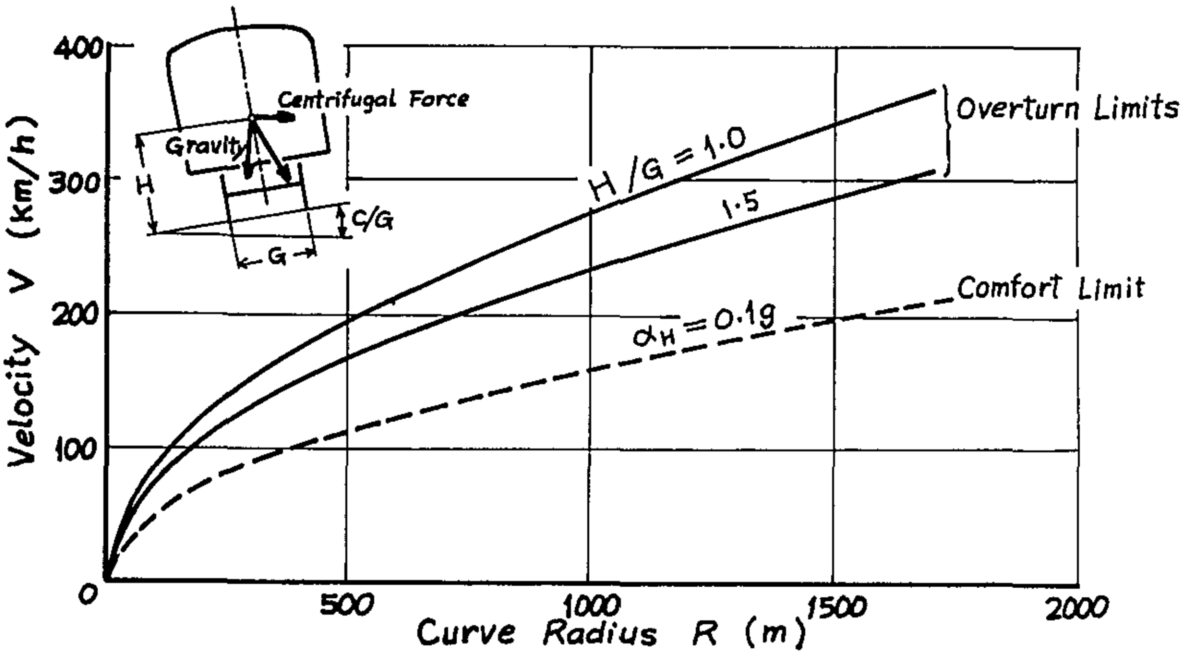


Fig. Ⅳ-2 An example of result of calculation for overturn of a car by lateral wind pressure

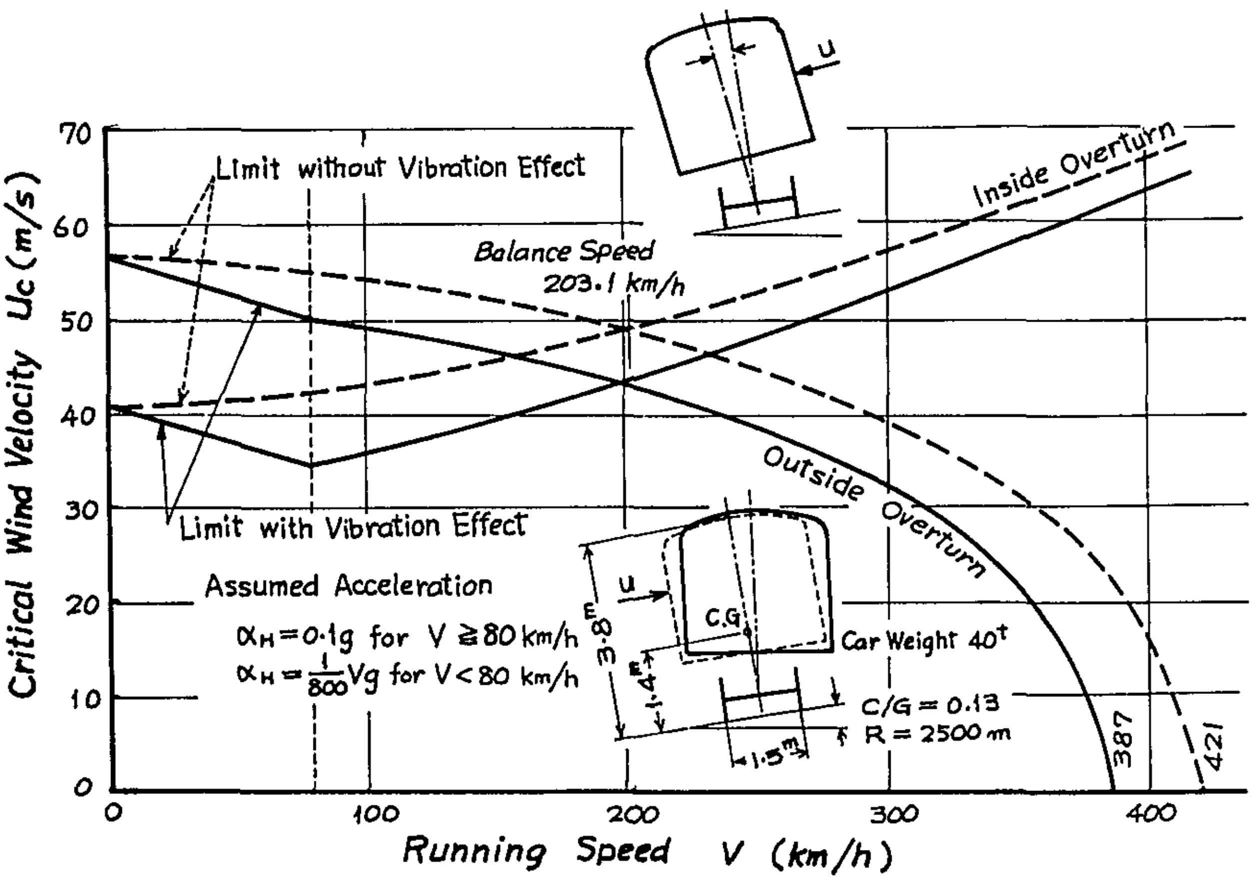
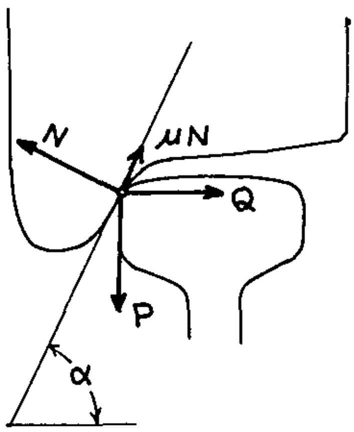


Fig. Ⅵ-3 Condition for derailment



$$\frac{Q}{P} = \frac{\tan\alpha - \mu}{1 + \mu\tan\alpha}$$

Fig. Ⅳ-4 Allowable limit of derailment quotient

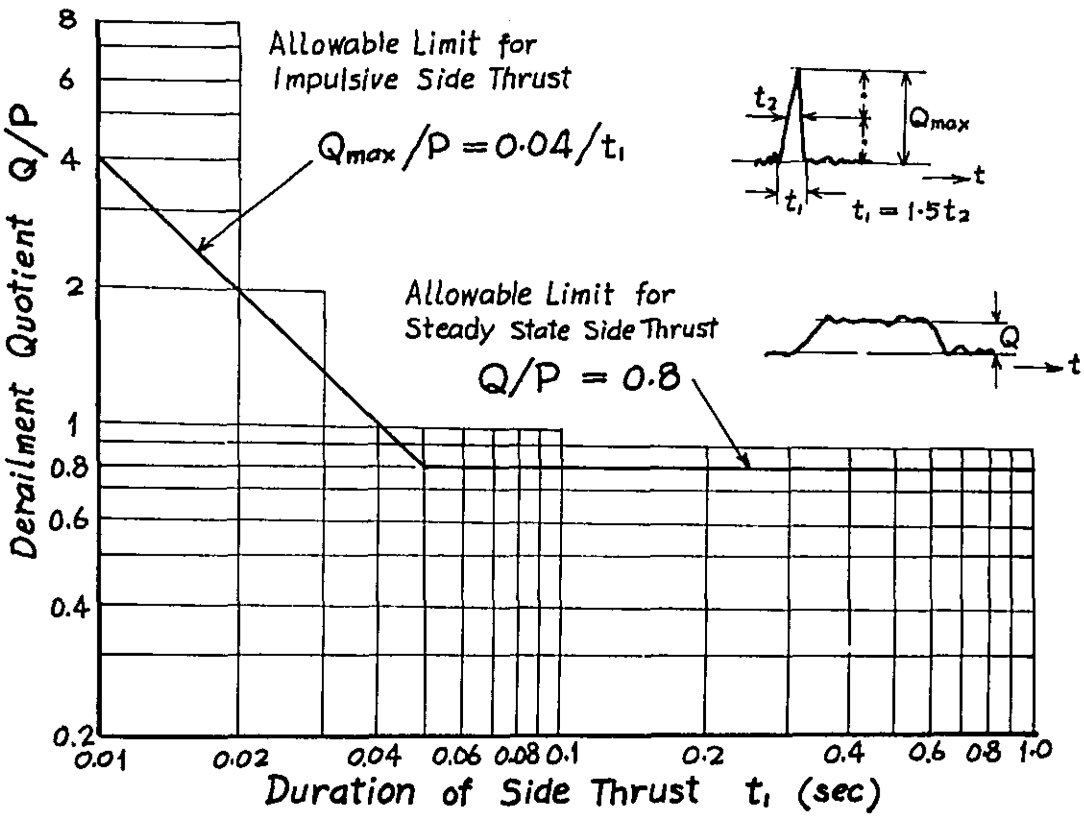


Fig. Ⅳ-5 Typical wave form of side thrust

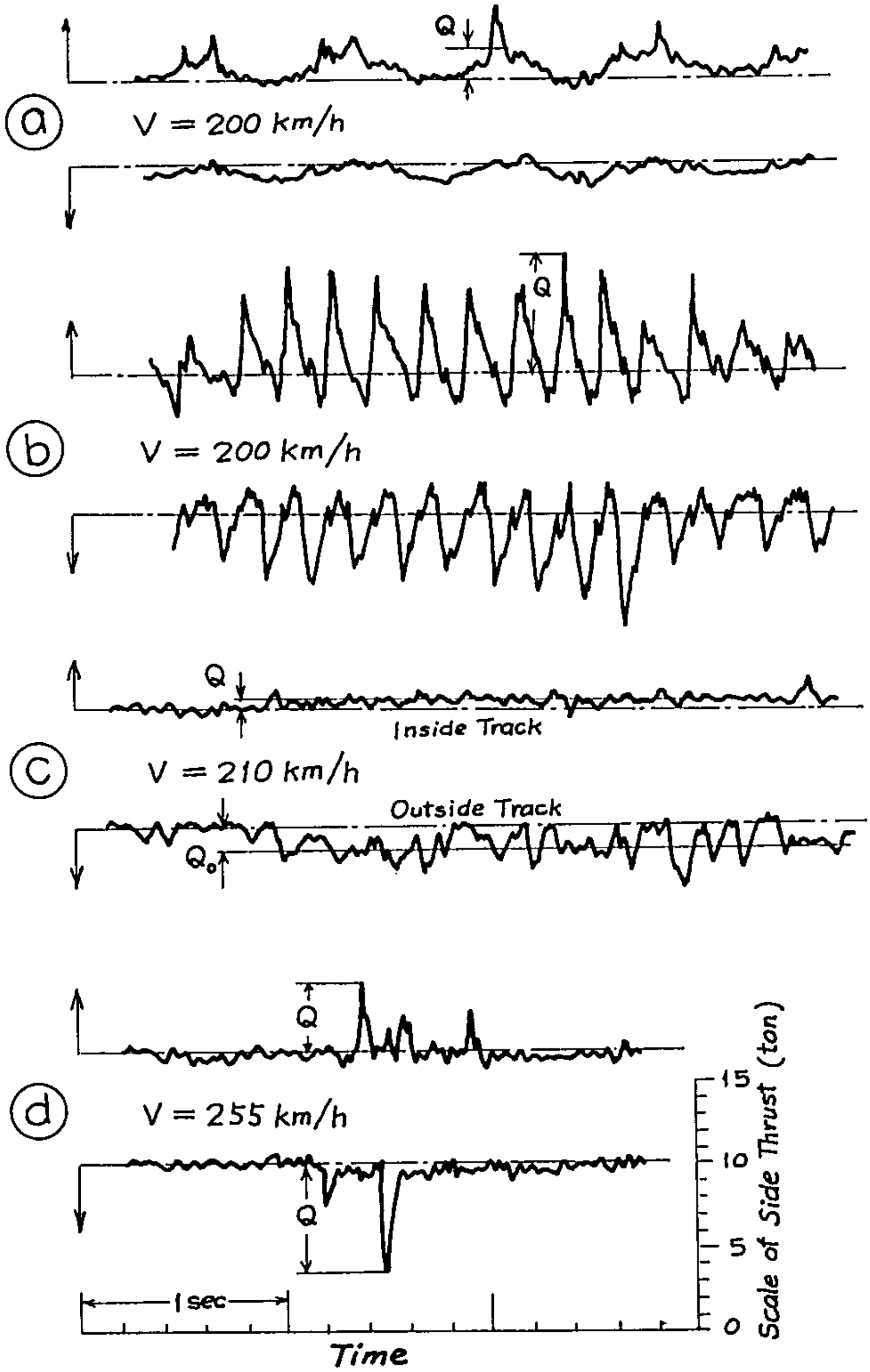


Fig. IV-6 "Buckling" of the train line

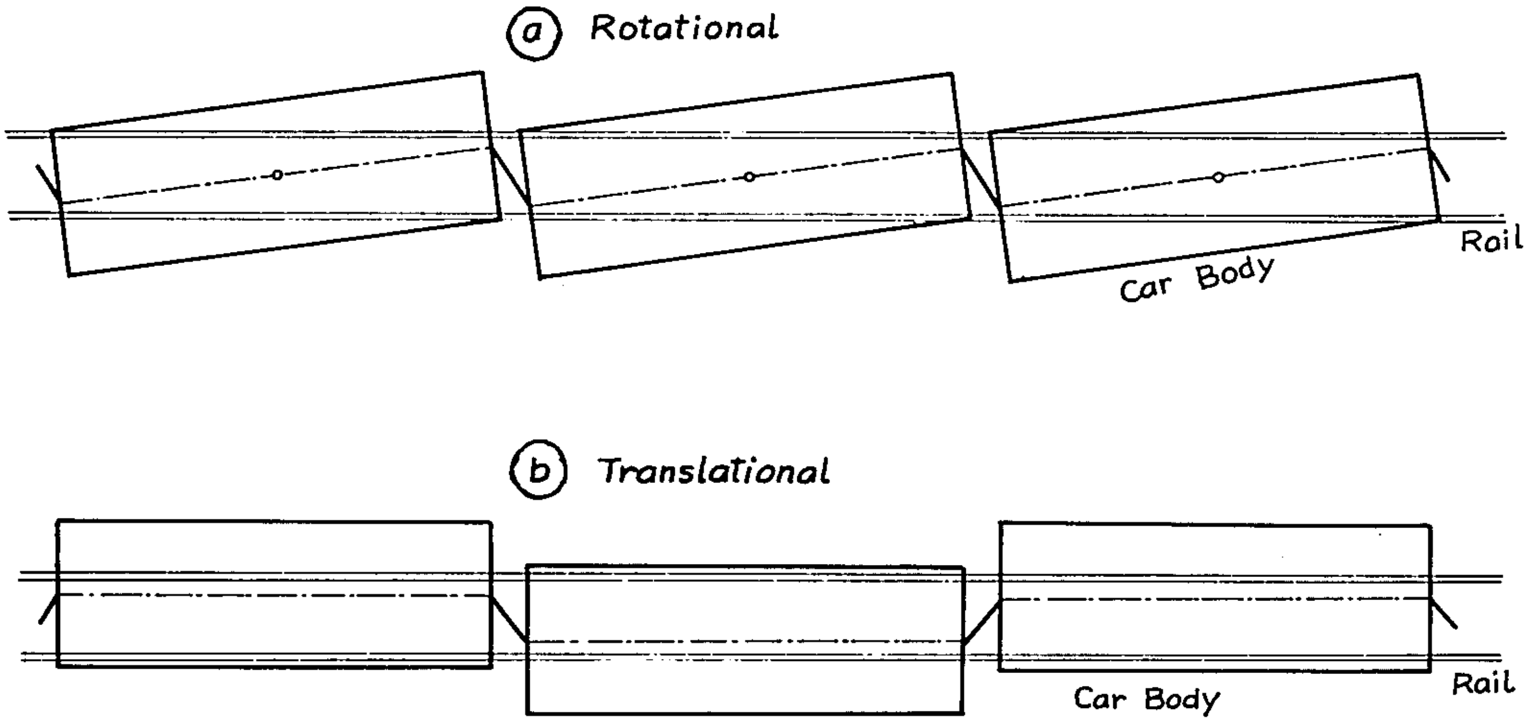


Fig. Ⅳ-7 Off loading of the wheel

due to the twist in the track or the car body

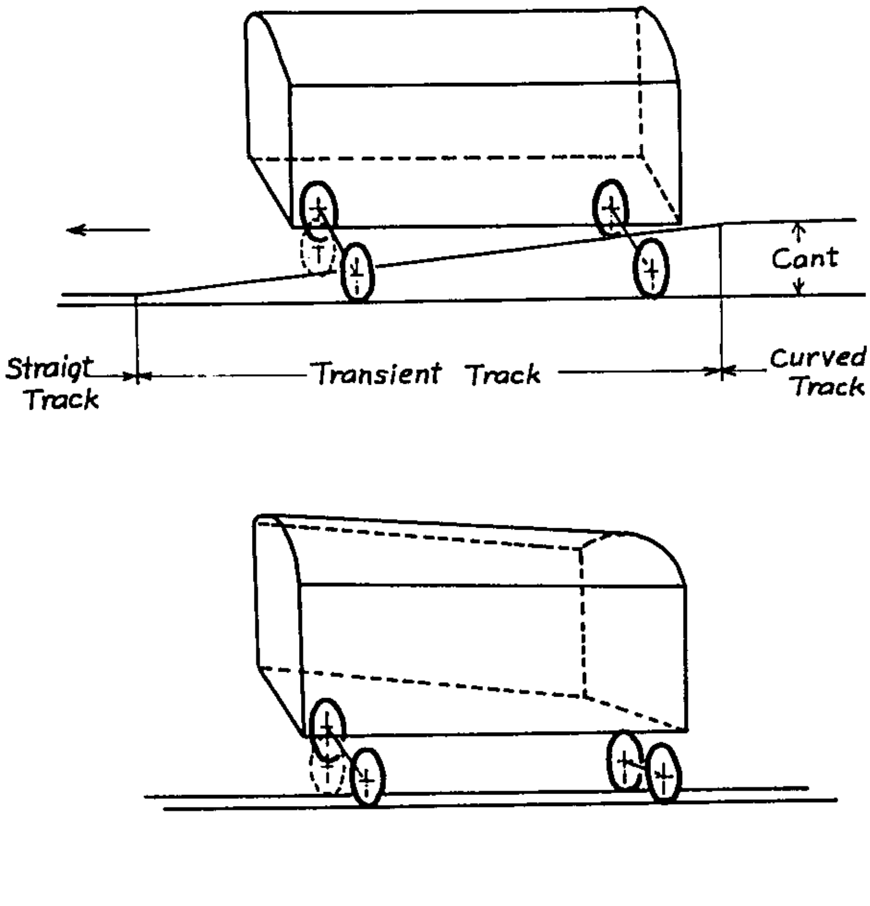
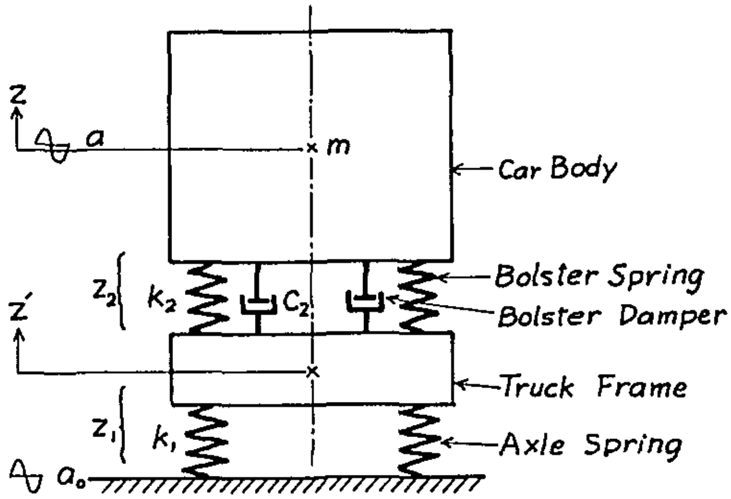
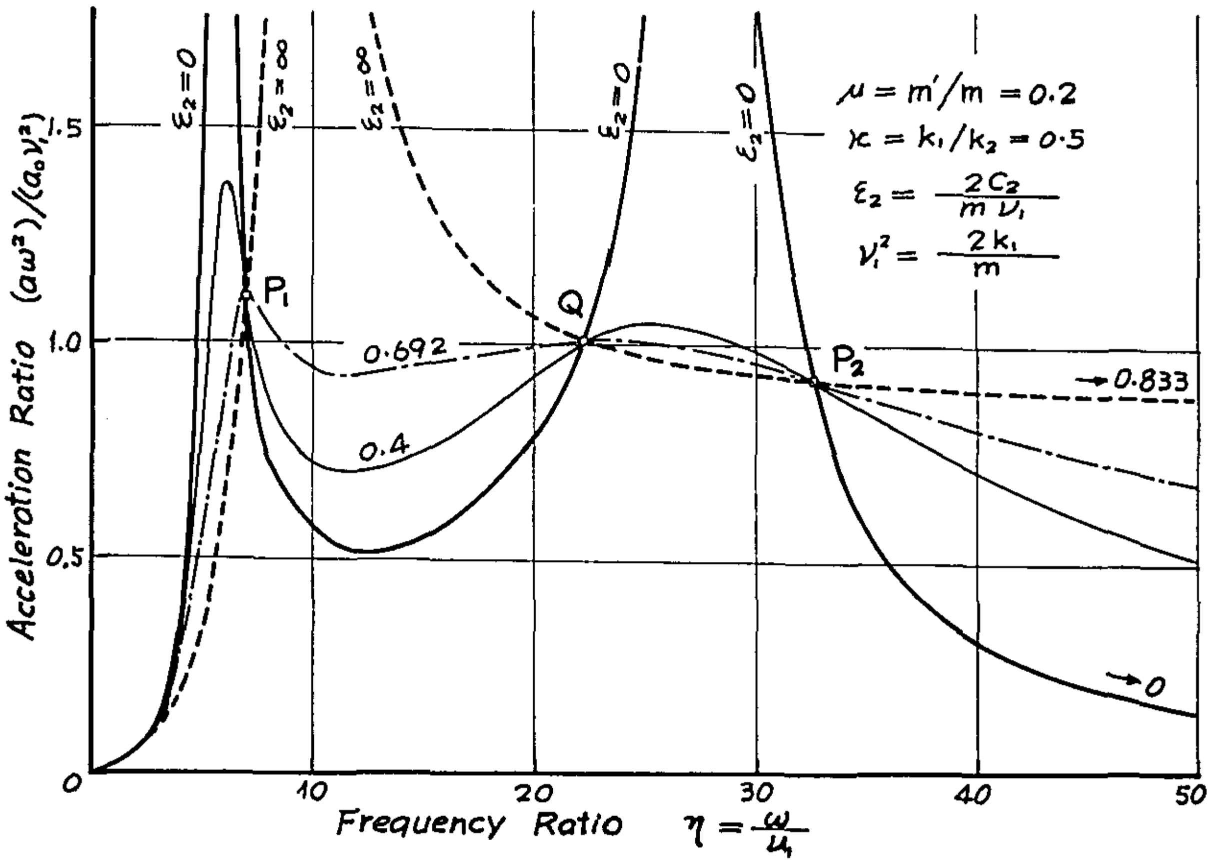


Fig. IV-8 Forced vertical vibration of bogie car



(a) Vibration system



(b) Frequency characteristics

Fig. Ⅳ-9 Hunting motion of a wheelset

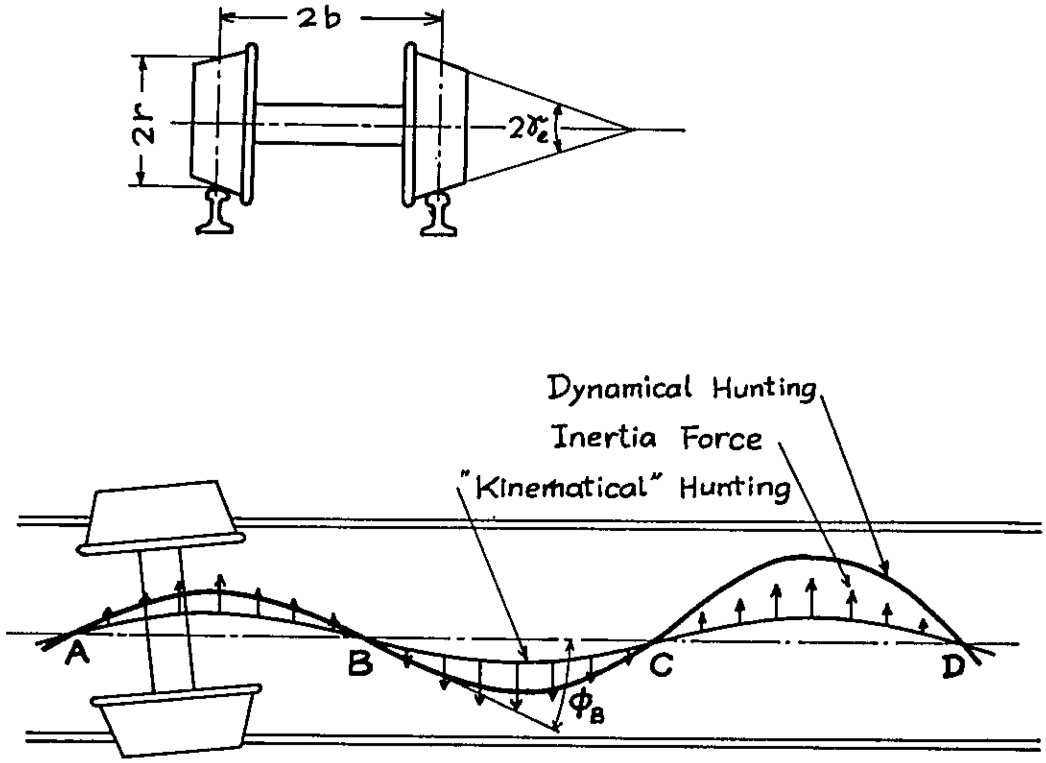


Fig. Ⅳ - 10 Typical wave form of car body vibration

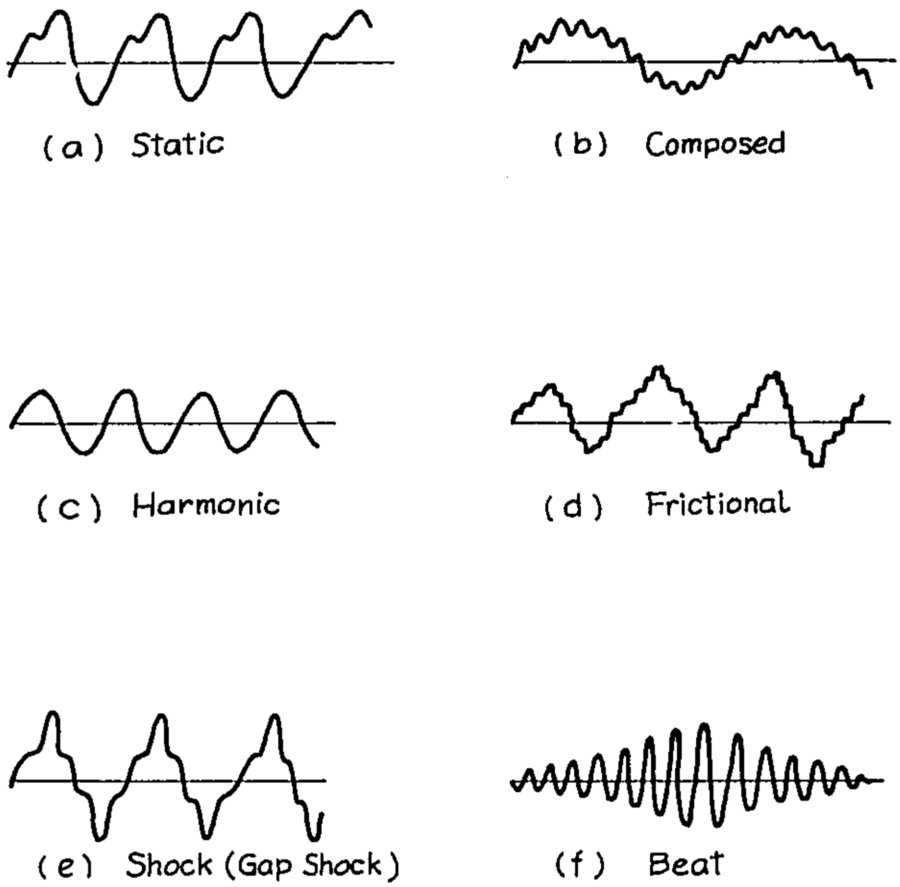


Fig. IV - 11 An example of frequency analysis of a railcar

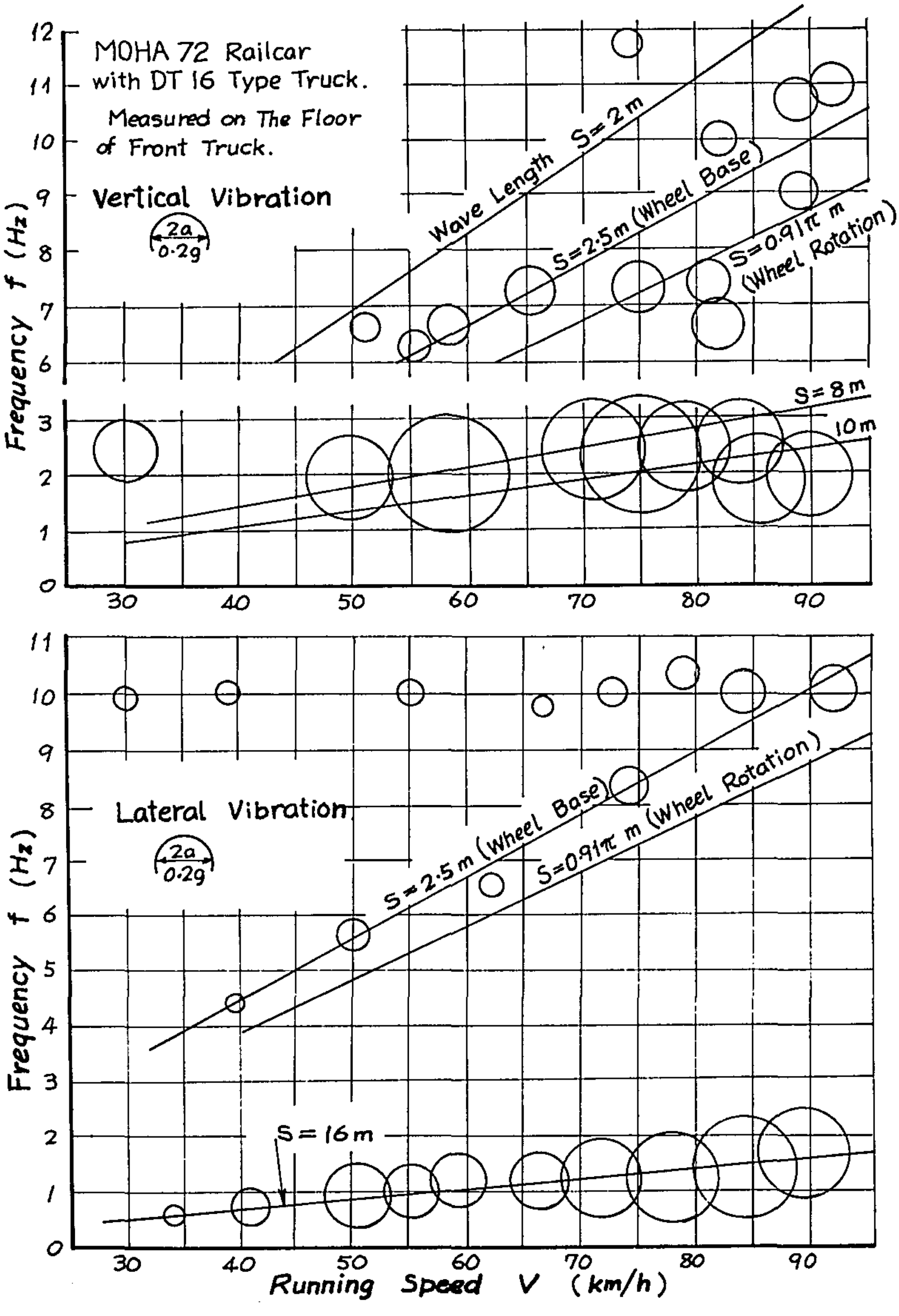


Fig. Ⅳ 12(a) Criteria for riding comfort

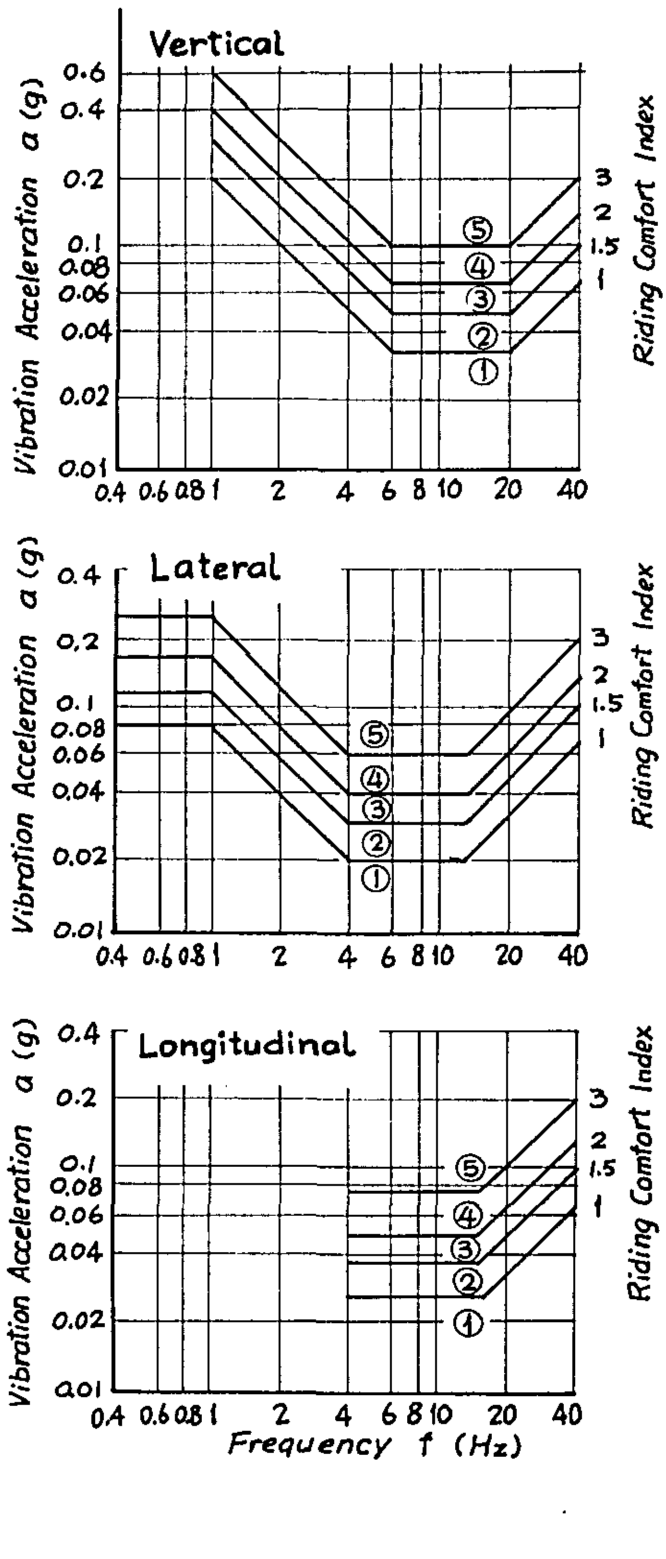


Fig. Ⅳ-12(b) Examples of riding comfort data

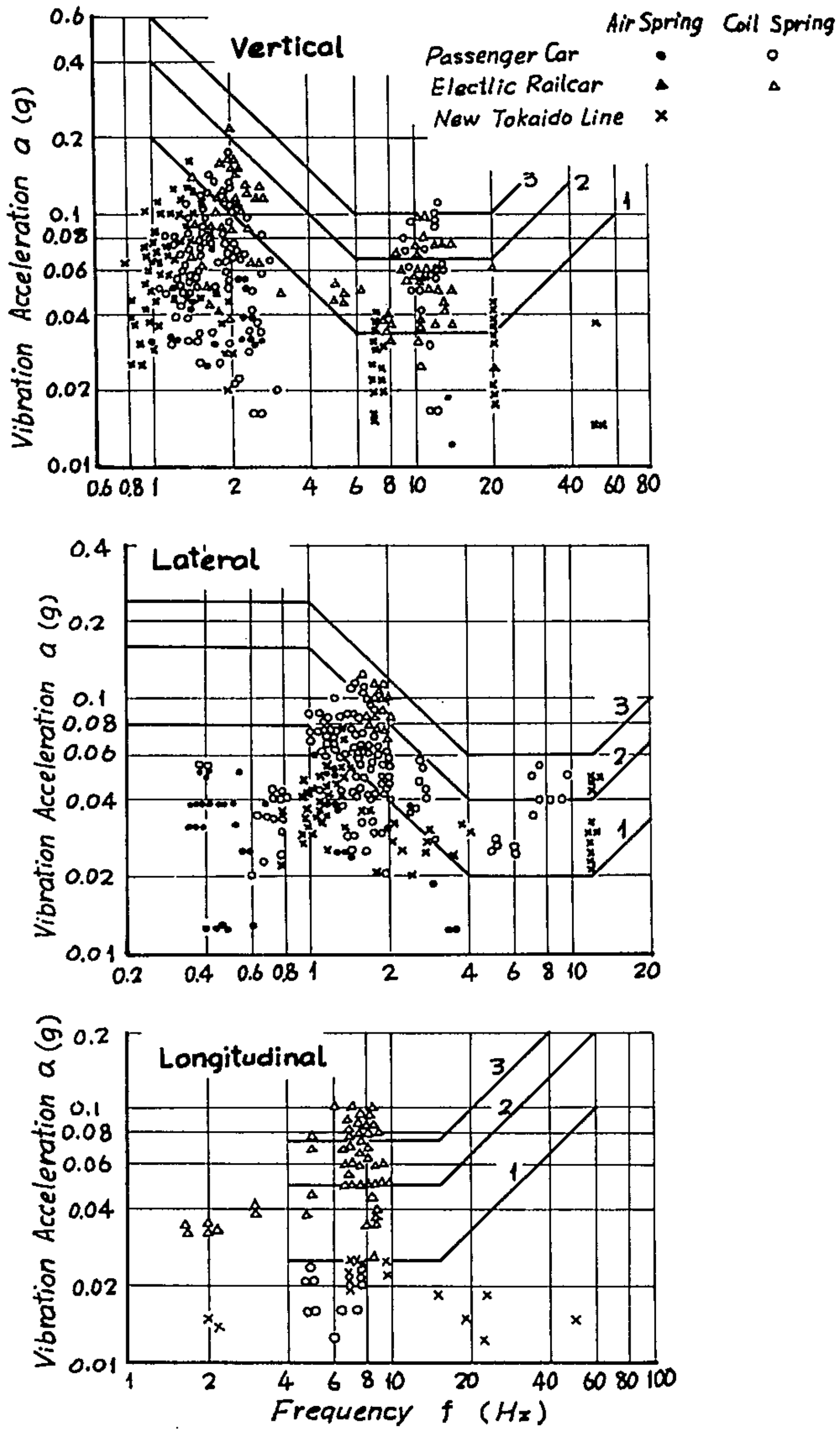


Fig. Ⅳ - 13(a) Criteria for lateral vibration of car booly

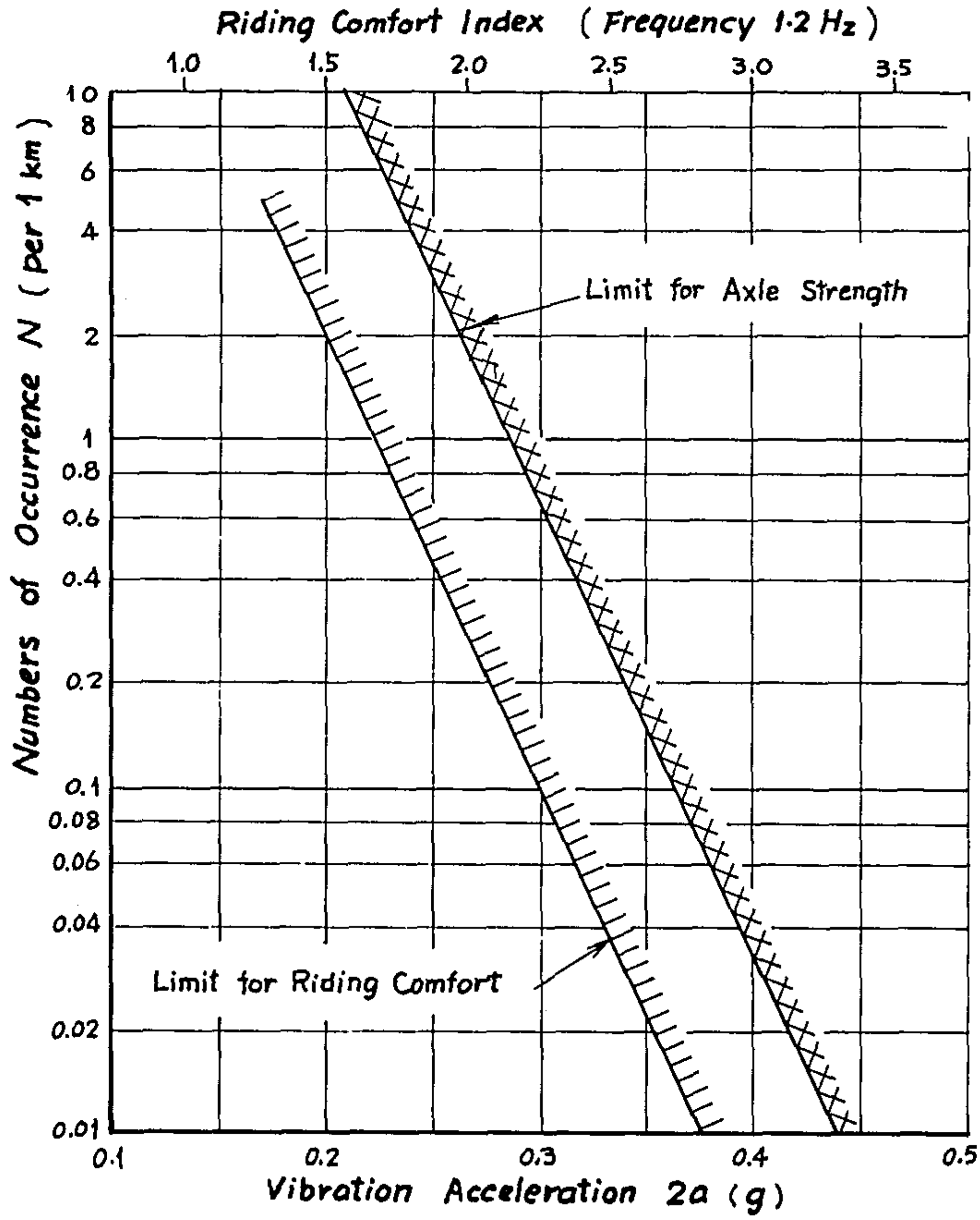


Fig. Ⅳ-13(b) Example of vibration test results

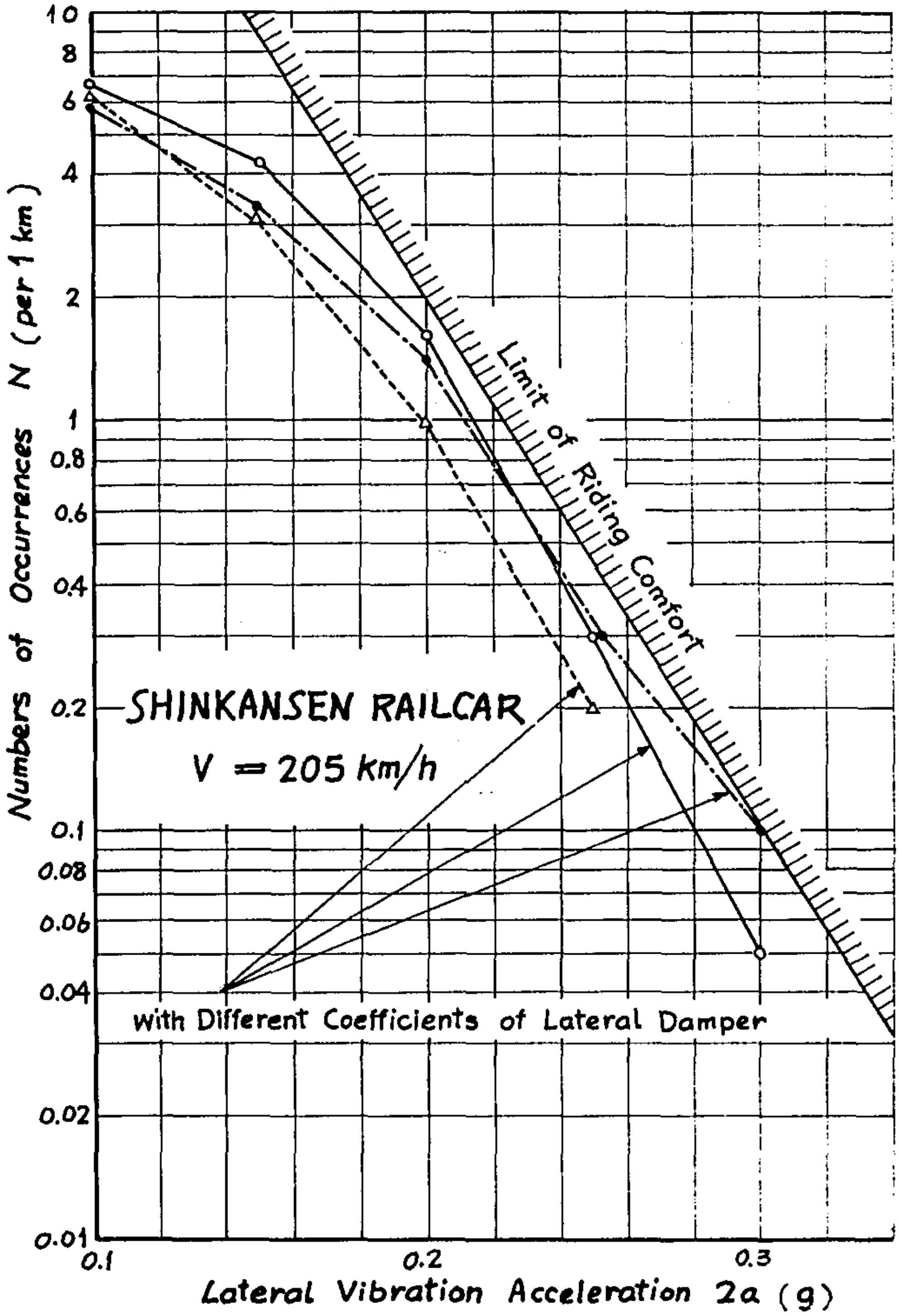


Fig. Ⅳ - 14 An example of the form to be filled with numbers of occurrence of vibration

Running Speed / Amplitude Reading	km/h 0 ~ 5	5 ~ 10	10 ~ 15	15 ~ 20			95 ~ 100	100 ~ 105
0 ~ 1 mm	IIIII	IIIIII	IIII	II				
1 ~ 2	II	IIII	IIIIII	III	II			
2 ~ 3			IIII	IIIII				
10 ~ 11							III	I
Mean Value	mm							

Fig. Ⅳ-15(a) Criteria for car bady vibrations

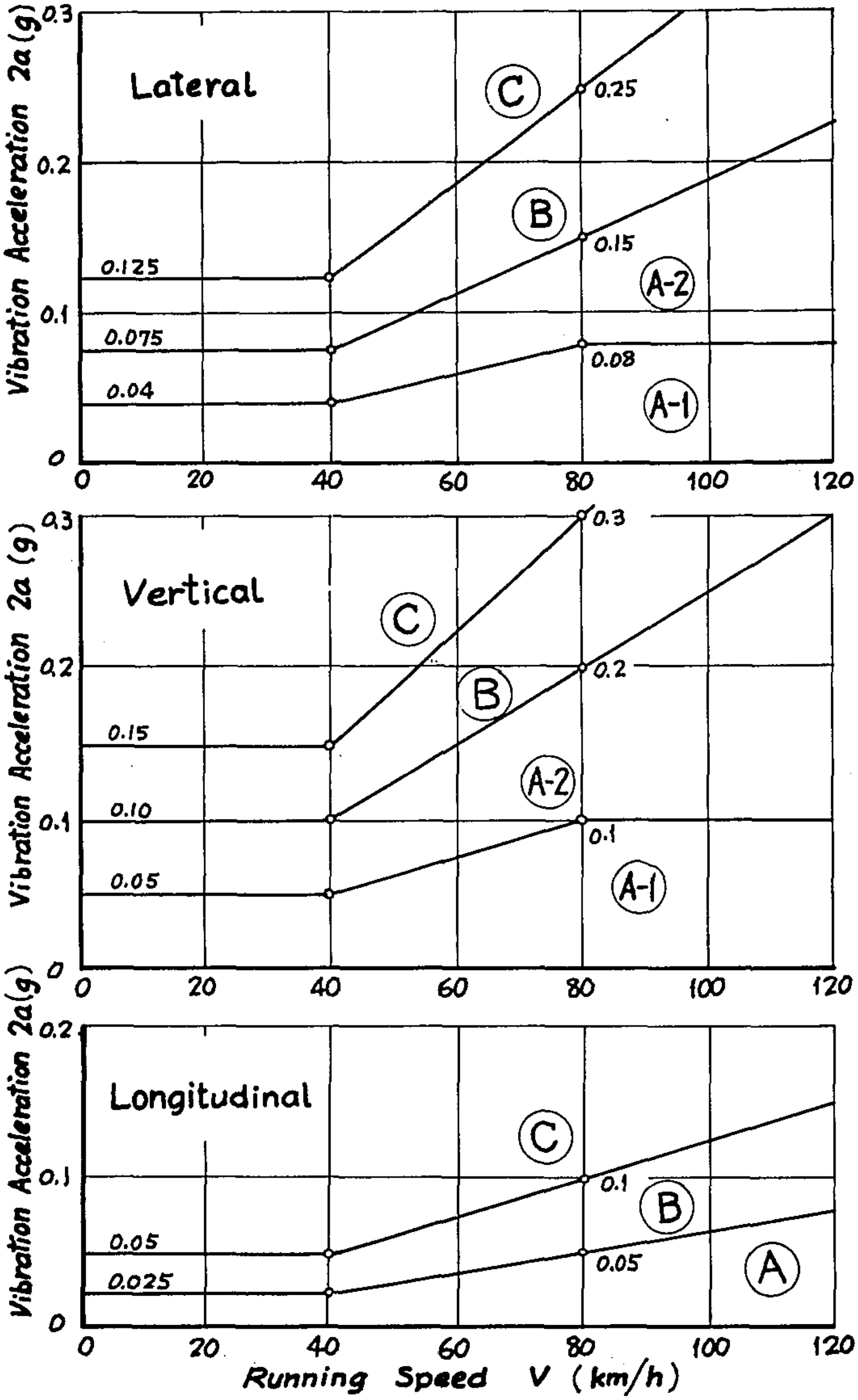


Fig. IV - 15(b) Examples of vibration test results