DOT-TSC-FRA-80-1

FOREIGN RAIL TECHNOLOGY APPLICATION STUDY

TRIP REPORT

By

G. Kaess and L. Fendrich

Contract DOT-TSC-1554

May 1979

FOR INTERNAL USE

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Translated by:
Walter Grant
Raytheon Service Company

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UNIT CONVERSION TABLE

Length	1 mile = 1.6 km	1 inch = 25.4 mm
	1 km = 0.62 miles	1 mm = 0.039 inch
Temperature	deg. F = 1.8 x deg. C + 32	
Curvature	deg./100 ft = 1746/R (in meters)	
Pressure	1 N/mm^2 = 145 psi	

ABSTRACT

The problems treated in this report refer to

- acceptance conditions for locomotives and cars
- safety, construction, and maintenance standards for rails and switches
- buckling safety of continuously welded rails and its influential factors
- track maintenance and inspection methods

of the German Federal Railway.

Furthermore, a series of other particular questions is addressed. These questions were posed to the authors by members of the Safety Track Program during the course of their stay. Within the individual sections, an attempt was made to present, explain, and justify the standards, regulations, and methods used by the German Federal Railway. The authors endeavored to present their experience and knowledge, which is of interest for the Safety Standards Performance Program.

Table of Contents

Index of Figures

Appendices

3.1 Article: "Track stability of the track at high speeds" by J. Eisenmann.

3.2 Description of an experiment to determine the equivalent moment of inertia of a track panel

4.1 Delivery conditions for rails

4.2 Delivery conditions for concrete ties

4.3 Delivery conditions for ballast

4.4 Drawing of the concrete tie B70 W-54

Figure References

Figures 2.5, 2.8, 2.9, and 4.6, after "The rail", Editor Fritz Fastenrath, 1977, Publisher: Wilhelm Ernst and Sons, Berlin, Munich, Düsseldorf (to be translated by Frederick Ungar Publishing Company, Inc., New York, New York).

Figures 2.2, 2.3, and 2.4, after "Research work on railroad tracks and street surfaces", Editor J. Eisenmann, 1974, Publisher: Wilhelm Ernst and Sons, Berlin, Munich, Düsseldorf.

Figure 1.1 after the ORE Report B55 RP 6

Preface

The present report was generated during the nine weeks' research which the authors performed within the framework of the project of the TSC on rail safety standards.

During the initial phase, the authors studied previously obtained research results, especially with respect to the interactions between the vehicle and the rail. They thus acquired an overview of prospective further procedures.

During the second phase, the authors held extensive discussions concerning the principal problem areas of the project. The object of these discussions was to define the question to which the authors could make a suitable contribution, as staff members of a European railway. As a result, an abundance of individual problems was listed. These problems have been classified under the main points of the present report. The spirit common to nearly all the special problems is the desire to improve existing track safety standards by including the interactions between the vehicle and the rail.

During the third phase, the authors held more profound discussions concerning particular problems. The objective of these discussions was to bring to bear on the project, in their full measure, previous knowledge and the totality of experience of the German Federal Railway. The present report reflects these efforts.

The authors are eager to thank the TSC, and especially Mr. David Reed, for the support they have provided. They feel specially obliged to Messrs. McConnell, Smith, Ehrenbeck and Murphy.

The problems treated in this report refer to the following:

- the interactions between the wheel and the rail, inasmuch as they are reflected in track standards and acceptance conditions for locomotives and cars,
- appropriate safety, construction, and maintenance standards for switches,
- the track lateral strength and the lateral buckling safety of a continuously welded rail,
- methods for track maintenance and inspection, the costs associated therewith, as well as the SAFE Test Program.

Within the individual sections, an attempt was made to present, explain, and justify the standards, regulations, and methods used by the German Federal Railway. The authors tried to be careful to indicate especially the experiences and the information of the German Federal Railway which are of interest for the Safety Standard Performance Program. The authors hope that they have thus initiated the desired transfer effect.

In order to achieve a better general understanding of conditions prevailing at the German Federal Railway, a few important characteristics will be explicitly mentioned below:

- The German Federal Railway is a state enterprise. It practically has a monopoly on rail travel.
- The owner - the FRG - has imposed and legally promulgated relatively stringent constraints with respect to operating safety and technical specifications.
- The line network of the German Federal Railway is very dense and has predominantly two-track lines.
- The line network has a high operating load. The ratio of passenger trains to freight trains is 60% to 40%, relative to the number of trains.
- Supervision and maintenance of the line network is performed by the German Federal Railway under its own responsibility.
- Studies in the area of tracks primarily serve to reduce maintenance costs in the long term and to raise the speed.
- The German Federal Railway regards the construction and maintenance of tracks as one unit. Technical-constructive developments are therefore always checked for their possible effects on track maintenance and on the lifetime of track materials.
- The German Federal Railway engages in intensive planning for all track work, in order to guarantee a cost-favorable execution, which is coordinated with operations.

1. Rail and vehicle standards

1.1 Introduction

The development, rationale, and handling of specific standards of the German Federal Railway, with reference to track geometry and to vehicles, must always be viewed in relationship to the general presuppositions which guide track maintenance. By way of introduction, these will be briefly indicated.

The lines of the German Federal Railway are devoted almost exclusively to passenger traffic and freight traffic. Among other things, this means the following: On the one hand, the desired track quality is defined with a view to the comfort of passenger trains; on the other hand, track quality is primarily impaired by freight train traffic with its higher axle load. For this reason, the ratio of freight train loads to passenger train loads on the individual lines, the bandwidth of speeds for passenger trains (140/160/200 km/h) and of freight trains (80/100/120 km/h), as well as the axle loads of locomotives and freight trains (at this time up to 22.5 Mp) are especially significant as influential parameters.

As regards vehicles, it should also be stressed that the overwhelming majority of passenger trains and freight cars of the individual European railways travel on the entire European railway network. Consequently, the existing standards are essentially held in common.

In order to evaluate the condition of the track, three measurement systems are currently being used by the German Federal Railway:

- Measurement of track geometry by means of track inspection cars on all lines,
- measurement of accelerations and lateral suspension forces in a passenger car in order to monitor important lines used by high-speed passenger trains,
- measurement of accelerations and wheel forces with locomotives and passenger cars, to monitor lines for passenger trains with a speed of 200 km/h.

Reactions in the vehicle and on the rail are considered normative for evaluating the track quality. The following are considered as evaluation criteria:

- accelerations on the vehicle body for traveling comfort,
- forces between the wheel and the rail for safety, component stress, and wear, as well as
- forces between the tie and ballast, for the generation of track quality defects.

The essential evaluation standards and guide values, which are currently used by the German Federal Railway for maintenance measures, refer to the following:

- comfort rating figures,
- derailing safety,
- safety against track displacement and

- experience concerning the development of vertical and horizontal track quality defects.

These points will be discussed individually.

1.2 Geometry and other standards of the German Federal Railway

The track standards of the German Federal Railway do not differ strictly according to safety, comfort, and maintenance tolerances, since very close interactions exist in this regard, and since these interactions are not completely known in detail. It must at first be accepted as a fact that some of the standards have already been used for decades and that their validity has been confirmed in practice. Other standards, on the other hand, have been worked out and introduced only recently; these essentially refer to speed ranges above 140 km/h.

An attempt will be made below to assign the standards introduced by the German Federal Railway to the areas of safety, comfort, and track maintenance:

1.2.1 Safety tolerances

Warping

The essential track geometry tolerance relevant to safety is warping. The law prescribes that a value of 1:400 be maintained; i.e. maintenance measures must be taken if manual measurements or an inspection vehicle ascertains that this tolerance has been exceeded. A time period for removing the defect has not been prescribed, since transgression of this value does not yet lead directly to derailment.

Reasons for a warping tolerance

When running over a track warp, the vertical wheel forces are necessarily displaced. The possibility of derailment exists in the case where the wheel whose vertical wheel force is reduced by the track warp must simultaneously accept a horizontal guide force. This case can occur at track defects in tangent sections and particularly during outward runs from curves with super-elevation. A questionnaire submitted to the administrations of the SNCF (French Railway), DB (German Federal Railway), NS (Dutch Railway), OBB (Austrian Railway), and PKP (Polish Railway) determined that the existing warps extended down to the limit warp of 10 0/00 in dependence on the base 2a; 2a here is the wheel-base with a 2-axle vehicle or the distance between pins for double-truck equipment. (ORE Reports B55/RP5 and RP6.)

In the case of vehicles without truck, the limit warp is as follows:

$$g_{lim} = (20/2a + 3) \leq 10 \ [0/00]$$

In the case of vehicles with trucks, the following hold:

formula 1: $g_{lim} = (20/2a + 2) \leq 10 \ [0/00]$

formula 2: $g_{lim} = (20/2a + 3) \leq 7 \ [0/00]$

In the case of vehicles with truck, the warps are superposed. Track warps for the distance between pins and for the truck wheel base must always be determined according to both formulas (1 and 2) and must be superposed.

For testing the torsion characteristics and consequently the derailing safety of a vehicle, that formula is always decisive which poses the more severe requirements to the vehicle.

Figure 1.1 shows the critical limit warping in dependence on the value 2a.

In practice, these critical limit tolerances must be used to guarantee an adequate safety margin. For this purpose, the Federal railway has let stand the long proven tolerance 1:400 (g_{lim} = 2.5 per mille).

The safety margin is necessary in order to equalize

- measurement errors and the influence of different measurement bases,
- different axle loads and axle distances of the car stock
- the temporal development of track quality impairment, and
- time until removal of the defect.

<u>Track gage</u>

The minimum gage - 1430 mm - as well as the maximum gage - 1460 or respectively 1465, as well as the wheel set dimension and its tolerances, are prescribed by law.

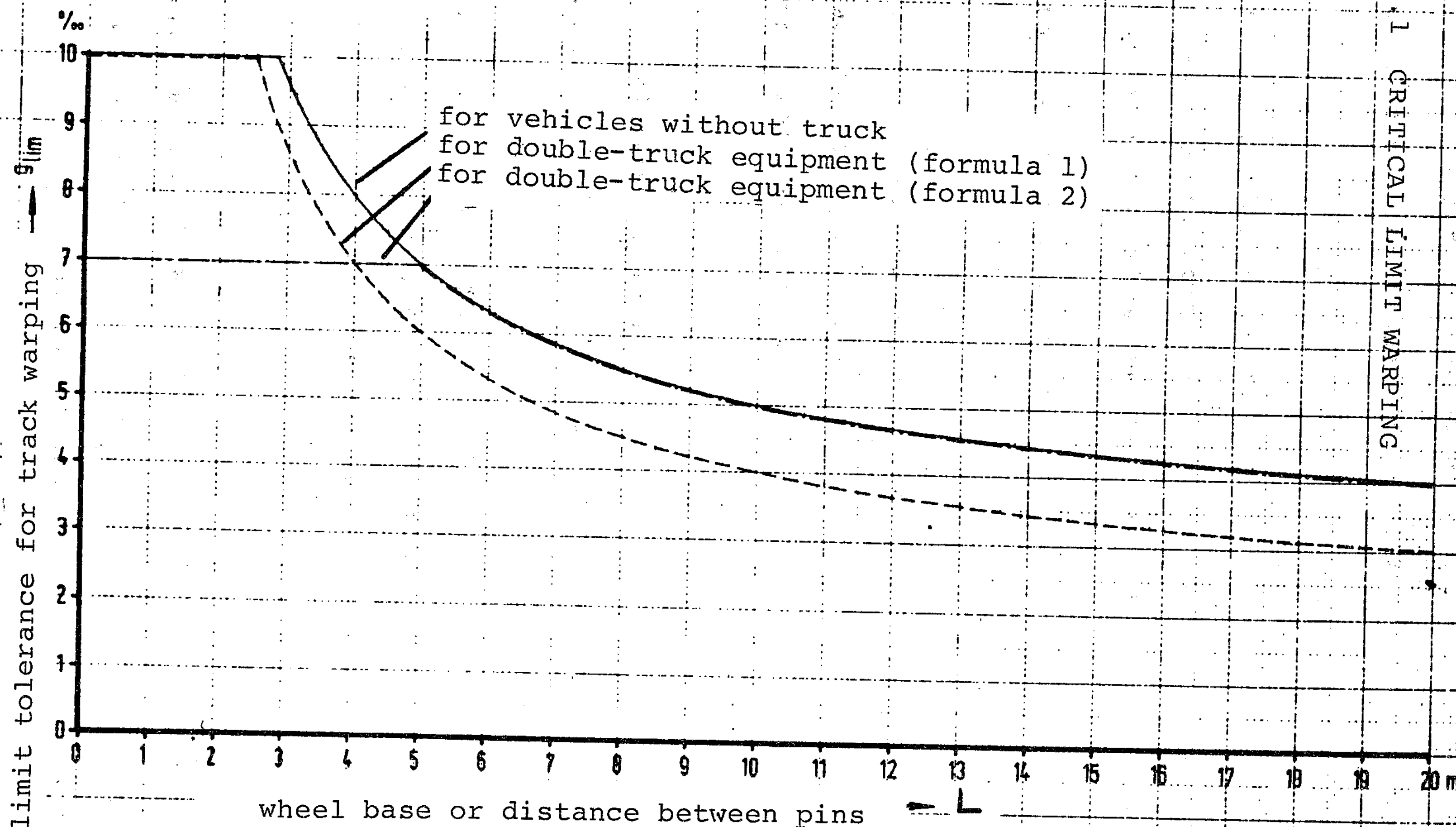


FIGURE 1.1 CRITICAL LIMIT WARPING

Bild 1.1

Reasons for the gage

The minimum gage is prescribed for reasons of technical uniformity for international European traffic. Maintenance of the maximum dimension prevents the wheel set from falling through (trivial case). The minimum dimension serves two functions: on the one hand, it guarantees a vibration-free run, and, on the other hand, it favors a sinusoidal run of the wheel set; in this way, it is possible to achieve uniform wear of the wheel and rail profile.

Alignment Defects

The maximum permissible theoretical track alignment defect is 25 mm. This defect is used as a basis for calculating track stability. It is in the first instance independent of the measurement base. In practice, track alignment is evaluated, not according to the maximum track alignment defect, but according to differences in offset, i.e. there may be no sudden discontinuities in alignment. The permissible offset difference is limited according to the permissible line speed limit.

Reasons for alignment defects

From theoretical and practical studies, it is known (see references on point 3) that discontinuities in track alignment may be a cause for track buckling in continuously welded rails.

Horizontal forces between wheel and rail

The SNCF has determined a limit tolerance with respect to the track stability, under the effect of horizontal and vertical forces from operating loads. The German Federal Railway has taken over this limit tolerance.

Accordingly, the sum of horizontal forces Y, measured at one axle, may not exceed the following value:

$$\Sigma Y \leq 0.85 \ (10 + 2Q/3) \quad [kN]$$

Here, 2 Q (kN) is the static vertical axle force.

In order to determine the relevant forces ΣY, the German Federal Railway uses measurement wheel sets, both for locomotives and for cars. The relevant value ΣY is defined as that force magnitude, which has an effective length of at least 2m, independent of speed.

Reasons for the horizontal forces

For the reasons, reference is made to the explanations under Point 3.

Ratio of horizontal to vertical force Y/Q

For the wheel profile customary with the German Federal Railway, with a wheel flange angle of 70 degrees, the derailment criterion is as follows:

$$(Y/Q)_{lim} \leq 1.2$$

This means that derailment will not occur under the quasistatic action of the forces Y/Q, if the above value of the force ratio is maintained.

With a wheel flange angle of 60 degrees, the value $(Y/Q)_{lim}$ falls to about 0.85.

The derailment limit of a wheel has always been reached when the contact point of the wheel profile and the rail begins to make the transition from the tapered region of the wheel flange into the region of the head rounding.

Reasons for the ratio Y/Q

The above limit tolerance was determined on the basis of extensive studies of the committee ORE B55. It is based on warping conditions prevalent in Europe. Experience shows that warp derailments occur exclusively at very low running speeds. Consequently, the committee has limited itself to the quasistatic curvature-conditioned guiding forces, and has disregarded the dynamic components of the guiding forces. Confirmation of this procedure was provided by dynamic experiments performed within a research framework. These experiments are reported within the framework of the ORE Committee B 55, RP4 and RP6.

1.2.2 Comfort Tolerances

The riding comfort for passenger cars, which is desired by the German Federal Railway, is specified by

- line layout
- lateral acceleration data,
- riding quality figures for cars.

Superelevation

The maximum superelevation on a curved track is 150 mm. 5.85" Special regulations hold for tracks in local traffic and for station tracks.

Reasons for superelevation

This quantity was specified because of the high negative lateral acceleration towards the low rail in tight curves during slow travel (0.89 m/sec^2). This quantity is based on law.

Larger superelevations (180 mm) have been used experimentally. However, because of the predominantly mixed traffic load prevailing with the German Federal Railway, these superelevations have not proven themselves. With slower runs, the low rail is stressed too severely, especially by freight traffic.

Deficiency and excess of superelevation

The German Federal Railway permits a maximum superelevation deficiency (for passenger trains) of 130 mm. 5.07" The maximum superelevation excess (for freight trains) is 70 mm. 2.77"

The applicable excesses and deficiencies of superelevation have been specified in dependence on the proportion and actual speed of passenger and freight trains.

Reasons for the amount of deficiency and excess superelevation

The forces exerted on the curved track by slow and fast trains should agree as much as possible as regards their sum (forces directed to the high rail and to the low rail).

The present regulation is the result of long years of experience. With respect to maintenance costs for curved tracks, which are travelled by freight trains and passenger trains, it represents a favorable solution.

Lateral acceleration tolerances

The German Federal Railway distinguishes comfort tolerances from a quasistatic and dynamic point of view (Figure 1.2).

The quasistatic acceleration tolerance, which occurs only on curves, is 1.3 m/sec^2. The dynamic tolerance is 2.5 m/sec^2. In curved tracks, the quasistatic and dynamic tolerances are superposed; this absolute acceleration is limited to 3.5 m/sec^2.

Reasons for the lateral acceleration tolerances

The quasistatic tolerance of 1.3 m/sec^2 is obtained from the permissible superelevation deficiency of 130 mm, corresponding to 0.85 m/sec^2, and the roll angle of the car body, which results from centrifugal force on curves.

The dynamic tolerance of 2.5 m/sec^2 is based on the determination of the lateral suspension force, which was determined earlier, simultaneously with the acceleration measurement.

At that time, it was assumed that the lateral suspension force has a linear correlation with the sum of the horizontal wheel forces (Sum Y). Since the wheel forces are limited by the tolerance for track stability (in this case 39.8 KN for the measurement vehicle), the lateral suspension force was limited to 70 percent of this value - i.e. 28 KN. With this lateral suspension force, a lateral acceleration of 2.5 m/sec^2 appeared, and an absolute acceleration of 3.5 m/sec^2 on curves.

In the meantime, it was recognized that measurement of the lateral suspension force provides little information as a measure of the horizontal stress on the track. The dynamic and absolute acceleration tolerances derived therefrom, however, have proven quite suitable with reference to riding comfort (dining cars). For this reason, they were retained.

Wz*-values

The comfort of passenger cars and the riding quality of freight trains are evaluated according to a method developed by "Sperling". Sperling studied the vibration sensitivity of human subjects. He determined a regular relationship between frequency, amplitude, and action time. For a given frequency, his basic formula is as follows:

$$Wz = 0.896 \times \sqrt[10]{a^3/f} \quad \text{and}$$

*Wz = Wertziffer = quality factor

$$Wz = 0.896 \times \sqrt[10]{a^3/f} \times F(t)$$

here, a = peak acceleration in cm/sec^2

f = vibration frequency in Hertz

F(t) is a valuation factor which depends on frequency and on the direction of the vibrations, and which takes into account the sensitivity of the human body with respect to mechanical vibrations.

The accelerations measured in the vehicle body all have a frequency spectrum. Consequently, it is necessary to subdivide this spectrum into frequency bands and to integrate. Automatic devices were developed for this evaluation.

A value of Wz = 1 corresponds to an excellent riding quality or just barely perceptible vibrations. A Wz value of 3.0 to 3.25 represents a limit value for cars of a passenger train (acceleration b = approximately 1.0 m/sec^2). A Wz value of 4 - 4.25 represents a limit for freight trains (acceleration b = approximately 1.6 m/sec^2). A Wz value of 5 implies an operationally dangerous condition.

The SNCF has developed a similar procedure, which is based on the peak time.

Furthermore, the ISO has developed a procedure which has general application for vehicles. This procedure, too, leads to a fatigue time; it distinguishes three stages:

1. Reduced working capability, given by a family of curves, which represent the acceleration amplitudes as a function of

frequency for a given fatigue time. It represents the time, after which a human person subjected to these vibrations would perceive his working capability as reduced.

2. Permissible stress, which may not be exceeded, without special precautionary measures, even if the human subject undertakes no type of work. It corresponds to acceleration values twice as high as the limit for reduced working capability.

3. Reduced comfort, which corresponds to acceleration values that are 3.15 times lower than the limit for reduced working capability. The ISO curves correspond to frequencies between 1 and 80 Hertz. Two different computational methods are specified. Application of the ISO methods to railroad vehicles has certain disadvantages, so that an adaptation of the procedure is being worked out.

Reasons for the Wz values

The Wz values have been determined on the basis of extensive experiments. Up to now, no other method is known which would better characterize the quiet running of the vehicles. The German railway and also other railway administrations have had good experience in applying this method to passenger cars and freight cars, so that this method is also recommended and permitted for application by the UIC. Details of the method can be taken from the references. They can be requested from the Bundesbahnzentralamt (Federal Railway Central Office), Weserglacis 495, Minden (Westphalia), West Germany.

References:

Hellberg/Sperling, Verfahren zur Beurteilung der Laufeigenschaften von Eisenbahnwagen (Methods for Evaluating the Running Characteristics of Railway Cars), Organ für Fortschritte des Eisenbahnwesens (Organ for Advances in Railway Matters), 69th year, Number 12, 1941.

Sperling/Betzoldt, Beitrag zur Beurteilung des Fahrkomforts in Schienenfahrzeugen (Contribution to the Evaluation of Riding Comfort in Rail Vehicles), Glasers Annalen, October 1976.

ORE Utrecht, Report C 116, RP 8, April 1977.

1.2.3 Maintenance Guide Values

During the course of decades of maintenance practice, the German Federal Railway has assembled much empirical experience. This experience is in the first instance reflected, not exclusively in directives, guide values, or measured values, but is to a considerable extent transmitted as more or less global know-how within a well-trained maintenance staff.

Because of considerable reductions of personnel, less-experienced personnel has in recent years also been present in the track maintenance service. Consequently, it has proven suitable to provide this remaining personnel with more objectivized evaluation standards, in order to facilitate evaluation of track condition.

Geometric Guide Values

With respect to geometric guide values, the German Federal Railway makes the following distinctions for various speeds and operating modes

- quality tolerances according to track work
- maintenance guide values and
- tolerances for track geometry.

The individual values are apparent from Figure 1.3.

These values are based on the measuring system used by the small inspection car of the German Federal Railway (see Point 4.2.2). They represent provisional values. In order to facilitate evaluation of the analog record, the individual measured quantities, except the gage, are referred to the "peak-peak" displays of the measurement record (Figure 1.4), since the reference base for individual parameters cannot be precisely defined.

Reasons for the geometric guide values

The quality limit-tolerances describe the track quality which can normally be demanded, after track operations have been carefully performed with modern track maintenance machinery. When refurbished track materials are utilized, as is the rule on lines with a speed up to 80 kmh, certain subtractions must be made from these requirements. This is expressed in the figures of Figure 1.3.

Acceptance of track operations is implemented by a measurement run with a small track geometry inspection car; this checks adherence to these values.

The maintenance guide values describe the condition of individual track parameters. A track will, as a rule, have these parameters

Figure 1.2. LIMIT VALUES FOR ACCELERATION MEASUREMENTS

1	2	3	4	5	6	7	8
Running no	Operating and building measures to be initiated, depending on the line load and on the extent to which the measured value has been exceeded	tangent		curved			
		b_{dyn}	F_{y3}	b_{quasi}	b_{dyn}	b_{abs}	F_{y3}
		m/s^2	KN	m/s^2	m/s^2	m/s^2	KN
1	minimum triggering value	1,3	17	1,2	1,3	-	17
2	maintenance measures must be initiated within 1-3 months	≧1,5	≧22	-	≧1,5	≧2,3	≧22
3	speed limit around 20 - 50 km/h must be posted with great urgency	≧2,5	≧28	≧1,3	≧2,5	≧3,5	≧28
4	exceptions, if removal of the trouble spot can certainly be accomplished within 5 days	2,5*) bis 3,0	-	-	2,5*) bis 3,0	3,5*) bis 4,0	-

*This exception does not hold , when a quasistatic acceleration $b_{quad} \geq 1.3\ m/s^2$ has been determined.

Likewise, measures must be immediately taken, if $F_{y3} \geq 28$ KN has been measured for the lateral suspension force. The trouble spot must be monitored until this defect has been removed.

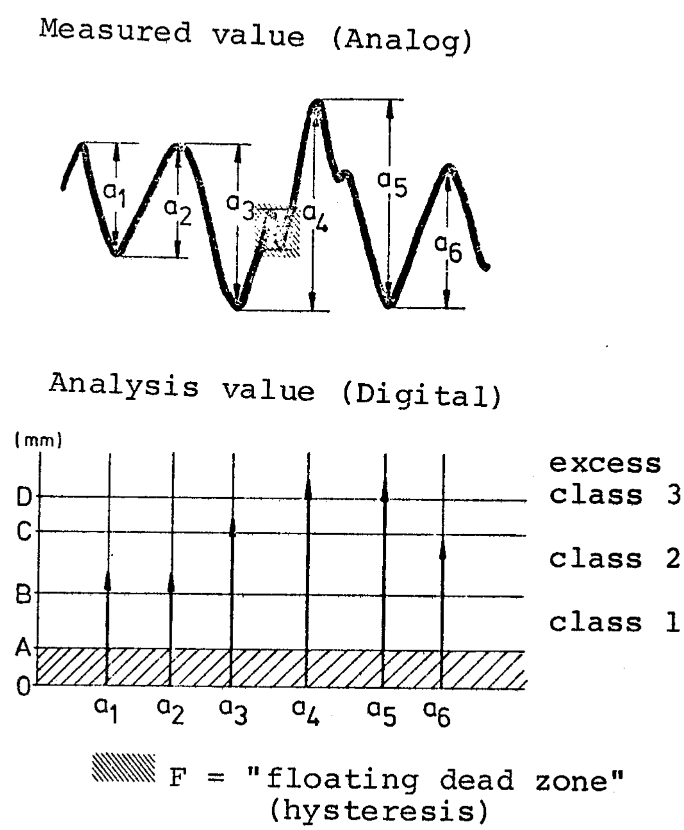
line	Speed range km/h	Criterion	Track parameter				
			Change of track surface (mm)	warp defect 0/00	gage (mm)	superelevation defect (mm)	offset difference (mm)
		c	d	e	f	g	h
		Evaluation system	peak/peak	peak/peak	from 1435 (+/-)	peak/peak	peak/peak
1	> 140	quality tolerance according to track operations	6	2,5	-5 +5	4	6 1) 2)
2		maintenance guide values	11	4	+10	10	12
3		tolerance for track geometry	16	5	-5 +30	20	14
4	> 80 bis ≤ 140	quality tolerance according to track operations	6	2,5	-5 +10	4	6 1) 2)
5		maintenace guide values	12	4	+15	12	14
6		tolerance for track geometry	16	5	-5 +30	20	16
7	≤ 80 passenger and freight traffic	quality tolerance according to track operations	8	2,5	-5 +15	5	8 1)
8		maintenance guide values	14	4	+20	15	16
9		tolerance for track geometry	18	5	-5 +35	25	18
10	≤ 80 only freight traffic	quality tolerance according to track operations	8	2,5	-5 +15	5	8 1)
11		maintenance guide values	16	4	+20	20	18
12		tolerance for track geometry	20	5,7	-5 +35	30	22

1) values refer to the guide rail

2) permissible deviations from the desired alignment (at the benchmark point according to AzObv 13) after renewals a) guide rail ± 15 mm

Figure 1.3 Preliminary limit values for rail classification

Figure 1.4 ANALYSIS OF PEAK TO PEAK MEASURED VALUES



if it is intended for a maintenance measure (tamping and alining), within the framework of track operations planning. These maintenance measures are performed about 10 to 12 months later, i.e. the maintenance guide values take into account the further impairment of track quality, which is expected by the time maintenance is implemented.

The tolerances for track geometry are based on the experience that, simultaneous with their occurrence, the safety and comfort tolerances named under Points 1.2.1 and respectively 1.2.2 will also be reached. If these limit tolerances are found during an inspection car run, this defect (trouble spot) is immediately removed. The defect is generally a local one.

Acceleration guide values

The comfort measurements cited under Point 1.2.2 are also used to evaluate the track conditions with respect to the necessity of maintenance measures (see Figure 1.2). The reason for this is that they take into account the behavior of the vehicle.

Reasons for acceleration guide values

The predominant sense of this measurement lies in the fact that the vehicle reactions can also be measured on track quality defects, which under some circumstances may be quite small but periodic. This is especially significant for the velocity range > 140 kmh. In order to anticipate the securing of comfort on high speed lines, and in order to be able to perform maintenance measures according to schedule, i.e. primarily in a cost-favorable way the tolerances which have been presented were promulgated.

Measurements of track geometry from the vehicle provide little information with respect to the condition of switches. Consequently, in addition to local evaluations by manual measurements, acceleration measurements are especially important for switches.

1.2.4 Additional Remarks

Because of the rapid development of measurement and evaluation technology, the present status is one in which new measurement and evaluation methods have already been introduced, but older measurement and evaluation methods are simultaneously still being handled. This temporal juxtaposition of geometric and dynamic measurements provides valuable information and experience concerning the interactions of rail and vehicle. In particular, with respect to the tolerances that must be adhered to at high speeds (200 kmh), new information is expected, which will lead to a reduction of track maintenance expenditure.

It is intended to acquire, by means of one vehicle, the various evaluation parameters - safety, comfort, maintenance guide values - with respect to track-geometric and dynamical characteristic values.

For the rest, this juxtaposition of several systems is characteristic of the track maintenance philosophy represented by the German Federal Railway. This philosophy considers experience, empirical findings, and systematic research and development as equally important.

1.3 Measurement Methods for Checking the Forces between the Wheel and the Rail

Determination of forces in the wheel-rail area has always been especially significant for determining running safety and riding quality as well as for determining stresses on the wheel and rail. The measurement of running characteristics of vehicles over large sections of rail is especially significant for the track, so that the reaction of the wheel to various track defects, elasticities of ballast, and subsoil and track construction can be understood. Because measurements at the rail are very laborious, such measurements can sense the progression of forces only over a short section. When measurements are made from the vehicle, the forces between the wheel and the rail are derived from the material strains of the wheelset, which in turn are determined from strain measurements. Up to now, especially such methods have been developed in which the forces are acquired through strains in the wheel spokes or wheel discs.

The method presently used by the German Federal Railway is based on measuring the bending strain of the wheelset axle. With this method, it is possible to make a continuous measurement of the Y- and Q-forces at the wheel contact point.

The theoretical foundations, measurement equipment, and measurement transducers, the adjustment and calibration of the measurement equipment, as well as the implementation of corrections can be seen in the appended literature (Appendix 1.1).

Another publication (Appendix 1.2) reports on the experience with practical application of this measurement method, to determine the forces between the wheels and the rail of passenger cars, freight cars, and locomotives. By using further measurement planes, the driving and braking forces can also be eliminated, so that the forces appearing at the wheel contact points can be measured without falsification. These forces are the Q forces in the vertical direction (Z axis), the guide forces Y in the horizontal direction (Y axis), and the tangential forces T_x in the longitudinal direction (X axis).

The wheel set axle method is currently used only for the approval of new and existing vehicles for higher speeds. Furthermore, the method is used to test line sections which are run at speeds greater than 160 km per hour. Further details are explained in Section 4.

1.4 Vehicle Standards

In the area of vehicle standards, provisional international approval conditions (UIC Memoranda) exist only for freight cars and passenger cars, since smooth international traffic must be guaranteed for these types of vehicles. Such international guidelines do not yet exist for locomotives, since locomotives are used almost exclusively in the national domain. For the few international traffic relations, approval conditions for locomotives are discussed and agreed upon bilaterally. For the domain of the German Federal Railway, approval conditions for locomotives have been worked out. Their essential content will be presented.

1.4.1 Vehicle Standards for Freight Cars

The approval and test conditions for freight cars are contained in the UIC Memorandum 432 VE.

All freight cars permitted in European traffic must be able to move at least 80 km per hour at the maximum load intended for this speed range. Both for existing freight cars and for newly built freight cars, directives exist concerning the self-weight (tare) to which they must adhere. In the case of truck freight cars, the self-weight may not fall below certain values, e.g. 16 tons. There are three types of traffic, which correspond to speed limits of 80, 100 and 120 km per hour. The cars are characterized in correspondence with their traffic types, and in particular:

approved for 80 kmh speed, no identification

approved for 100 kmh speed, identification S

approved for 120 kmh speed, identification SS

<u>Running Characteristics</u>

Before a railway administration classifies a car construction type as suitable for S or SS, it must test its riding quality by experiments. Upon request, the owner railway must provide other railways with the measurement records and experimental results. Except in cases of an objection, detailed reasons for which must be presented to the owner railway, the railways must accept cars of this construction type in S or SS traffic of their lines, with the wheelset load provided for in the individual line categories.

Experimental Conditions

Running experiments must be performed with empty and with loaded cars. The car here must have the maximum weight, with which it is intended to be used in S or SS traffic. The load must be distributed as in regular operation. The wheel rim profiles must be used in a worn state or must be corrected to a usual wear profile. The car must run loose-coupled as the last car.

Condition of the Track

The experiments are to be performed on dry rails. The experimental section must include curves, in which a superelevation deficiency of 130 mm is reached at speeds of 100 or 120 kmh respectively. The geometric characteristic of the unloaded rail is determined by the following values. However, these values do not represent limit values in the sense of riding ability:

Quantity	Characteristic	Value
Track surface in the longitudinal direction	Usual low point at a chord of about 10 m	5 mm for a track with continuously welded rails 10 mm in the joint with a jointed rail
Cross level	Usual defect	7 mm
Track alignment	Usual offset defect at a chord of 10 m	6 mm
Gage	Average value on the length of respectively (2):	
	1000 m in a tangent track	1430-1433 mm >1433-1437 mm >1437-1442 mm

Quantity	Characteristic	Value
	200 m in a curved track	
	Usual defect relative to average value	6 mm
	Maximum change of gage	2 mm per meter
	Minimum value of track gage	1430 mm

Measurements

The experiments must be performed on a tangent section and in curved tracks, with a speed of 110 km per hour for S cars and respectively 130 km per hour for SS cars. This means that the experimental speed is 10 percent higher than the intended operating speed. With the experiments on tangent sections and on curves, at least the following characteristics must be measured:

Characteristic	Unit	Symbol
Accelerations of the car body and the wheel set (or respectively kingpin)	m/s^2	
- transverse, leading		$\ddot{y}_I$*
- transverse, trailing		$\ddot{y}_{II}$*
- vertical, trailing		$\ddot{z}_{II}$*
Wheelset bearing - lateral forces	KN	H
- for cars with individual wheelsets at the leading and trailing wheelset		
- with truck cars, at the wheelsets where the greatest forces are expected (1)		

These characteristics must be determined as functions of the speed by

- their maximum value (with H, measured at 2 m)
- their arithmetic average
- their standard deviation S
- their fundamental frequency (Hertz), applies only to acceleration

The maximum forces are, as a rule, expected with truck cars (TTX); they occur on tracks and at clear trouble spots in tangent sections at the leading wheelset of the leading truck; they also occur when resonances and instabilities appear in tangent sections, on the trailing wheelset of the trailing truck. If not all the wheelsets of the vehicle are equipped with H*-force measurement devices, the vehicle may be turned around for the experiments. Furthermore, the following measurements are recommended:

- guide forces Y between the wheel and the rail (kN)
- wheel forces Q between the wheel and the rail (kN)

Both forces are to be determined at the first wheel set.

These characteristics must be determined in dependence on velocity, by means of the following:

their maximum value (Y measured at 2 m)
their arithmetic average and
their standard deviation.

*H = horizontal

Evaluation Criteria

The following criteria must be fulfilled during the experiment, so that the running conditions can be declared acceptable:

Characteristics	Evaluation criteria
Riding quality (vertical and lateral) Wz after Sperling (2)	$Wz \leq 4.25$
or	
Standard deviation (S) of the evaluated acceleration	$S_p \leq 0.13$ g
Lateral forces (ORE B56/RP3)	H_{max} (2 m) $\leq 0.85\ (10 + p/3)$
at the bearing boxes (PRUD'HOMME)	$H_{average} + S \leq 0.5\ (10 + p/3)$
	p = wheelset forces (kN) (=2Q)

The standard deviations are calculated in the range from 0.5 to 1 Hertz without frequency-dependent evaluation of acceleration, and in the range from 1 - 25 Hertz with a frequency-dependent evaluation of acceleration, whose drop amounts to two octaves.

1.4.2 Provisional Vehicle Standards for Passenger Cars

The provisional UIC standards for passenger cars are excerpted in Appendix 1.3. They no longer correspond to the most recent state of revision. This most recent status is not known to the authors, since the last relevant discussion took place during their stay in the USA.

For detailed answers to this complex of questions, reference is made to the following address:

Bundesbahnzentralamt
Weserglacis
D495 Minden (Westphalia)

It is requested that direct contact be established with this agency, in order to secure an exhaustive transfer of know-how.

1.4.3 Provisional Vehicle Standards for Locomotives

The German Federal Railway has not introduced generally applicable standards for locomotives; rather a specifications manual is set up for each type of locomotive (passenger train locomotive, freight train locomotive, shunting locomotive). This specifications manual is always coordinated to the special requirements.

The requirements refer to general specifications, such as

- utilization of the locomotive
- maximum axle load
- inspection schedules

and also to the mechanical portion, such as

- running characteristics
- trucks
- locomotive loads

and also to the electrical parts.

With respect to technical running acceptance, the following provisional conditions have been specified:

Acceptance runs on predominantly tangent sections of 100 km length and on a section with many curves, likewise 100 km in length. The suitability of the line section, i.e. representative track conditions with good track as well as track shortly in need of maintenance, is determined by a comparison standard vehicle, whose running behavior is known. The line section must contain curves, which are run at 0.85 m/sec^2 unequalized lateral acceleration. The following quantities are measured both with the experimental vehicle and with the comparison vehicle:

Forces:

$$Y_{11},\ Y_{12},\ Y_{21},\ Y_{22}$$

$$Q_{11},\ Q_{12},\ Q_{21},\ Q_{22}$$

Accelerations:

$$\ddot{y}_I{}^* \text{ and } \ddot{z}_I{}^*$$

The following characteristic quantities are formed from the measured values:

$$Y_{ij}/Q_{ij}\ , \qquad \Sigma\ Y_i/(10 + \Sigma Q_i/3)\ , \quad \Delta Q_{ij}/Q_{0ij}$$

Index i for the wheelset position, where 1 is the first wheelset in the direction of travel

Index j for the wheel position where 1 indicates the right in the direction of travel and 2 indicates the left in the direction of travel.

These designations correspond to international agreements for measurement quantities, as these are contained in the ORE Report B55, RP 6.

For evaluation purposes, the acceptance lines are subdivided into sensible sections, corresponding to their geometry. The time records of the measurement quantities and characteristic quantities are digitized. For each section and each quantity, the frequency-sum distributions or digital values are determined. The determined characteristic values of the distributions may not exceed the associated limit values. The latter have not yet been definitively specified.

It is not permissible for the locomotive to exceed limit values when the reference vehicle simultaneously adheres to these limit values.

For answering detailed questions, reference is made to the following address:

Dezernat 108,
Bundesbahnzentralamt
Arnulfstrasse 19
8000 Munich 2
West Germany

With respect to acceptance conditions for locomotives, the authors refer to the ORE Report B 10/RP 12 of 1969. This report explains the fundamental principles concerning technical running experiments on locomotives. It can be obtained in English from the ORE. Among other things, it includes the following chapters:

Evaluation criteria for running characteristics,

Derailing safety,

Standard experimental program,

Measurement technology and

Evaluation.

The specifications are in the meantime partly outdated. They are replaced by national specifications, depending on the state of technology. The basic information in this report, however, is still valid today.

1.5 Summary and Conclusions (Recommendations)

The track standards introduced for safety, comfort and maintenance as well as for the acceptance of track work, and the vehicle standards are based on experience and on the application of results of international research and development.

Because of the introduction of 200 kmh speed, measurements of geometry, acceleration, and force have for some time been performed side-by-side and have been related to one another. This has resulted in new information regarding the vehicle/rail interaction. This information results in an ongoing adaptation of the standards.

Because the standards have economic significance for practical application, an extensive research program was initiated and is currently being implemented.

By comparing current track standards in the USA and track standards in Europe, the authors believe that they are able to draw the following conclusions (recommendations):

1. Presently valid standards for warping in US track categories 1-3 appear to be too generous relative to the understanding of the ORE B55 (RP 6) committee. Taking into account measurement

tolerances and transmission errors of the measurement system, an adequate safety margin would have to be maintained with respect to critical derailing warp.

2. The horizontal and vertical forces (Y and Q), which appear at the wheels, should be measured by means of measurement axles for locomotives and cars, so that adherence to the derailing criteria Y/Q can be checked. On this point, too, the investigations of the ORE committee B55 provide information.

3. Track stresses by lateral forces from running vehicles should be limited in accord with the relation

$$\Sigma Y \leq 0.85\ (10 + 2Q/3) \quad [\mathrm{kN}]$$

4. Acceptance criteria should be specified for the approval of vehicles. These criteria must be guided by American track conditions.

5. The meaningful application of standards for tracks presupposes that trained personnel, familiar with track behavior (time development of track quality deficiencies), is utilized in track maintenance. A minimum staff of experienced personnel should be retained.

2. Component Standards

2.1 Rail Stresses

2.1.1 General Considerations

The rail profile and the rail strength determine the possible stress on the rail and its support. Until the year 1965, the German Federal Railway had used rails with maximum 49 kg/m weight (S 49) and 700 N/mm^2 tensile strength. With these rails, however, rail ruptures occurred to a considerable extent. These were caused by the introduction of higher axle loads and higher train speeds. The laboratories of Munich Technical University, Institute of Prof. Dr. Eng. Eisenmann, performed detailed investigations on this point. These investigations clarified that the reason for these ruptures was the fact that the long term stability and form stability* of the rail had been exceeded. For this reason, stress limits were in the meantime introduced for the rail S 49. Depending on the proportion of freight train load and on the average axle load of the freight train, these limits lie at 240 - 400 million tons load.

In order to achieve the longest possible load-independent lifetime, the rail S 54 (54 kg/m weight) was introduced in 1965, and the rail UIC 60 (60 kg/m) was introduced in 1970. Both rails have a tensile strength of 900 N/mm^2. This measure envisioned the anticipated further increase of axle load and speed. Today,

---

*Form stability means the permanent stability of a construction part of arbitrary shape. It depends on its rated strength (with rails, e.g. 700 or 900 N/mm^2 tensile strength). Form stability is not a pure material characteristic, but a strength characteristic that is affected, and generally reduced, by the shape and working of the material.

these rails are installed exclusively, depending on line load. The results of detailed studies regarding the long term stability of these rails will be reported below.

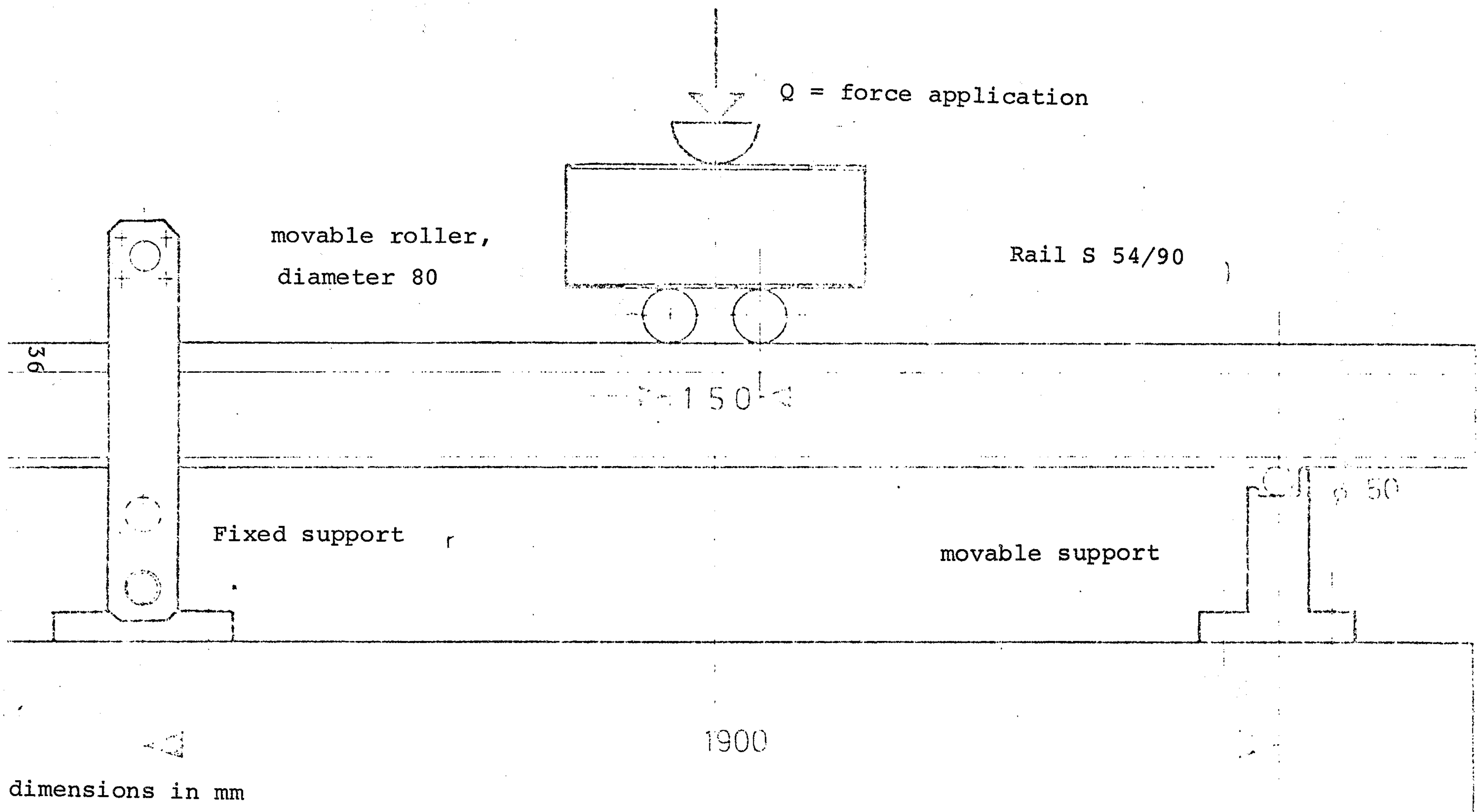
2.1.2 Stress on the rail base

In order to determine the permissible bending stress on the rail, in dependence on the relevant influential factors, the form stability or the permanent stability of the rail must be determined.

For this purpose, the German Federal Railway has caused the form stability of individual rail profiles to be determined by means of tests and experiments at Munich Technical University. Both new and corroded rails were involved. The experimental setup for long term vibration experiments is shown in Figure 2.1. The support width of 1.90 agrees with the basic value L of the long tie structure after Zimmermann. This generates a sequence of moments similar to that appearing in a track under an individual load. This load is applied to the rail through two movable rollers at a distance of 150 mm. The bending stress, generated at the rail base in the area of the bending zone, is measured by means of strain gauges. The continuous vibration experiments are performed at a constant understress of σ_u = 50 N/mm^2 and at a test frequency of 5 Hertz. The overstress is varied in such a manner that both the short term and long term strength ranges are included (Figure 2.2).

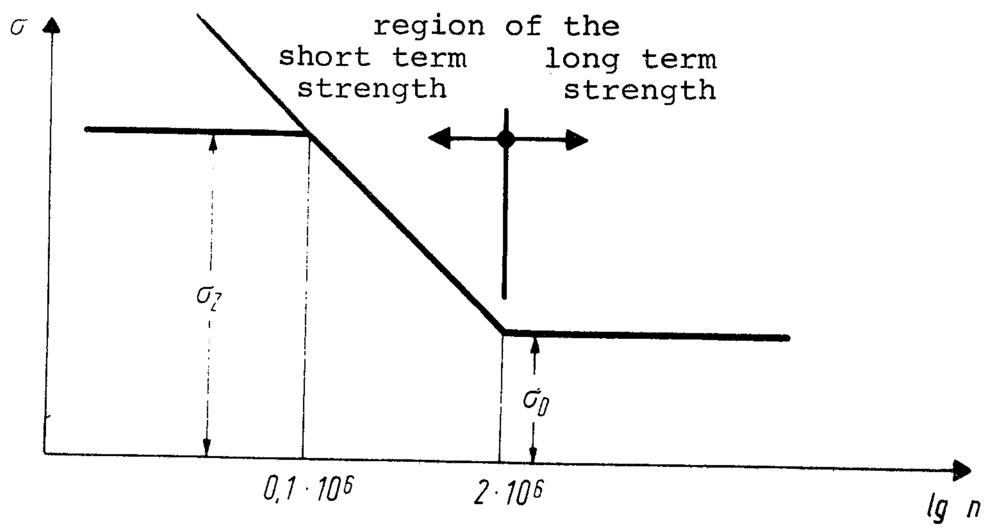
From a large number of experiments, the "Wöhler curve" can be determined as the result. This curve provides information

Figure 2.1 EXPERIMENTAL ARRANGEMENTS FOR CONTINUOUS VIBRATION EXPERIMENTS ON RAILS



dimensions in mm

Wöhler Curve



σ_D = long term strength (see Smith Diagram)

σ_Z = short term strength

n = number of load repetitions

The Miner hypothesis

for $0.1 \times 10^6 < n < 2 \times 10^6$

$$\sum_{i=1}^{i=k} N_i/n_i \leq 1$$

N = present load alternation

n = permissible load alternation

k = number of load types

(cumulative load)

Figure 2.2. WÖHLER CURVE OF CONTINUOUS VIBRATION EXPERIMENTS ON CORRODED RAILS S54 (900 N/mm^2 tensile strength).

concerning the number of sustainable load alternations, in dependence on the overstress σ_o or respectively in dependence of the vibration width 2 σ in N/mm^2.

The result of such studies for corroded rails S 54 (900 N/mm^2) is shown in Figure 2.3.

When the results of such investigations are compared for new rails and corroded rails, it becomes clear that e.g. the rusted rails have a form stability that is about 25 - 40 percent less as compared to that of new rails.

The relevant form stability is determined by the vibration width which the rail can sustain at 2 x 10^6 load alternation without fracture. The form stability or permanent stability of the rail declines with the magnitude of the understress superposed on the rail.

The connection between understress and permissible alternating stress (vibration width 2σ) can be represented in a continuous strength diagram after Smith. The permissible form strength towards the top is limited by the respective plastic limit of the material, which lies at approximately 55 percent of the tensile strength. The Smith diagram for rails with a rated strength of 700 and 900 N/mm^2 is shown in Figure 2.4.

The German Federal Railway counts with a constant self-stress of the rail in the amount of 80 N/mm^2 and with a temperature stress of 100 N/mm^2 as understress. The diagram shows that the vibration width and consequently the permissible form strength decreases as the understress increases, corresponding to increased tensile stress in winter at lower temperatures. Accordingly, a safety margin against the occurrence of rail fractures may exist during summer, but it is reduced in winter at lower temperatures. This generally appears clearly in a considerable increase of rail fractures during the first cold days of winter.

If the existing stresses differ severely, where only a few load alternations occur at a stress high enough to exceed the form stability, the possibility exists of setting up an exact long term strength investigation according to the "Miner hypothesis":

$$\sum_{i=1}^{i=n} N_i/n_i \leq 1$$

Here, N_i is the number of existing load transitions, n_i is the number of permissible load transitions. The Miner hypothesis presupposes knowledge of the curve $N = f(\sigma)$. It leaves out of account experimental parameters such as average load, spectrum, sequence, and interference, all of which likewise affect the lifetime.

The existing bending stress is calculated according to the Zimmermann/Eisenmann method - principle of continuous mounting on elastic single support. The relevant influential parameters are the following:

axle force

distance between ties

effective tie support surface

rail shape

Fig. 2.3: LONG TERM VIBRATION EXPERIMENTS WITH CORRODED RAILS S 54/90

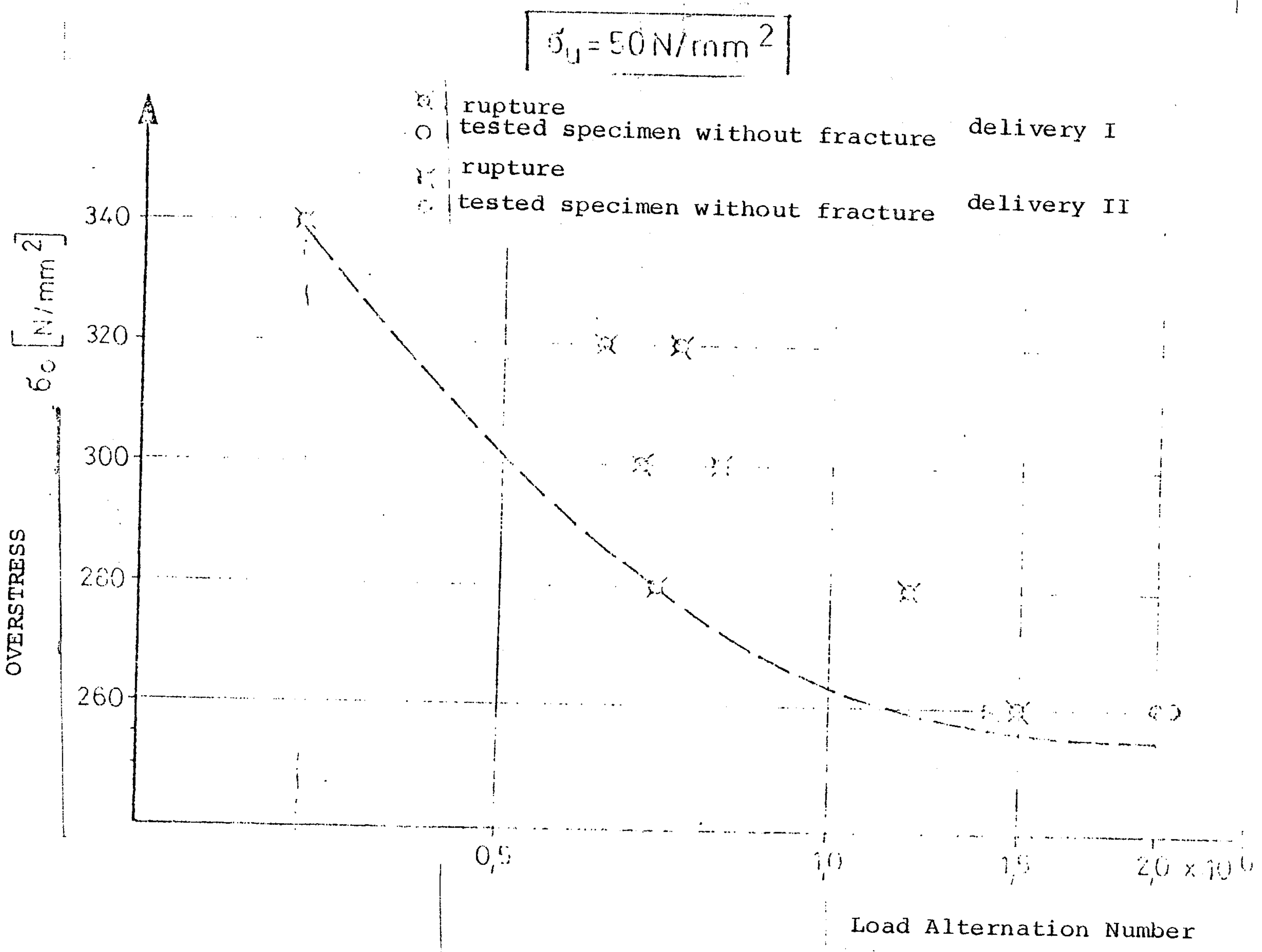
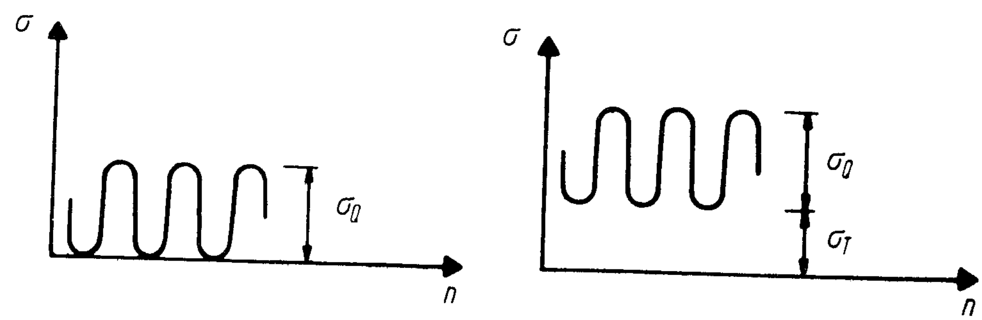


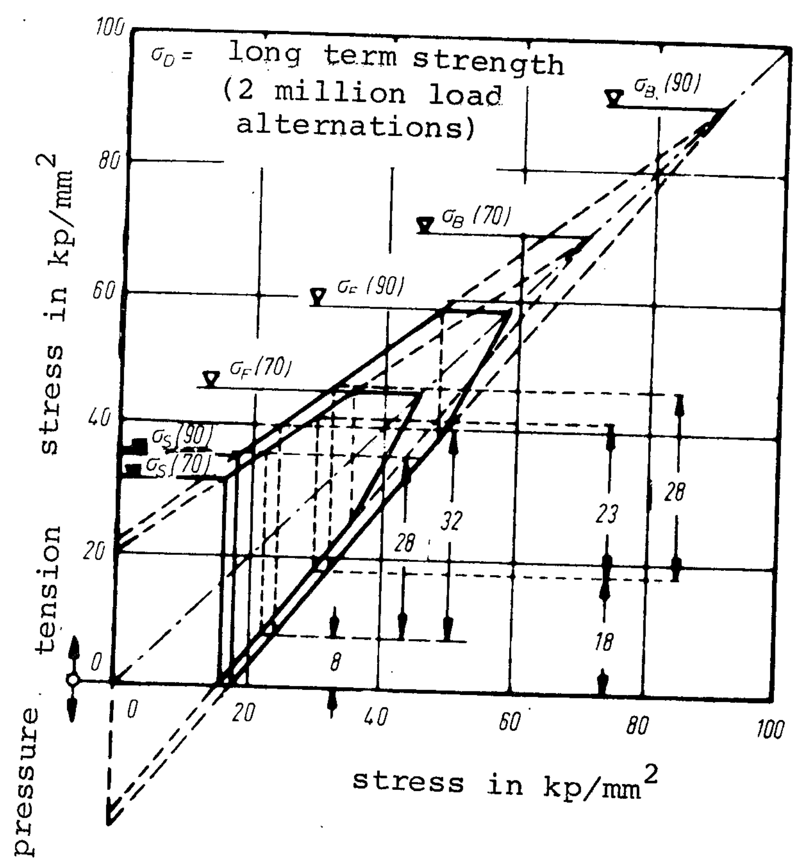
Figure 2.4:

Smith diagram for rails with a rated strength of 70 and 90 kp/mm^2



σ_Q = stress due to traffic load

σ_T = stress due to temperature (hindered contraction)



elasticity of ballast and of subsoil and

track conditions, so that the influence of the dynamics can be taken into account.

The allowability of certain rail shapes, relative to the existing track construction and prevailing axle force, can be determined by comparing the actual bending stress with the permissible long term bending stress.

2.1.3 Stresses on the Rail Head

High axle forces can cause the permissible Hertz compression to be exceeded in the rail head. This becomes noticeable by the appearance of rail break-outs (shelling) at the rail. The stress condition causing this phenomenon can be studied by means of the half-space theory and the formulas of Boussinesq. For a calculational assumption regarding the maximum main shear stress, it appears reasonable to start from the following formula:

$$\tau_{max} \simeq cp$$

p = contact pressure in the wheel/rail contact surface

c = a factor depending on the contact surface, which can be assumed at 0.3.

When calculating the contact surface and the contact pressure, the starting point is the respective actual contact geometry. A study of contact relationships between the wheel and the rail, of the German Federal Railway, showed that the calculation is permissible with the simplifying assumption of a rolling contact with a rectangular support surface. Such a study would also have to be performed for the contact conditions of American railways, in order to secure the transferability of the formulas to American conditions. The following simplified solution results:

$$p = 138\sigma \sqrt{Q/r} \qquad [N/mm^2]$$

With $\tau_{max} = 0.3p$, the following formula is obtained for the maximum shear stress

$$\tau_{max} = 413 \sqrt{Q/r} \qquad [N/mm^2]$$

r = wheel radius in mm

Q = wheel force in kN

This relation shows that the maximum rail head stress depends on the ratio of the wheel force to the wheel radius. When the wheel force Q is increased, either the wheel radius r should also be increased, or the permissible shear stress must be increased by using a rail with higher tensile strength. Otherwise, rail head shelling must be expected.

Experiments on a plastic rail, with cast-in strain gauges, confirmed that half-space theory was valid and was transferable to the rail.

The permissible shear stress is decisive for calculating the permissible wheel force or the permissible radius. By means of the theory of constant deformation work, the permissible shear stresses can be determined for the stress condition near the wheel-rail contact point. This wheel-rail contact point is distinguished by omnidirectional pressure. The permissible shear stresses depend on the permissible tensile stresses of the rail steel for the bi-axial stress state. The permissible shear stresses are calculated as follows:

$$\tau_{perm} = \sigma_{perm}/\sqrt{3}$$

If it is further considered that the present case involves a long term strength problem, as a consequence of the repeated action of the load, 50 percent of the respective tensile strength can to a first approximation be used for the permissible rail stress.

The resulting permissible shear stress is as follows:

$$\tau_{perm} = 0.5\sigma/\sqrt{3} \cdot 1/\gamma$$

where σ = the tensile strength

γ = a safety coefficient.

The unavoidable scatter of rail quality, especially the effect of interior inclusions, is taken into account by means of a safety coefficient. For railways with only freight train traffic and without excessive speed, the safety coefficient can be dispensed with. Thus γ becomes 1. By means of the shear stress appearing at the rail head, and by means of the permissible shear stress, the permissible wheel force can now be calculated in dependence on the wheel radius. The calculation is as follows:

$$Q_{perm} = 5.26 \times 10^{-7} \cdot r \cdot (\sigma/\gamma)^2 \quad \text{(kN)}$$

where

r = radius in mm

Q = effective wheel force in N

σ = tensile strength (N/mm^2) and

γ = safety coefficient

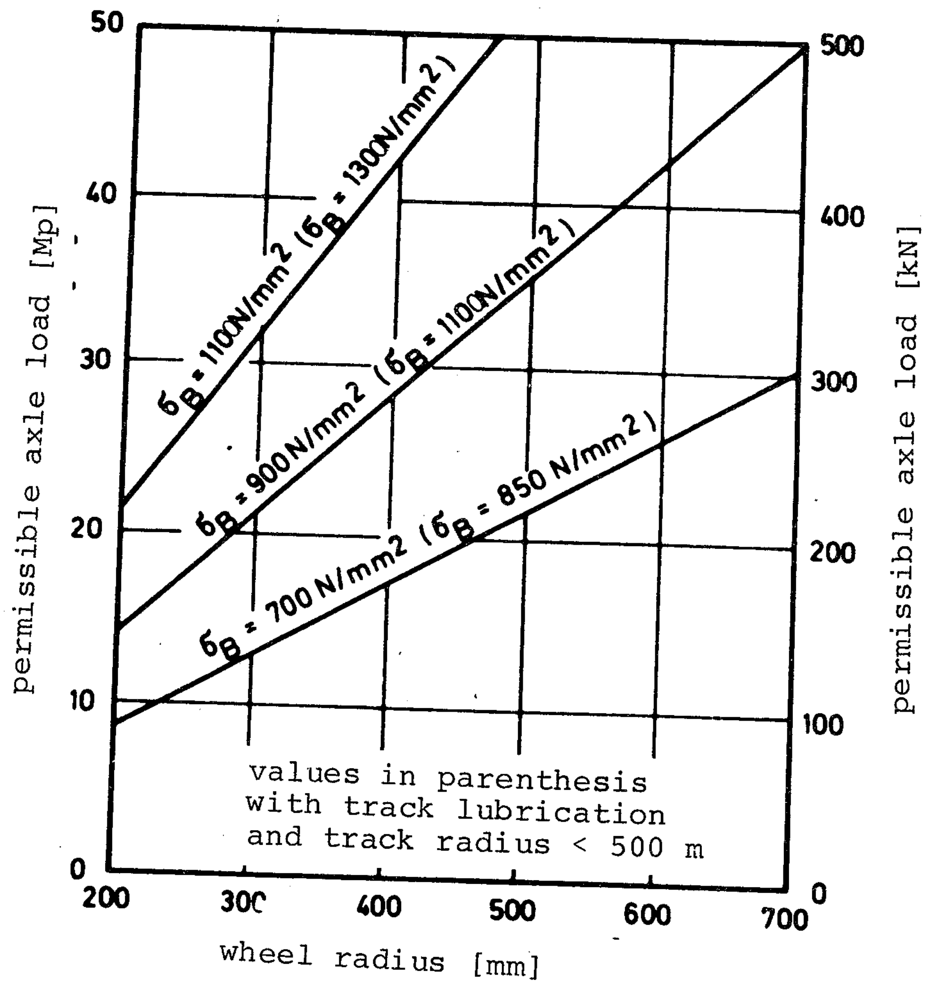
The results of the theoretical investigations that have been performed are shown in Figure 2.5.

From this it is apparent that the permissible axle load 2Q and the permissible axle radius depend strongly on the strength of the rail steel. This was also the reason why the German Federal Railway simultaneously chose a higher rail tensile strength of 900 N/mm^2, as compared to the 700 N/mm^2 for the S 49 rail, when it made the transition to the rail profile S 54 and UIC 60. If wheel forces exceeding the long term strength occur only rarely, rail head shelling will not yet occur, but only rail head pinching. If, to a first approximation, a plastic limit lies at 65 percent of the rail steel tensile strength, the permissible wheel force is as follows:

$$Q = 8.02 \times 10^{-7} \cdot r \cdot (\sigma)^2 \qquad (kN)$$

For particular lines, individual studies may involve a considerable percentage of wheel forces which exceed the permissible long term stress of the rail. In this case, use can again be made of the Miner hypothesis.

FIGURE 2.5 RELATION BETWEEN PERMISSIBLE AXLE FORCE, PERMISSIBLE WHEEL RADIUS, AND RAIL STRENGTH



Assuming that it is valid to apply the derived relation to American circumstances, the permissible wheel force for the American rail RE 136 was calculated in Appendix 2.1 according to the calculational method of the German Federal Railway.

2.1.4 Summary

In order to check the stress condition of a rail for a given load, the load cases of rail head stress and rail base stress must always be studied separately. Rails with higher tensile strength are the best means to prevent the permissible shear stress at the rail head from being exceeded. Use of a larger rail profile is the best means to prevent the permissible bending stress at the rail base from being exceeded.

Details of the calculational method for the rail head and the rail base can be taken from the following book:

The Eisenbahnschiene (The Railroad Rail)

Editor: Dipl. Eng., Dr. Eng. E.h. Fritz Fastenrath,
Wilhelm Ernst and Son, Publishers, Berlin 1977.

This book will shortly appear in English, Frederic Ungar Publishing Co., Inc., 250 Park Avenue South, New York, NY 10003.

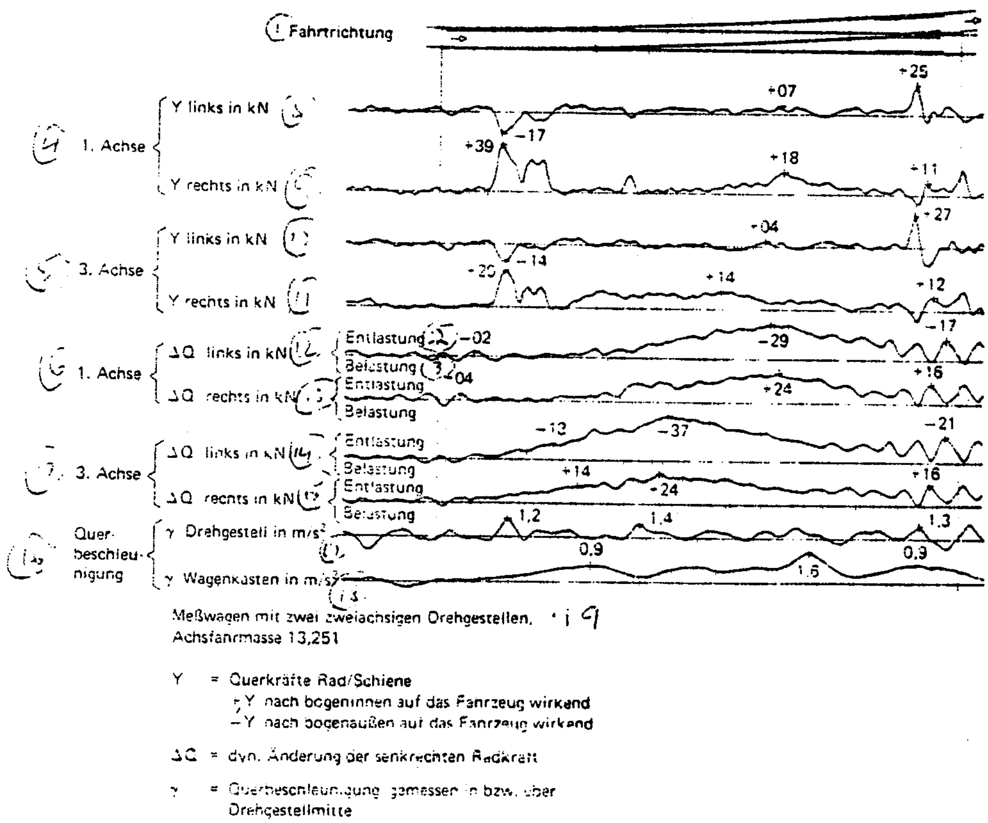
2.2 Switches and Crossings

In switches and crossings, the interaction of forces, accelerations, and motions between the vehicle and the track is particularly important. Figure 2.0 shows a measurement record with a typical progression of forces between the wheel rim and the rail, the wheel load, and the accelerations at the track and on the car body, for a run through a branched track of a switch at a speed of 125 km/h. The peak values of the forces and accelerations occur in the region of the frog and in the region where the switch tongue begins.

Despite every care in the construction, manufacture, and maintenance of switches, it is unavoidable that switches inevitably affect the running of a vehicle. Among other things, they cause rolling vibrations of the vehicle body. This process is not damaging, if the first vibration has already decayed before the next excitation occurs. Rhythmic disturbances by several successive switches or constraint points, on the other hand, can lead to the buildup of vibrations. Consequently, care must be taken that high speed tracks have a sufficiently large distance between switches and other constraint points, so that the above-described effects on vehicle behavior are excluded.

The riding quality of fast vehicles decisively depends on the geometric course of the contact between the wheels of a wheel set and the rails. It is necessary for the wheels to roll without having to slide. Interfering effects from track alignment defects, changes in gage, etc., are neutralized up to a certain point, in the following manner: The wheel-rail contact point migrates laterally. Differences in effective rail head diameter are caused by the conicity of the wheel rim running surface. For the two wheels of one axle, this difference corrects the position of the wheelset on the track.

Figure 2.6 Typical measurement record of a **run** over a switch



1 direction of travel
2 relief
3 stress
4 first axle
5 third axle
6 first axle
7 third axle
8 Y left in kN
9 Y right in kN
10 Y left in kN
11 Y right in kN
12 ΔQ left in kN
13 ΔQ right in kN
14 ΔQ left in kN
15 ΔQ right in kN
16 lateral acceleration
17 γ truck in m/s^2
18 γ body in m/s^2
19 measurement car with two two-axle trucks, running axle weight 13.251

Y = lateral forces wheel/rail
+Y acting on the vehicle toward the inside of the curve
-Y acting on the vehicle toward the outside of the curve

ΔQ = dynamic change of vertical wheel force
γ = lateral acceleration measured at or above the center of the truck

In switches, the steady rolling of the wheels is regularly disturbed. This interfering effect increases with speed. It can be limited, but not quite eliminated, by setting tight operating tolerances and by intense maintenance measures.

Under these perspectives, the German Federal Railway has specified safety, construction, and maintenance standards for switches.

2.2.1 Safety Standards for Switches

For switches, in principle the same safety standards are used as for rails. This specification was made on the basis of the experience that

- system-conditioned lateral forces, caused by running vehicles, are relatively high, and
- the profile of the switch tongue is considerably weaker than the rail profile, but
- as regards panel rigidity, the switch as a whole has a large equivalent moment of inertia.

Furthermore, it has been specified that every switch and crossing must be thoroughly investigated once a year in the speed range up to 160 km/h, and twice a year in the speed range above 160 km/h, as regards its proper overall condition. For this purpose, so-called switch maintenance test booklets have been set up. As an example, Appendix 2.2 presents the booklet for switches and crossings with UIC 60 rails.

2.2.2 Switch Construction

An unpleasant lateral jolt occurs when entering the tongue area.

In order to reduce this jolt as much as possible, one has for several years striven to make entry of the wheel into the tongue arrangement as tangential as possible. In order to intercept the impact, the switch tongue must be switched in such a manner that the deflection of the leading wheel runs as continuously as possible, i.e. the impact angle must be kept appropriately flat. Guide forces and lateral accelerations of a passenger car in a switching curve have a relationship to the impact angle according to studies of the ORE Committee D 72. The guide force Y, and consequently the stress on the switch rails, here increases relatively strongly with the impact angle.

The individual construction elements of various types of switches will not be listed here. Documentation concerning this point can be provided subsequently, if necessary.

2.2.3 Maintenance of Switches and Crossings

In general, what has been said for track maintenance is also applicable to the maintenance of switches and crossings.

In the view of the German Federal Railway, full friction-type locking of the track fasteners is the most important precondition for good quality and long durability of switches and crossings.

In the opinion of the authors, the maintenance of switches and crossings is less a question of maintenance standards than of well-trained and skilled technical personnel. Consequently, this topic will not be treated in detail within the scope of this report.

Relevant documents can be requested subsequently, as needed.

3. Continuously Welded Rails

The German Federal Railway was one of the first railways which systematically developed and introduced continuously welded rails. Today, about 85 percent of all the rails and switches are continuously welded. This development is characterized by the mutual existence and mutual influence of theoretical investigations, experiments, and practical experience. During the course of decades, the main factors have been successfully researched, as regards their significance and as regards their order of magnitude, and these factors have been combined in a relatively simple theory. By means of actually occurring track bucklings, the correctness of the calculational methods could be confirmed.

It is not the loaded but the unloaded state that is decisive for the stability of continuously welded rails. The reason for this is that the possible lateral forces for the loaded state are limited. In calculating the track stability of the unloaded track, however, it is taken into account that the critical condition in front of and under vehicles is given in the domain of the lift-off wave of the track panel. To this extent, the term "unloaded track" quite distinctly refers to the vehicle.

Point four of the present report discusses the considerable advantages of continuously welded rails in view of actually effected reduction of maintenance costs.

3.1 Track Stability of the Unloaded Track

The German Federal Railway early began to research quantitatively the theoretically proposed parameters, and to convert these parameters into construction principles and into practical fabrication and maintenance methods.

3.1.1 Theory (Analysis)

The calculation of the continuously welded rails is generally based on the investigations of H. Meier, who already began his work in the year 1937. The essential assumption of his analysis is that the lateral strength increases linearly for small lateral displacements and is then constant. During the course of the analysis, the calculation proceeds with a constant lateral strength as a simplifying assumption.

The basic studies of Professor H. Meier were converted by his successor, Professor J. Eisenmann, into a relatively simple computational method for determining the critical temperature increase Δt in tangent tracks and curved tracks. Relative to the respective track parameters, and basing itself on a prescribed track quality defect f, this computational method determines the temperature increase at which there exists danger of track buckling, i.e. at which a transition into a labile (unstable) state takes place.

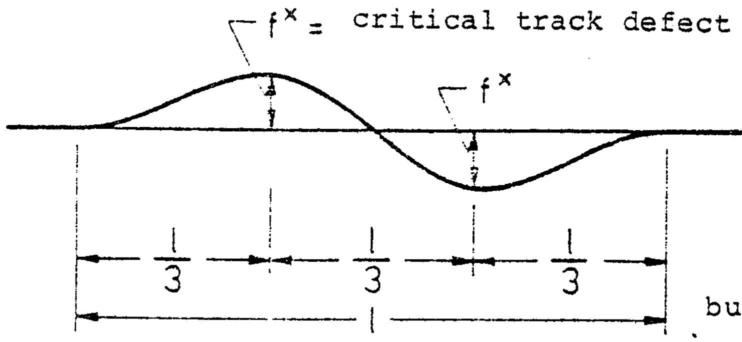
In order to avoid track buckling, the value Δt must therefore be larger than the actual temperature increase ΔT. If practically occurring deviations in the tightening and welding of rails are taken into account, this temperature increase is about 45 - 50°C.

Additional forces occur beneath trains - lateral forces, braking forces, as well as vibrations. Consequently, on the basis of present experience, and depending on speed, the temperature increase of the rails should lie 10 to 50°C below the computationally determined temperature rise. The German Federal Railway counts with a safety temperature of 18°C at a speed of 100 kmh and 50°C at a speed of 200 kmh. A linear interpolation is made between these values.

The sequence of calculations and other details are shown in Figure 3.1. It should be noted that the formulas for curved tracks are valid only for radii from about 350 m to 1,100 m. For larger radii, the formulas for tangent tracks should already be used. With smaller radii, the calculation yields values that are too unfavorable, since the rail in practice unstresses itself somewhat by small lateral motions. The parameters relevant for studying track stability are partly known, and must partly be determined by experiments relative to the respective track construction.

The following are known:

- E - elastic modulus of the rail steel
- α - temperature expansion coefficient of the rail steel
- F - surfaces of <u>both</u> rails
- H - track radius
- f - prescribed track quality deficiency (this must be assumed in accord with local contingencies. The

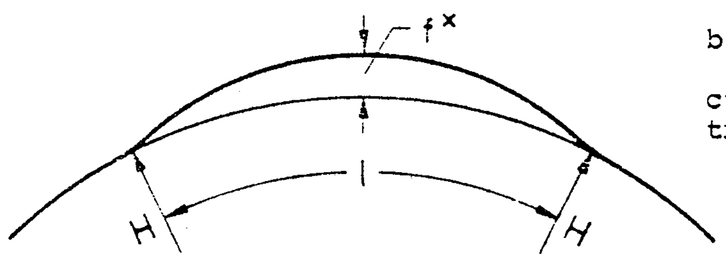


$$l = 3\cdot\pi\cdot\sqrt{\frac{2\cdot E\cdot J}{P_o}}\ [\mathrm{cm}];\quad f^x = 8{,}7\cdot w\cdot\frac{E\cdot J}{P_o^2}\ [\mathrm{cm}]$$

the resulting critical temperature rise

$$\Delta t = \sqrt{\frac{8{,}7\cdot J\cdot w}{\alpha^2\cdot F^2\cdot E\cdot f}}\ [{}^\circ\mathrm{C}]$$

Figure 3.1: Computing rules for evaluating track stability at high rail temperature (method of Prof. H. Meier)



$$l = 2\cdot\pi\cdot\sqrt{\frac{2\cdot E\cdot J}{P_o}}\ [\mathrm{cm}];\quad f^x = \left(w - \frac{P_o}{H}\right)\cdot\frac{16\cdot E\cdot J}{P_o^2}\ [\mathrm{cm}]$$

the resulting critical temperature

$$\Delta t = -\frac{8\cdot J}{\alpha\cdot F\cdot H\cdot f} + \sqrt{\left(\frac{8\cdot J}{\alpha\cdot F\cdot H\cdot f}\right)^2 + \frac{16\cdot J\cdot w}{\alpha^2\cdot F^2\cdot E\cdot f}}\ [{}^\circ\mathrm{C}]$$

where

E = elastic modulus of the rail steel in kp/cm^2

α = temperature expansion coefficient of steel

F = surface of both rails in cm^2

J = equivalent moment of inertia of the track panel in cm^4

H = track radius in cm

w = track lateral strength in kp/cm

f = assumed track deficiency in cm (1.5 to 2.5 cm)

$P_o = \alpha\cdot\Delta t\cdot E\cdot F$ = critical track pressure force in kp

German Federal Railway counts with 1.5 - 2.0 cm).

The parameters:

I - equivalent moment of inertia

w - track lateral strength

must be determined by experiments. Closer explanations on this matter are given under Point 3.1.3.

A calculational example is shown in Figure 3.2. However, the term "neutral temperature" is mentioned in the text. This is the temperature at which, in the present case, the rails are welded and are friction-locked with the ties - "tightened".

Figure 3.3 contains another example. It is here investigated whether track displacements at low temperatures cause a signification falsification of stress conditions in the track. Such displacements occur during the course of time because of track maintenance. It appears that the neutral temperature is falsified by 14°C, when the assumed radius changes from 500.00 m to 499.95 m. Consequently, at high rail temperatures, the track in its freshly tamped state is no longer safe.

References:

Eisenmann: Track Stability of Rails at High Speeds (enclosed as Appendix 3.1).

Meier: Simplified Method for the Theoretical Study of Track Buckling, Organ für die Fortschritte des Eisenbahnwesens, (Organ for Railroad Progress), No. 20, October 1937, Julius Springer Publishing Company, Berlin.

Meier: New Developments in Track Construction, Verkehr und Technik (Traffic and Engineering), 1963, No. 78.

Biermann, Raab: On the Development of Continuously Welded Rails, Eisenbahntechnische Rundschau (Railroad Engineering Survey), No. 8/1960.

Eisenmann: Research Work in the Area of Railroad Track and Street Surfacing, Wilhelm Ernst and Son, Publishing Co., Berlin-Munich-Düsseldorf, 1974.

Prud'homme: The Resistance of Rails Against Stresses Emanating from Vehicles. Monthly Journal of the International Railway Congress Association,August/September 1967.

Raab: Stability Relationships with Continuously Welded Railway Rails. Eisenbahntechnische Rundschau (Railroad Engineering Survey), 1958.

Kerr: The Effect of Lateral Strength on Lateral Track Bucklings. Schienen der Welt (Rail International), June 1976.

Other references can be found in the cited publications.

3.1.2 Experiments for Testing the Theory

Fundamental experiments were performed in 1958/1959 by Professor Raab in collaboration with Professor Birmann in Karlsruhe, under contract with the German Railway. These experiments are extensively described in the literature, so that their presentation can be dispensed with in the present report.

Since that time, the German Federal Railway has no longer performed such experiments or contracted for them, but has exlusively limited itself to detailed research on the relevant parameters

Assumptions:

Track form	:	UIC 60-1667-B 70 W
Track condition	:	freshly refurbished rails
Radius	:	H = 500 m
Equivalent moment of inertia	:	$J_E = 2900\ cm^4$
Rail cross section	:	$F_{2CS} = 153.72\ cm^2$
Elastic modulus	:	$E = 2.1 \times 10^6\ kp/cm^2$
Heat expansion coefficient	:	$\alpha = 0.0000115\ 1/°C$
Assumed track quality deficiency	:	f = 1.5 cm

Decisive track lateral resistance:

decisive $\omega = 14.1 \cdot 0.5$[1]$\cdot 0.8$[2] $= \underline{5.64\ kp/cm}$

$$\text{critical } \Delta t = -\frac{8\ J_E}{\alpha \cdot F \cdot H \cdot f} + \sqrt{\left(\frac{8J_E}{\alpha \cdot F \cdot H \cdot f}\right)^2 + \frac{16J_E}{\alpha^2 \cdot F^2 \cdot E \cdot f}\ \text{decisive } \omega}$$

$$= -\frac{8 \cdot 2900}{0.0000115 \cdot 153.72 \cdot 50000 \cdot 1.5} + \sqrt{\left(\quad\right)^2 + \frac{16 \cdot 2900 \cdot \text{maßg. } \omega}{0.0000115^2 \cdot 153.72^2 \cdot 2.1 \cdot 10^6 \cdot 1.5}}$$

$$= -175 + \sqrt{30619 + 4714 \cdot 5.64}$$

$$= -175 + 239 = \underline{64°C};$$

Figure 3.2: Example for calculating critical temperature rise

Condition: critical $\Delta t \rightarrow$ actual $\Delta T^{(3)}$ + safety margin[3]

Test at neutral temperature $t_N = 17°C$

at the rail temperature T = 60°C

and at a speed maximum v = 100 km/h

actual T = 60 -17 = $\underline{43°C}$

Required safety margin for v = 100 km/h = 18°C

From this follows: 64 > 43 + 18

64 > 61

This means: condition is fulfilled - track is safe!

[1] The track lateral resistance is strength as a result of re-furbishing

[2] The track lateral resistance is strength as a result of the lift-off wave or vibration

[3] The actual temperature rise, actual ΔT, and the safety margin are more precisely defined in the paper by Professor Eisenmann "Track stability at high speeds", see Eisenbahningenieur (Railway Engineer) 25 (1974).

Assumptions:

Track form	:	UIC 60 -1667 - B 70 W
Radius	:	R = 500 m
Track length considered	:	1/4 circular segment
Neutral temperature	:	t_N = 20°C

As a result of refurbishing, the curve shifts inwards at lower temperatures by 5 cm on the average.

Desired: Length change ΔL
Falsification of neutral temperature

Figure 3.3: Example for calculating falsification of the neutral temperature

$$\Delta l = L_1 - L_2 ;$$

$$L_1 = \frac{2 \cdot R_1 \cdot \pi}{4} = \frac{2 \cdot 50000 \cdot \pi}{4} = 78539,815$$

$$L_2 = \frac{2 \cdot R_2 \cdot \pi}{4} = \frac{2 \cdot 49995 \cdot \pi}{4} = 78531,96$$

$$\Delta l = 785398,15 - 785319,60 = 78,6 \text{ mm}$$

From this follows: $\Delta t = \frac{\Delta l}{\alpha \cdot L} = \frac{78,5}{0,0000115 \cdot 500000} = 13,7 \sim \underline{14\,°C}$

thus the real neutral temperature is: 20 - 14 = 6°C

Condition: critical Δt > actual ΔT + safety margin

Test at the falsified neutral temperature

at a rail temperature of 60°C and

at a speed max v = 100 km/h

actual ΔT - 60 - 6 = 54°C

Required safety margin for v = 100 km/h = 18°C

From this follows:

64[1] < 54 + 18

64[1] < 72

This means: condition is not fulfilled

track is no longer safe!

[1]Critical Δt as in the example of Figure 85 8506 - 78 -78

"track lateral strength" and "equivalent moment of inertia".

For the rest, the German Federal Railway values its experience, with about 50,000 km continuously welded rail, more highly than tests and experiments. The credibility of such experiments is significantly relativized through the experimental conditions.

There likewise is no present intention of performing such experiments, since the correctness of the calculational assumptions has been fully confirmed in practice.

It should be noted that not a single track buckling occurred with the German Federal Railway during the year 1978.

References:

Raab: Stability Relationships with Continuously Welded Railroad Rails. Eisenbahntechnische Rundschau (Railway Engineering Survey), No. 8, 1960.

Raab/Biermann: On the Development of Continuously Welded Rails. Eisenbahntechnische Rundschau (Railway Engineering Survey), No. 11/12, 1958.

3.1.3 Experiments on Component Research

Experiments of the German Federal Railway are primarily intended for the quantitative determination of

- equivalent moment of inertia
- track lateral strength
- track longitudinal strength
- slip resistance.

Equivalent moment of inertia (panel rigidity)

In order to be able to make clear statements concerning the rigidity of the track panel in the horizontal direction, the track panel is approximately regarded as a frame, in which the rails are considered as chords and the ties as struts. For this purpose, the equivalent moment of inertia I of this frame is experimentally determined in a rail section without ballast, lying on rollers. The horizontal rigidity of the panels depends on the following quantities:

- tightening force of the fastening
- size of the rail/pad/plate contact surfaces
- elasticity of the fastening
- type of pads
- rigidity of the rails in the Y direction
- distance between ties.

The panel rigidity experiment must be performed for all track constructions which are intended to be used for continuously welded rails. For this experiment, a track panel is mounted similar to an individual support*. A load is stepwise applied to its center, and the associated bending is measured. The experiment is performed several times, in order to measure hysteresis. The sign of the force application is changed after every experiment. This requires resetting the presses.

Details and results are cited in Appendix 3.2.

*German: Einzelträger

Track Lateral Strength

The track lateral strength is the resistance of the track against internal and external forces acting perpendicular to the track axis. It is measured under static conditions, with a displacement of 2 mm. It depends on the form and condition of the track, and lies between 4 kilopound/cm (freshly refurbished wooden tie track K 49) and 15 kilopound/cm (stabilized concrete tie track B 70 W).

The normative displacement magnitude of 2 mm is based on experiments, according to which it was determined that the track lateral strength rises linearly with force up to a displacement of 2 - 3 mm. After this, however, it runs very degressively. With a displacement of 2 - 3 cm, it goes over into a constant, i.e. the track panel slides in the ballast bed without any increase of resistance. For safety considerations, and in agreement with the simplifying assumption of a constant track lateral strength an assumption basic to the theory, the strength at 2 mm displacement is regarded as normative.

The magnitude of the track lateral strength depends on

- track form
- ballast cross section
- ballast density (compaction level of the ballast)

The effect of track form on the track lateral strength is to be sought in the different static friction resistance, which is determined by the weight of track construction and by the surface characteristics (undersurface and lateral surface) of the ties. The number and form of the ties are less significant.

For a stabilized track, the track lateral strength is as follows:

Rail UIC 60 concrete tie B 70 = 14.1 kilopound/cm

Rail UIC 60 wooden tie = 11.5 kilopound/cm

Rail S 54 wooden tie = 11.0 kilopound/cm

The ballast cross section also affects the track lateral strength. The ballasting level of the inter-tie space (Figure 3.4) as well as the width of the ballast before the tie end (Figure 3.5) have a significant effect here.

Special significance must be ascribed to the ballast density, i.e. whether the ballast is decompacted or stabilized. In contrast to track forms and ballast cross section, the ballast is subject to more frequent change. Every maintenance of track and switches necessarily involves a decompaction of the ballast stones, which are wedged into one another through the operating load. This involves a significant reduction of the track lateral strength, to about 50 percent of its original value. This fact is taken account of as follows: After track operations, and when high temperatures are expected, the speed must be reduced for a short time, one to three days, depending on line load and on the shape of the ties. Details will be explained under Point 3.1.5.

Figure 3.4:

Effect of ballasting level of the inter-tie space

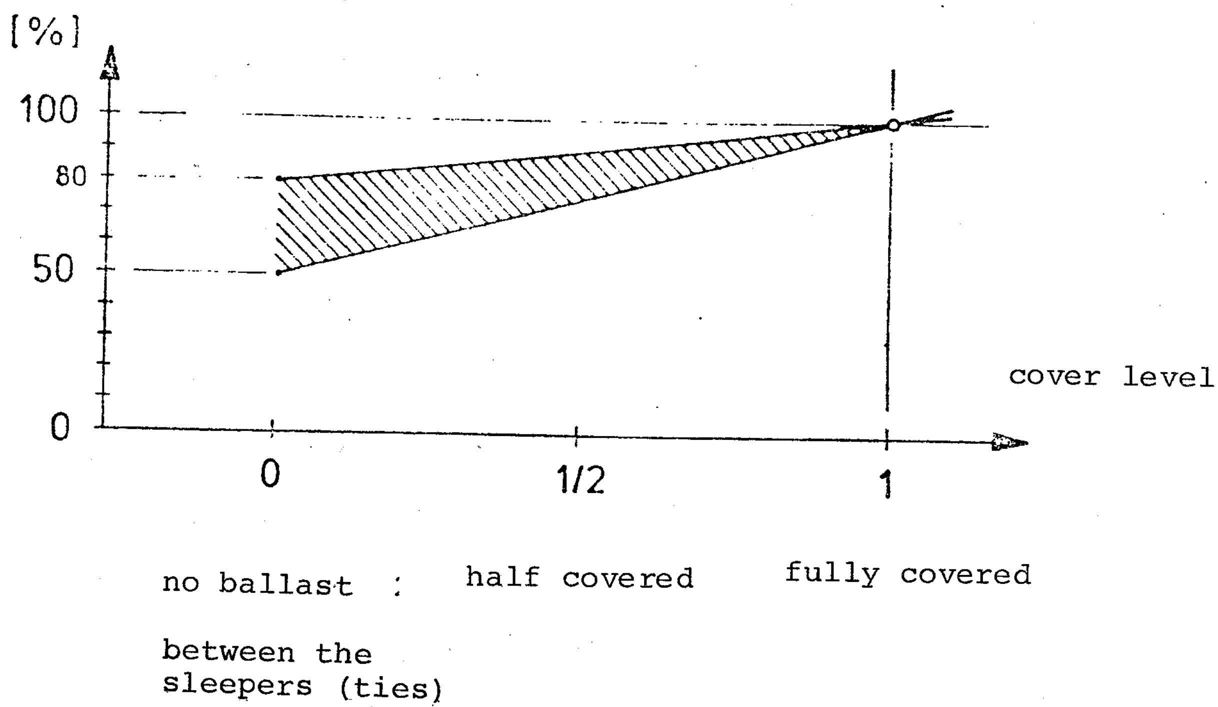
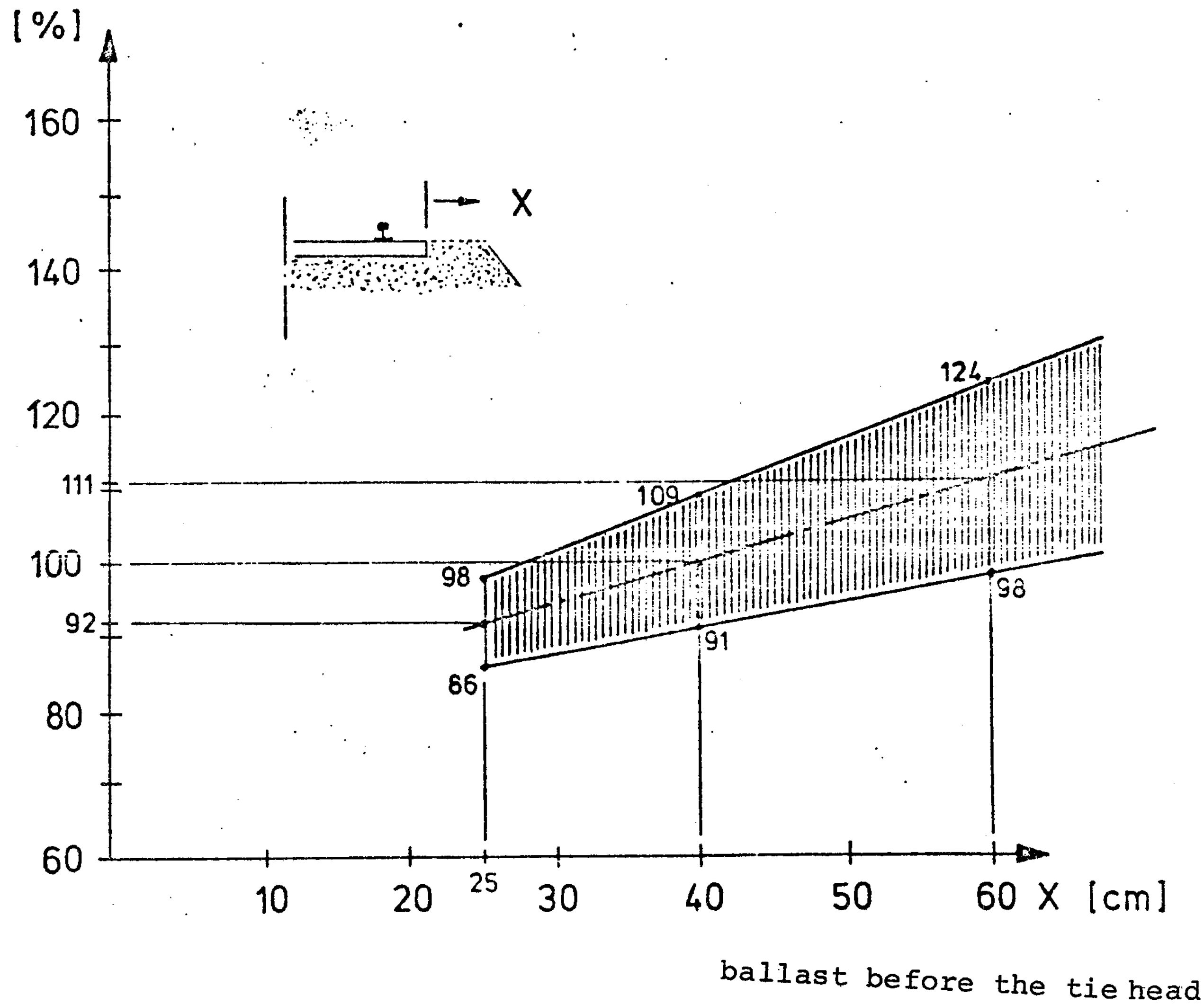


Figure 3.5 Increase of resistance in dependence on shoulder width



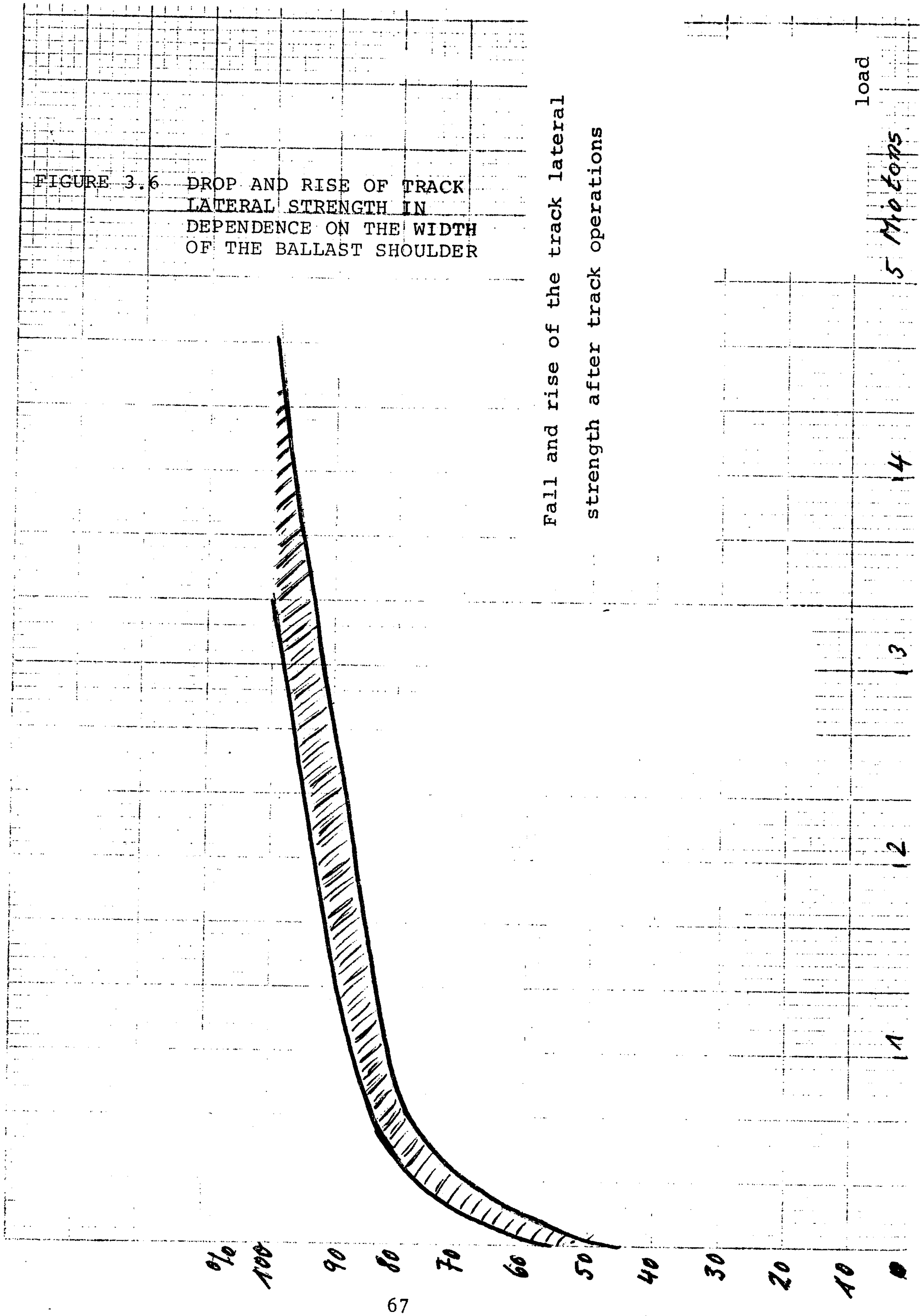
A substantially reduced track lateral strength also prevails in freshly renewed track. This is likewise taken into account by reducing the speed limit.

The track lateral strength of a maintained track and of a freshly renewed track can be regarded as equal at the time that train operation is resumed, for the purpose of evaluating track safety and for the purpose of calculating the speed limits. The loose ballast texture resulting from renewal receives a similar density as a freshly refurbished track, by means of the prescribed compaction of the inter-tie space.

The fall and rise of the track lateral strength after maintenance or after renewal, in dependence on track load (t), can be seen from Figure 3.6. The track lateral strength rises progressively, i.e. its greatest increase occurs during the first 100,000 tons track load. The strength value here rises from about 50 percent to about 65 percent of its final value. The data are shown in a distorted (logarithmic) scale. In the range from 3 million to 5 million tons track load, they only show an additional increase by about 5 percent - from 95 percent to 100 percent.

The force measured at an individual tie, with 2 mm displacement, is normative for determining the track lateral strength. This force must be measured on at least ten ties, and an average must be formed. During the measurement, care must be taken that the ballast cross section (ballasting) is correct and that the ballast is in the desired stabilized or unstabilized condition.

FIGURE 3.6 DROP AND RISE OF TRACK LATERAL STRENGTH IN DEPENDENCE ON THE WIDTH OF THE BALLAST SHOULDER



The measured value of a single tie must be divided by the distance between ties and must be multiplied by a reduction factor of 0.75. This factor takes care of the effect of neighboring ties with simultaneous displacement of a panel. The factor was determined by comparative studies on individual ties and by displacing track sections about 4 m long.

In order to perform the experiment, the tie is loosened from the rail fastening and is displaced with respect to the rail and the rail fastening. The German Federal Railway has developed a special device for this purpose. This device is specifically adjusted for the rail fastenings that are customary with the German Federal Railway. Plasser Company has developed a method to determine the track lateral strength by means of a track tamping machine. This method was tried out in conjunction with experiments performed by the Austrian Federal Railway.

Slip Resistance and Longitudinal Resistance

The slip resistance between the rail and the tie is especially significant with continuously welded rails. When a rail fracture occurs at low rail temperatures, an excessive fracture gap should thus be avoided. The slip resistance must here be greater than the longitudinal resistance of the ties in the ballast bed. The slip resistance should be 9-10 kN per support point. It should be about twice as great as the longitudinal strength of the ties; this takes into account the gradual decline of slip resistance as a consequence of reduced tightening forces in the rail fastening, as well as of a decline caused by the dynamic load effect. With the German Federal Railway, the longitudinal strength, relative to half the tie, amounts to 5-7 kN.

Figure 3.7 shows the experimental arrangement for determining the slip and lateral torsional resistance. The lateral torsional resistance permits an indirect judgment concerning the expected panel rigidity of the track panel.

Longitudinal forces from traffic

The additional longitudinal forces, which are caused in the rail by starting and braking, are not separately considered. They are covered by the safety temperature (see Point 3.1.1). Up to now, practical experience has confirmed the correctness of this procedure.

However, it was determined that the high rail can migrate in the case of very long curved tracks lying in a gradient. This effect is ascribed to the action of the starting and braking forces. Since such conditions are very rare in the line network of the German Federal Railway, this phenomenon is dealt with through local measures.

3.1.4 Other Measures Relevant to Continuously Welded Rails

As a practical application of research results, and on the basis of practical experience, the German Federal Railway permits all slip-proof track types - except only spiked tracks - to be continuously welded. Only with small radii of curvature must auxiliary measures be taken: either the ballast width before the head must be enlarged, or safety caps must be installed,

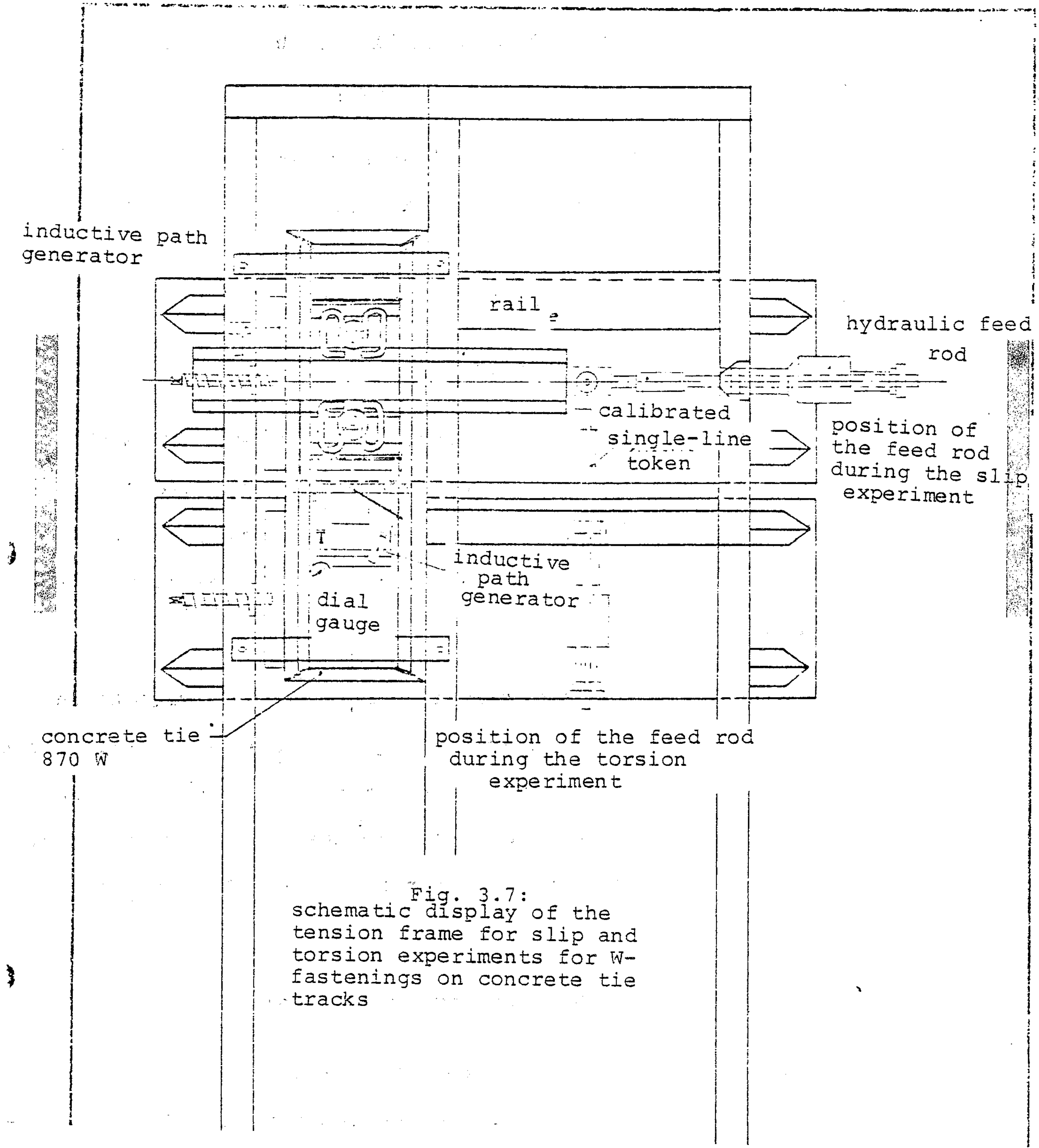


Fig. 3.7:
schematic display of the tension frame for slip and torsion experiments for W-fastenings on concrete tie tracks

FIGURE 3.8 CONDITIONS PERMITTING THE WELDING OF TRACK TYPES (GERMAN FEDERAL RAILWAY)

Oberbau 10		Regelbettungsquerschnitt (AzObv 6) 1	Verbreiterung des Regelbettungsquerschnittes an der Bogenaußenseite von 40 cm auf 50 cm 3	Regelbettungsquerschnitt und Sicherungskappen an den Schwellenenden der Bogeninnenseite an 4			Verbreiterung des Regelbettungsquerschnittes an der Bogenaußenseite von 40 cm auf 50 cm und Sicherungskappen an den Schwellenenden der Bogeninnenseite an jeder 3. Schwelle 8	Regelbettungsquerschnitt und Leitschienenoberbau 9
Schwellenart 11	Schienenprofil 12			jeder 5 3. Schwelle	jeder 6 2. Schwelle	jeder 7 Schwelle		
1	2	3	4	5	6	7	8	9
Holzschwellen 13	S 49 14 und leichtere Schienenprofile	500 m	350 m	350 m	290 m	190 m	260 m	220 m
	S 54	550 m	350 m	350 m	290 m	190 m	260 m	220 m
	UIC 60	600 m	450 m	450 m	360 m	250 m	260 m	–
Betonschwellen mit einer Länge L = 2,40 m 15	alle 16 Schienenprofile	400 m	290 m	290 m	230 m	190 m	230 m	–
Betonschwellen mit einer Länge L = 2,60 m (B 70) 17	S 54 und UIC 60 18	350 m	230 m	230 m	200 m	180 m	210 m	–
Stahlschwellen mit großen Kappen 19	alle 20 Schienenprofile	300 m	250 m	–	–	–	–	170 m

Bemerkung:
In Nebengleisen dürfen Bögen mit R = 180 m ohne Berücksichtigung der Schwellen- und Schienenbefestigungsart durchgehend geschweißt werden. Für Anschlußgleise gilt die Halbmesserbegrenzung nicht.
21

Note: see translation key on page 80.

1 TABLE 4. CONDITIONS WHICH PERMIT THE WELDING OF RAILS IN CURVES

2 standard ballast cross section (AzObv 6)

3 widening the standard ballast cross section at the high rail, from 40 cm to 50 cm

4 standard ballast cross section and safety caps at the high ends of the low rail, at

5 every third tie

6 every second tie

7 every tie

8 widening the standard ballast cross section at the high rail, from 40 cm to 50 cm, and safety caps at the high ends of the low rail at every third tie

9 standard ballast cross section and guide rail

10 track

11 type of tie

12 rail profile

13 wooden tie

14 S49 and lighter rail profiles

15 concrete ties with a length L = 2.40 m

16 all rail profiles

17 concrete ties with a length L = 2.60 m (B 70)

18 S54 and UIC 60

19 steel ties with large caps

20 all rail profiles

21 Remark: In secondary tracks, curves with R = 180 m may be continuously welded without considering the tie and rail fastening type. The radius limitation must not apply to branch lines.

1 fabrication of continuous rail
2 type of rail fastening
3 first order
4 all trucks
5 second order
6 continuous main tracks and severely stressed tracks in yards
7 other primary and secondary tracks
8 third order
9 all tracks
10 new concrete
11 old concrete
12 new wood
13 old wood
14 new steel
15 old steel
16 new concrete
17 old concrete
18
19 new wood
20 old wood
21 old steel
22 old concrete
23 new wood Gr 4/5
24 old wood
25 old steel
26 old concrete
27 old wood
28 old steel
29 may be continuously welded

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24
Art der Schienenbefestigung	1. Ordnung									2. Ordnung											3. Ordnung		
	Alle Gleise									Durchgehende Hauptgleise und stark beanspruchte Gleise in Rangierbahnhöfen							Sonstige Hauptgleise und Nebengleise				Alle Gleise		
	Schwellenart																						
	Beton neu	Beton alt	Holz neu Gr 1	Holz neu Gr 2	Holz neu Gr 4	Holz alt Gr 1	Holz alt Gr 2	Stahl neu	Stahl alt	Beton neu	Beton alt	Holz neu Gr 1	Holz neu Gr 2	Holz neu Gr 4	Holz alt	Stahl alt	Beton alt	Holz neu Gr 4/5	Holz alt	Stahl alt	Beton alt	Holz alt	Stahl alt
K																							
W																							
Hf Dna 4/110																							
A oder L																							
Länderformen																							

Durchgehend verschweißbar

Durchgehend verschweißbar mit Wanderschutzklemmen an jeder 5. Schwelle mit jeweils wechselnder Wirkrichtung

30 May be continuously welded, with rail anchor clips at every fifth tie, always with alternating effective direction

Figure 3.9: CONDITIONS WHICH PERMIT THE WELDING OF TRACK TYPES

installed in order to increase the track lateral strength. Figures 3.8 and 3.9 provide information regarding particulars.

Another exception from the order for the continuous welding of all rails is constituted by rails on unevenly resilient subsoil (e.g. subsidence areas) and rails in gravel and sand ballast.

3.1.5 Fabrication and Maintenance of Continuously Welded Rails

With the German Federal Railway, the rails are brought to their installation site undrilled and in lengths of 120 m. The manufacturer delivers 30 m long rails, and these are first connected in a welding plant by means of flash butt welding. The rails must be laid so that the branding marks of all the rails lie on the same side. This maximizes the probability of encountering an asymmetry directed in the same sense. If rails are installed elsewhere in a second location without further refurbishing, they must first be checked for the presence of defects. This is done by means of an ultrasonic rail testing train or by means of a manual ultrasonic device. Existing defects must be eliminated before re-installation.

During layout, the following gaps must, as a rule, be provided:

Rail temperature	Laying gap
less than 10°C	20 mm
from 10°C to 19°C	15 mm
from 20°C to 25°C	10 mm
higher than 25°C	0 mm

If the working procedure makes it impossible to adhere to these rail temperatures, the rails must be brought into their design position before welding. This is done by pulling them or, in the case of small misplacements, by cutting the rail ends. The laying temperature of the rails must first be measured and must be entered into a printed form. All the other data relevant to the welding process are also recorded in this form. These records are very important for all operations that must later be performed on the track. The temperature is measured at the side of the web that is turned away from the sun. This measurement is performed by means of a rail contact thermometer. The web must first be cleaned from rust. The rail contact thermometers must from time to time be checked for the reliability of their readings.

Now the rails are fastened to each tie. The joints are suitably secured with fish plates and screw clamps. The screw clamps must be secured against spontaneous loosening. Such joints can be traversed at a speed up to 120 kmh. Of course, it is here presupposed that the track is perfectly laid as regards alignment and elevation and that the track quality corresponds to the markers, that the track is well filled with ballast, and that the ballast has been compacted with at least two tamping runs.

If this rail is now to be continuously welded, the rails must be elongated by artificially applying heat, in such a manner as if they had been laid at the so-called design temperature. This step may be omitted if the welding operation is performed at a time when the external temperature corresponds to the design temperature. As a rule, continuous welding is no longer allowable at temperatures below + 3°C. It is understandable that the objective must be to keep the rail stress-free and at an external temperature which corresponds approximately to the average value between the maximum temperature in summer and the lowest temperature in winter. With such a mode of procedure, the pressure and temperature stress will be about equally large and in any case not excessive. However, compressive forces in the rail are more disadvantageous than tensile stresses in rails which in any case have high strength. Consequently, it is advisable to raise the average temperature slightly by + 5°C. Starting from an assumed maximum rail temperature of + 60°C and an assumed minimum temperature of - 30°C, the calculated design temperature is thus calculated as follows, as an example:

$$-30 + ((60 + 30) : 2) + 5 = 20°C$$

To equalize path deviations and inaccuracies, it is now permitted to perform the welding and tightening operations with an actual tolerance of + or - 3°C relative to the previously calculated value. The figure obtained in this manner is called the neutral temperature. For the reasons mentioned above, one should try to use the upper range as much as possible.

With the example above, the neutral temperature t_R is 23°C. The length change of the rail at the control points at 60 m and at its end is now calculated from the difference between the neutral temperature T_N and the initial temperature T_i. The formula for this is $\Delta \ell = \alpha \times \Delta t \times L$. α is the heat expansion coefficient of steel. Its value is 0.0000115. The distance of the control points should be at most 60 m on tangent tracks and at most 30 m on curved tracks.

In order to guarantee an unhindered length change during neutralization, the fastening means and the fish plates must be loosened, and the rail must be struck with rubber hammers while the temperature is acting upon it. Experiments to use vibrating machines in order to overcome the friction resistance were initially unsucessful. Special care must be taken that the rails are not prevented from expanding by jammed track fastenings, non-perpendicular ties, welding beads, rail anchor clips and grounding clips, or excessively small joint openings. The expansion must therefore absolutely be controlled at points identified by benchmarks.

When the rail has reached its calculated elongation, it must immediately be solidly tightened against the tie. Care must be taken that the application of heat is not stopped too early, especially at relatively low temperatures. In this case, the rail would again shorten before being tightened. On both sides of the final welding operation, the rails must be tightened along a length of three tie spacings, only after the weld has cooled.

The rail ends are now aligned exactly, with a ruler, according to the gage line, and the aluminothermic weld is then made according to the relevant directives.

Incidentally, in the case of rain and wind, the weld must be covered with thermal shields up to a temperature of 300°C. The purpose of this is to protect the weld from too sudden cooling, which would result in martensitic structures.

Subsequently, the above-mentioned three ties to the left and right of the weld are tightened, follow-up tamping is performed in exceptional cases, and the weld is ground. This must be done very carefully. A carelessly ground welding point would be acoustically perceptible during travel, both outside and inside the vehicle. It would also be recorded on the track measurement chart. Neither should an elevation remain nor should a recess be ground in. For this reason, a certain skill is necessary, and the work must be checked several times with a long steel ruler. Deviations in height are permissible up to + 0.2 mm and -0.4 mm, and deviations on the gage line side are permissible up to -0.3 mm but without a plus tolerance.

Defects must be neutralized within these tolerances with 1:500 ramps, and, in the case of tracks for speeds greater than 160 kmh, with 1:1000 ramps.

It should also be mentioned that special forms make it easily possible to weld together rails of different height. Such rails must be aligned with respect to track surface and gage line.

Switches are to be continuously welded with the rail. When switches are laid in welded tracks, lay-out gaps are to be provided at the beginning and end of the switch, according to a special table. The welding operations can begin after the first tamping. The final welds, however, may be made only after the last tamping. The like holds for the welding of switch tongues.

3.2 Track Stabiity of the Loaded Track

The track stability of the loaded track has been investigated by the National French Railways in very extensive experiments. By means of two different experimental methods, the track stability of the track was researched under the action of horizontal and vertical forces. On the one hand, horizontal forces from a neighboring track were applied to the axle of an experimental vehicle, and the resulting track deformation was measured. On the other hand, the track was directly loaded with the movable center axle of an experimental car.

The ΣY force was determined as the relevant force quantity. It was measured with a track panel displacement of 2 mm. All experiments were performed on a freshly tamped track, since such a track has the least track lateral strength.

The following law was determined by the experiments:

$$\Sigma Y_{critical} = 10 + 2Q/3 \qquad [kN]$$

For practical application, a sufficient safety margin is necessary with respect to this critical value. Consequently, the permissible summed Y force was defined as

$$\Sigma Y \leq 0.85\ (10 + 2Q/3) \quad [kN],$$

where the axle load is to be used for 2Q. The SNCF (French Railway) has determined that the effect of rail shape, rail temperature, and speed is small in the range of 0 - 60 km per hour.

At this time, within the framework of the ORE, experiments are being undertaken to ascertain whether the speed ($V \geqslant 60$ km/h) and the Y forces exerted by neighboring axles have an effect on the limit value. The German Federal Railway also intends to perform experiments relative to its track construction, so as to be able to estimate the safety margin existing at high speeds. Up to now, unrestricted application of this limit value has proven itself internationally. Track displacements by excessive lateral forces, exerted by the vehicle on the track, have not occurred, since the permitted lateral forces of all vehicles have been limited according to this standard.

For concrete tie tracks, the limit tolerance is somewhat higher. The limit tolerance is also considerably higher for stabilized tracks, but this is not taken into account for safety reasons. The relevant ΣY force is defined as that force which acts on the track for a length of at least 2 m.

References: Prud'homme, Yanin, The stability of track laid with continuously welded rails, Monthly Journal of the International Railway Congress Association, No. 6 and No. 7-8, 1969.

3.3 Summary and Conclusions

The theoretical foundations for calculating continuously welded rails are provided by the studies of Professor Meier/ Professor Eisenmann. These studies were performed in collaboration with the German Federal Railway. The most important influential parameters of this calculation are the panel rigidity and the track lateral strength of the track panel. Both of these parameters can be determined only experimentally. In numerous experiments, the German Federal Railway has determined these parameters for its various track constructions and maintenance conditions.

In applying these theoretical-experimental principles, the German Federal Railway has worked out detailed directives for the fabrication and maintenance of continuously welded rails. These directives have been used for years. Practical experience has shown that the track stability of the rail is guaranteed - as long as the prescribed guidelines are adhered to. Consequently, the theoretical foundations are also indirectly confirmed.

With reference to American conditions, the following conclusions are to be drawn, in the opinion of the authors:

1. Continuously welded rails should be introduced to a significantly greater extent, at least in heavily trafficked lines. The reason for this is that their economic advantages are obvious, both in the short term and especially also in the long term.

2. Wood, concrete or steel ties can be used for continuously welded rails. However, it is of decisive significance that a slip-proof and torsion-proof rail fastening be used. It is recommended that appropriate experiments be performed with

various rail fastenings, which are available on the market and which have already proven themselves. However, the results of such experiments must not be considered in isolation from the type of tie and their track lateral strengths. In every case, care must be taken that the track panel has adequate ballast.

3. Uniform guidelines - for example, corresponding to the directives or regulations of the European railways - can in the short term probably not be implemented in the area of American railways. However, with reference to safety requirements, the authors deem it sufficient and also practically possible that the railway companies be obliged to set up and to introduce

- enforceable computational principles
- production and maintenance guidelines, as well as
- verifiable records concerning inspection and maintenance measures

for continuously welded rails.

4. The production and maintenance of continuously welded rails requires qualified supervisory personnel, who are familiar with the relevant guidelines. It is recommended that appropriately trained supervisory personnel be properly educated and be continuously retained.

5. Gradual displacement of the loaded track could be the starting point of track buckling. In order to avoid this condition, the guide forces ΣY, which appear in the wheel/rail area, must stand in a certain ratio to the vertical axle

forces 2Q (Q = wheel force).

For European conditions, the following relation holds

$$\Sigma Y \leq 0.85\ (10 + 2Q/3) \qquad [kN]$$

It is recommended that this relationship be initially taken over also for American conditions and that the factor 0.85 be verified by experiments.

This provides a standard for the permissible ratio of vertical force Q to horizontal force Y under operating loads.

The experiments of the SNCF could not determine any effect of temperature-conditioned longitudinal forces on track stability.

4. Special Track Problems

While treating the relationships presented above, a series of further questions appeared. Detailed remarks concerning these questions will now be made from the perspective of a European railway.

The authors have inspected the design for the SAFE track design. They discussed this draft with the responsible gentlemen and they transmit their relevant experience.

The authors concerned themselves with the problem area of the industry impact assessment. They learned that a coherent presentation of track maintenance and track inspection methods of the German Federal Railway, with special emphasis on maintenance costs, is of interest.

Thereupon a survey is given of studies concerning the reduction of maintenance costs, studies which have been attacked or are being planned by the German Federal Railway. Research activities for improved inspection and evaluation methods here occupy the foreground.

Finally, a survey is given of the German Railway's experience with concrete ties and the associating fasteners.

4.1 Recommendations and Remarks concerning the SAFE Test Program

The idea for the design of a reference track, which should serve for the acceptance of vehicles was critically inspected.

Remarks are made below, concerning the individual segments of the design. References are given to the mode of procedure of the German Federal Railway.

Concerning 3.0

The German Federal Railway does not have any special experimental tracks for testing and accepting vehicles. The running behavior of vehicles is tested within the framework of long runs on tangent and curved track sections, which contain the most varied line layouts and track forms. The condition of the operating tracks chosen for testing varies from excellently maintained track to tracks where a lot of maintenance or track renewal are directly imminent.

During the long runs, the most varied track forms, track geometry conditions (superelevation, radius, grade, ramps), track quality defects, and maintenance conditions are encountered. Consequently, the results of measurement are then regarded as representative. The individual data measured during these long runs, and the average values calculated therefrom, must satisfy certain criteria. These criteria have already been discussed under Point 1.

Experience is therefore not available with the installation and maintenance of track quality and track geometry defects, which have been built in for experimental purposes.

Concerning 4.0

With the German Federal Railway, the worn out rail - with the exception of lateral rail head wear - does not differ from a new rail, as regards the geometry of the rail surface. The reason for this is that the contact geometry of the wheel and rail profiles has been optimized with respect to wear. Rail wear leads only to an enlargement of the gage and to consequences on running dynamics which result therefrom. In the view of the authors, the effect of a worn rail can therefore be simulated by building new tracks with a larger gage.

According to international acceptance conditions for freight cars, the vehicle being tested must run as the last car. In particular, it must run loaded as well as empty. The reason for this is that the largest forces are always measured at the axles of the last car.

Concerning 5.0

To ascertain running behavior at high speeds, it is deemed necessary that the geometry of the rail profile and of the wheel profile be additionally recorded, so that the equivalent conicity can be determined. The equivalent conicity depends on the wheel profile, rail profile, cant and gage.

Reference: ORE Report C 116.

Concerning 6.1

A track reaches an essentially stabilized state at the earliest after about one million gross ton load. For this reason, the authors regard the following sequence suitable for preparing the experiments:

1. Track fabrication, including the necessary tamping and alignment work
2. One million gross ton load
3. Retamping and realignment
4. One million gross ton load
5. Continue with point 3 of the design

Furthermore, the authors propose the following schedule for measuring the track geometry

1. measurement before the beginning of the experiments
2. measurement after one million ton load
3. measurement after five million gross tons following the

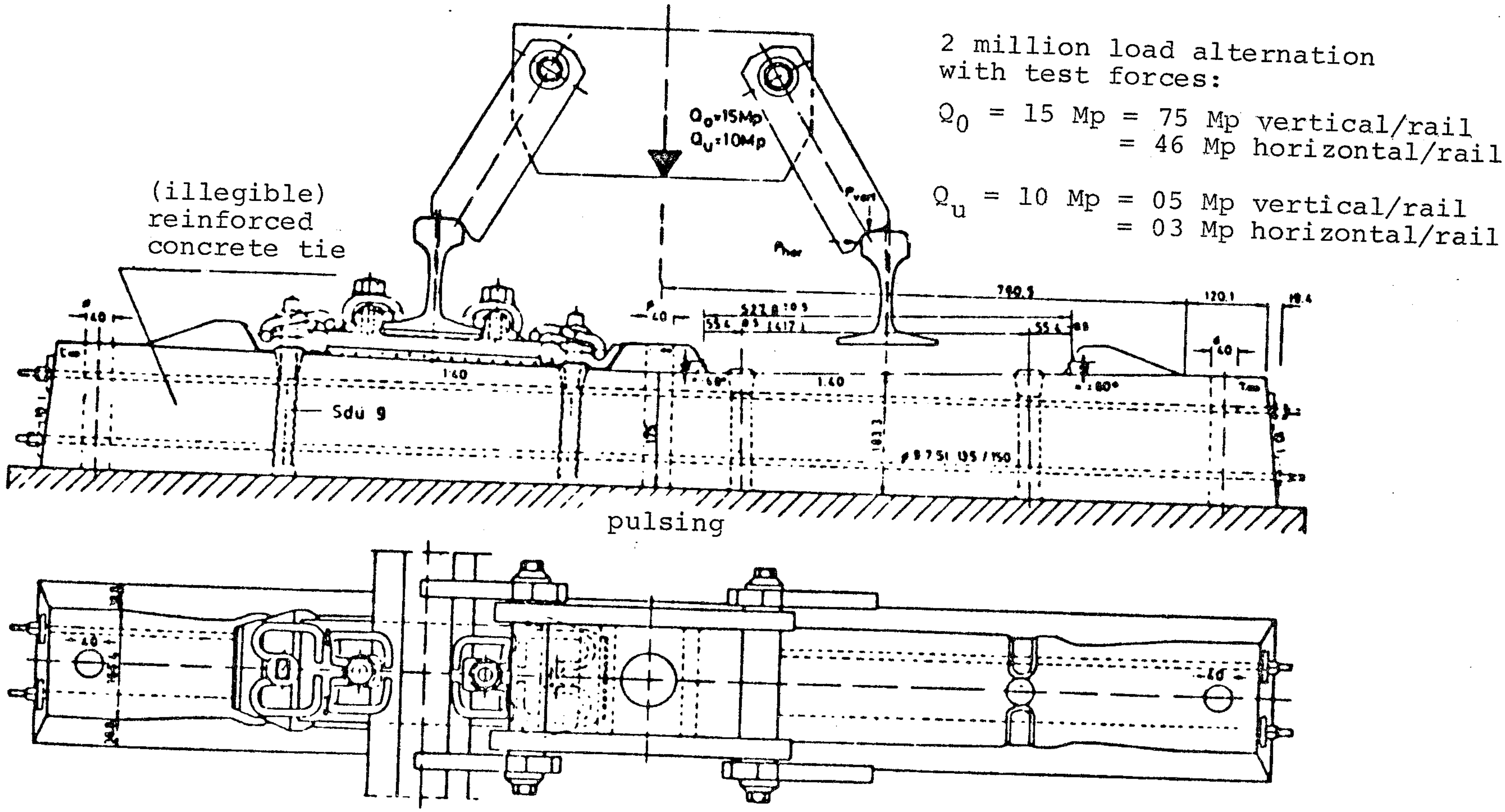
beginning of the experiment, and then after every 25 million gross tons load, but at least before and after termination of every experimental series.
In this way, it is possible to document a change of track quality.

Concerning 6.2

The authors advise against varying the ballasting modulus by means of various elastic pads under the rails or under the plates. This can cause the forces between the wheel and rail to be falsified, since the effect of the double elastic rail fastening is simultaneously altered. With a strongly elastic pad, the rails and ties are in addition turned and tilted in an uncontrolled manner during the tamping and alignment process.

In view of the special requirements which are imposed on the rail fastening for the SAFE track (can be regulated laterally and in elevation), the authors consider it absolutely necessary that this fastening be previously tested in a suitable test device (fastener testing equipment).
An appropriate test device is shown in Figure 4.0.

Figure 4.0 TEST ARRANGEMENT FOR THE FASTENER TESTING EQUIPMENT



It is feared that the rail fastening shown in Figure 8, page 17, is not adequate for the stresses, in view of the shear stress on the tie screw.

As regards Figure 7, page 16, it should be remarked that the textile mat should be installed - if at all - between the sub-soil and the sub-ballast.

The thickness of ballast under the tie should not be "at least 12 inches", but this dimension should essentially be held constant, so that the settling differences resulting from different ballast thicknesses are kept small. The like also holds for the thickness of the sub-ballast.

The requirements for sub-ballast should be defined, in order to prevent the utilization of unsuitable materials.

Concerning 7.0

In order to specify critical positions along the track, it is urged that these positions be determined by means of vehicle reaction measurements.

Concerning 7.1

For wayside measurements, the measurement base should extend to at least 5 m length, i.e. about 8 ties, in order to prevent effects from different tie mounting conditions (tamping density, looseness).

The parameters of wayside measurements should also include the time duration and intensity of precipitation. These factors have an effect on:

- the rail, since the forces between the wheel and the rail are greater when the rail surface is dry than when the rail surface is wet,
- the ballast, since heavy rain falls trigger shifts in the ballast texture and in the sub-soil and can thereby cause considerable track changes.

As regards Figure 11, page 22, it should further be remarked that track quality should also be checked by means of fixed points (every 20 m), so that absolute settling and alignment changes of the track can be determined.

As regards Figure 12, page 23, it should be remarked that the permanent vertical and the dynamical tie motions should also be included. Furthermore, lateral tie motion should always be measured at the individual tie.

4.2 Track Maintenance

4.2.1 The Track Maintenance Methods of the German Federal Railway

For the delivery of track materials, the German Federal Railway has in principle set up delivery conditions, which specify the quality requirements, the warranty period, and the scope of tests to be undertaken by the manufacturer. The delivery conditions were set up in collaboration with industry. In particular, the following exist:

Delivery conditions for rails (Appendix 4.1)

Delivery conditions for concrete ties (Appendix 4.2)

Delivery conditions for wooden ties

Delivery conditions for ballast (Appendix 4.3)

Delivery conditions for rail fastenings and

Delivery conditions for sub-ballast.

The delivery conditions for rails are valid internationally.

The quality requirements applicable to material deliveries are relatively strict. Consequently, the material put into the track damages relatively rarely. For example, in accord with acceptance conditions for rails, all rails are already tested ultrasonically in the rolling mill. Welds produced in the welding plant connect individual rails with a length of 30 m, into rails with a length of 120 m. These welds are also tested ultrasonically in the plant. By means of improved production methods and increased quality control, it has been possible to increase the warranty period for concrete ties from three years to five years, without a price increase.

With the German Federal Railway, track maintenance is decisively influenced by the systematic refurbishment of used track materials. Within the framework of well-considered cascading for rails, ties, and switches, new materials are installed only in heavily loaded tracks. However, at the end of a previously known service time - if necessary after refurbishing - they are reused in less heavily loaded tracks. Consequently, a long overall lifetime results. For this reason, the German Federal Railway makes a distinction between renewal (new materials) and replacement (refurbished materials) for ties, switches, and rails.

The basic idea of track maintenance consists of laying the longest possible track sections with new or refurbished materials, depending on the line's load. It follows from this that relatively little expense is required for running track maintenance. This expense is essentially limited to mechanical tamping and alignment of the track. In this way, a uniform, although gradually deteriorating track quality is secured over the longer sections.

This track maintenance method also favors the operation on these lines. The high line load requires that maintenance work requiring much time be known at least two years in advance, in order to give operations the opportunity of reducing the number of trains on these lines appropriately, by rerouting them during the construction work. High track quality is also required because the passage of numerous passenger trains, which run according to a rigid schedule, does not permit a reduction of speed resulting from poor track quality.

The application of highly mechanized methods for the track renewal has succeeded in clearly reducing the expense per kilometer of track renewal during recent years (Figure 4.7).

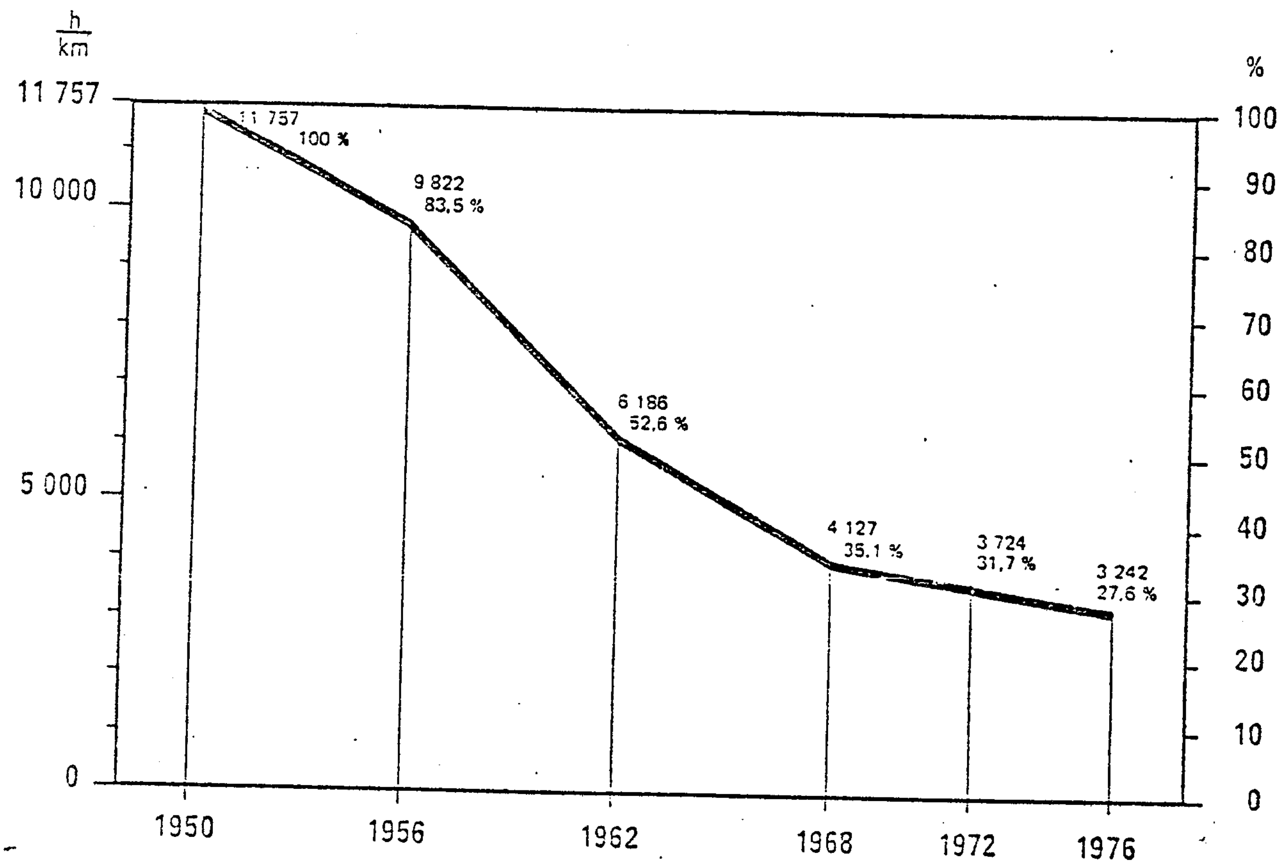
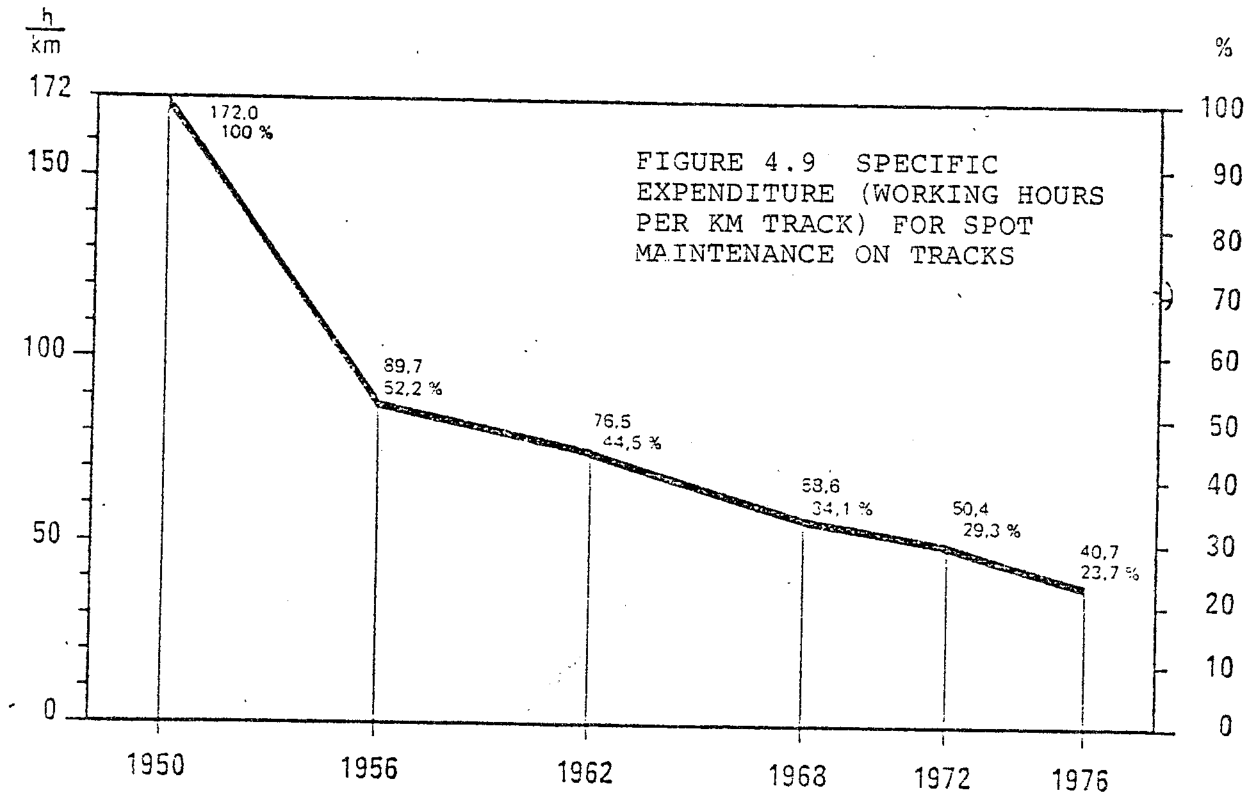
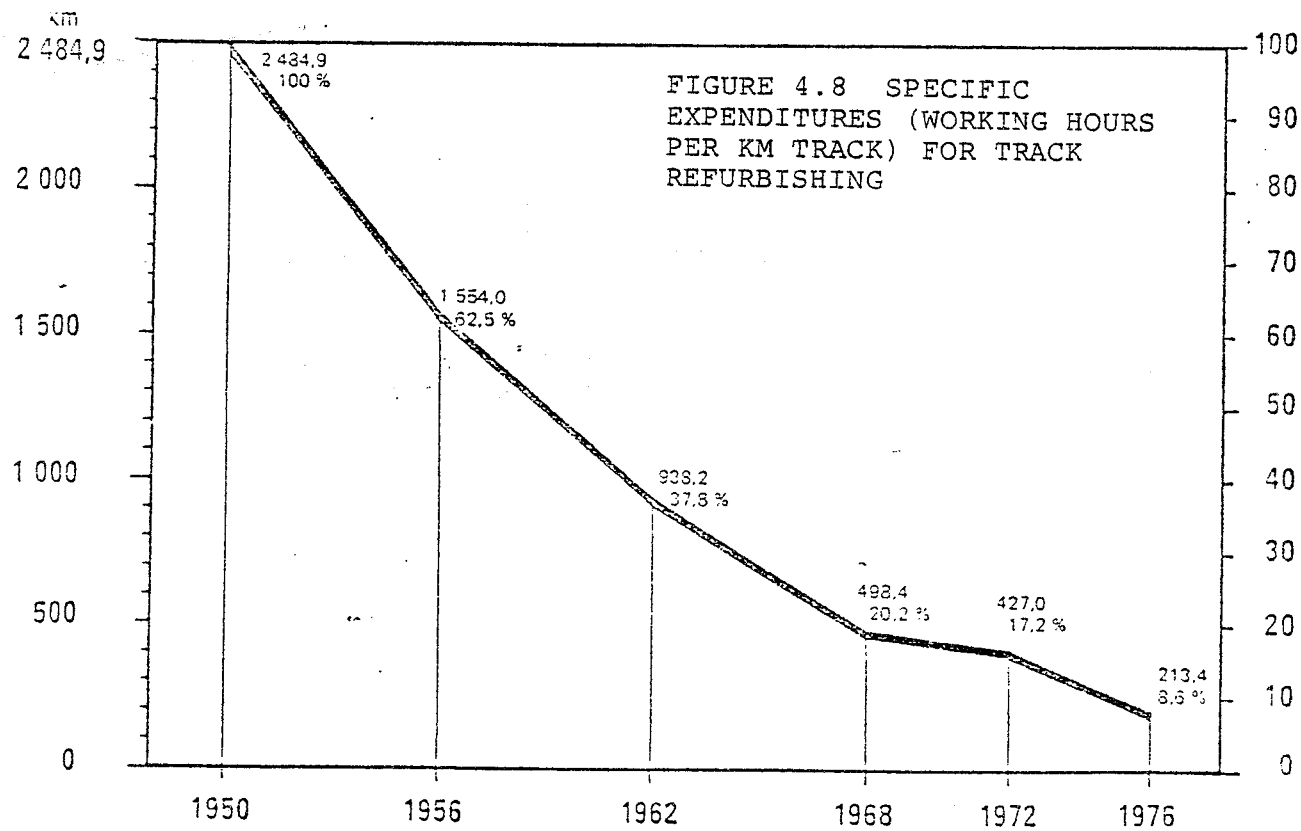


FIGURE 4.7 Specific expenditures (working unit-hours per km) for track renewal with concrete ties



Specific expenditures (working unit-hours per km) for spot track maintenance



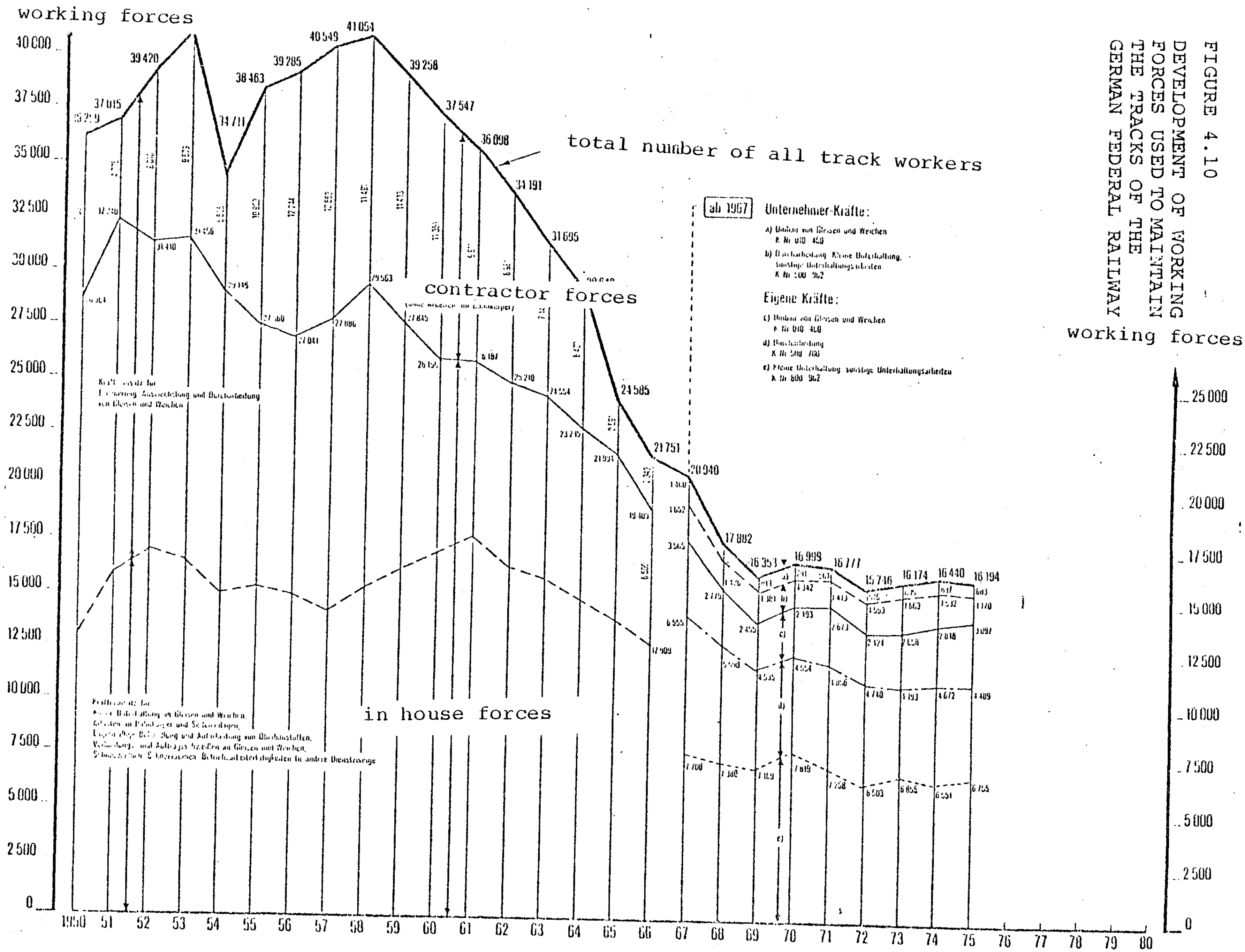
Specific expenditures (working unit-hours per km) for maintenance

Track maintenance (tamping and alignment) is carried out only with modern combined tamping and aligning machines. Their capacity is set at 15 percent from the German Federal Railway and at 85 percent from commercial track enterprises. Maintenance has been essentially mechanized, the working speed of machines has increased more and more, and the working quality of machines has reached a high state. Because of all these factors, the expenditures for maintenance of tracks and switches (Figure 4.8) would be considerably reduced.

On the other hand, there is also some spot maintenance which serves only to remove individual defects. The extent of spot maintenance at this time is 40 hours per kilometer of track per year. This is continuously being reduced. In particular, it has drastically declined because of the introduction of continuously welded rails (Figure 4.9).

In all, the German Federal Railway could realize considerable reductions of expenditure in virtue of its maintenance systematics. Total expenditure, especially the deployment of working forces (Figure 4.10), could be substantially reduced while output was increased. Taking into account the wage and price increases, expenditures for total track maintenance of the German Federal Railway today are 20 percent less than during the 1967 comparison year.

FIGURE 4.10
DEVELOPMENT OF WORKING FORCES USED TO MAINTAIN THE TRACKS OF THE GERMAN FEDERAL RAILWAY



Development of working force deployment for track maintenance

96

4.2.2 Inspection of tracks and switches of the German Federal Railway

The system for inspecting tracks and switches serves both for safety control as well as for status control, to specify the required maintenance measures. It comprises:

- local monitoring by visual inspection
- geometric measurements
- acceleration and force measurements
- ultrasonic measurements and
- rail surface measurements.

4.2.2.1 Local Inspection

The German Federal Railway has arranged about 2100 areas for the responsible inspection of track conditions. Each area on the average comprises 31.8 km track and switches. It is organizationally subject to a local service office, which is responsible for maintaining construction installations. Each of these service agencies comprises about 20 inspection areas. Track conditions are locally inspected by manual measurements, depending on need, as well as by runs on the cabin of locomotives and by runs in the last car of passenger trains. All inspection activities, which are undertaken by technical personnel, are retained in special records. With high-speed tracks (above 120 kmh), the turnaround time is eight weeks; with very high speed tracks (above 160 kmh), it is four weeks.

Furthermore, primary tracks are checked on the average once a week by track maintenance personnel.

4.2.2.2 Geometric Measurements

Besides wayside inspection, the German Federal Railway performs systematic measurements of track geometry. These measurements are evaluated.

Large Inspection Car

The German Federal Railway has for 20 years owned a large track inspection car. This unit is deployed over the entire network and traverses about 40,000 measurement kilometers annually. It is pulled or pushed by a locomotive. Its purpose is to provide a precise overview of the state of the primary lines of the German Federal Railway network and to secure comparability among individual regions - see small inspection car. All primary lines are recorded once a year, and especially important sections, twice a year. All secondary lines are recorded every two to four years. The lateral alignment, longitudinal section, and cross-level of the two rails, as well as gage and track warping are continuously measured and recorded. Besides line kilometers, construction works - bridges, tunnels, and grade crossings - are recorded by an observer and marked on the measurement chart. The maximum measurement speed is 90 kmh. The measured data are acquired, transmitted, and recorded mechanically.

The measurement chart is evaluated by a specialist during the measurement run, and is provided with maintenance directives. In order to obtain qualitative information concerning the stability of track quality and the working quality of maintenance measures, the following data are subsequently entered on the measurement chart by the local service agency:

- track construction and age of track materials
- rolling mill and welding plant of the rails
- last automatic tamping and alignment work, specifying the construction type of the tamping machine and the company performing the work
- radius of curvature (design value)
- superelevation and length of superelevation ramps (design value).

The results from evaluating a measurement chart prepared in this fashion form a basis for evaluating the necessity of maintenance measures. An explanation is available for reading and evaluating the measurement chart of the large inspection car. This document contains a catalog of typical geometric defects and specifies the maintenance measures necessary to remove these defects.

It is intended to build a new inspection car. All track geometry parameters will then be measured essentially contact-free.

Present work with the large track geometry inspection car is characterized by its pragmatic mode of procedure. During the course of about 20 years, it has been possible to provide each individual engineer, technician, or foreman, occupied with track maintenance, with a qualitative value scale regarding track geometry.

Small Inspection Car

Besides the large inspection car, the German Federal Railway has available five small inspection cars for regional use. These cars have the following purposes:

- measuring the track geometry before the performance of maintenance work
- acceptance measurements after the performance of work
- status measurements on tracks which are not traversed by the large inspection car.

These vehicles have only been procured a few years ago. Besides the two supporting axles with nine tons axle weight each, each inspection car has three more measurement axles. The measurement signals are electrically transmitted for recording. This makes it possible to use a classification unit and to define and evaluate track conditions by quality factors (Wz).

Initially, quality factors were worked out separately only for the parameters: track surface, warping, and alignment. It is intended to combine these into one quality factor. The process is still being developed. The following parameters are sensed by the measurement axles and are recorded in analog form:

track quality of the right rail, scale 1:1

track quality of the left rail, scale 1:1

warping, scale 3:1

gage, scale 1:1

superelevation, scale 1:1

offset, right rail, scale 1:2.5

offset, left rail, scale 1:2.5

One measure for the quality of an inspection car is the transfer function of its measurement equipment. The transfer function depends on the number and mutual distance of measurement axles. It is desirable for the transfer function that the individual parameters be as close as possible to 1, i.e. that errors in their magnitude be neither exaggerated nor diminished. The transfer function of the vehicle is plotted in Figure 4.1. The measurement speed is 80 km per hour.

4.2.2.3 Acceleration and Force Measurements

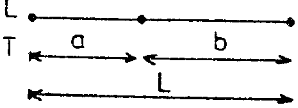
Acceleration and force measurements have been regularly scheduled since 1970, as a necessary supplement to geometric measurements. All the important primary tracks, about 6,500 km, are here measured three times per year.

With the acceleration measurement runs, the track quality is checked at maximum speed, by means of the lateral acceleration

Figure 4.1 TRANSFER FUNCTION FOR MEASURING THE TRACK SURFACE AND ALIGNMENT (SMALL INSPECTION CAR)

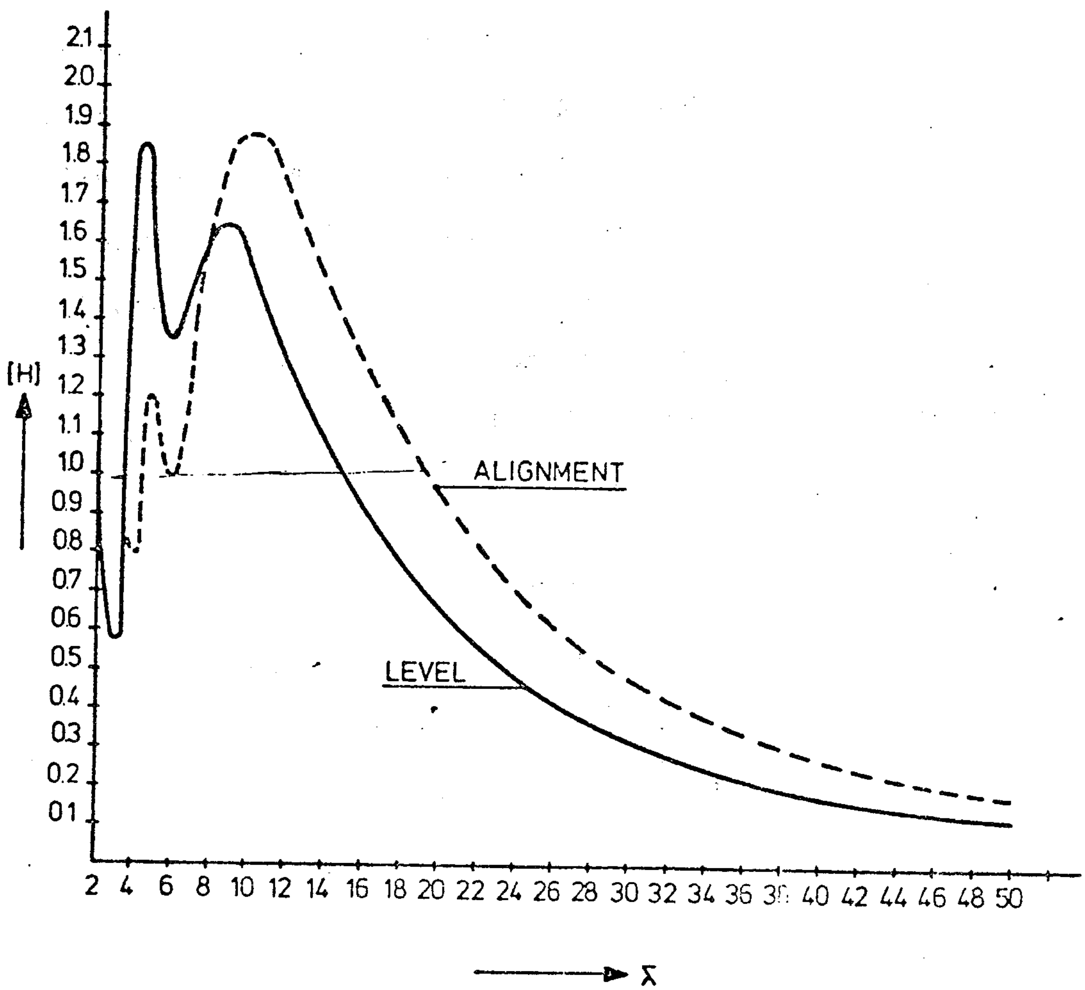
MATISA M-441 track recording car used by the DB

Transfer functions for LEVEL and ALIGNMENT



$$|H| = \sqrt{1+\frac{a^2}{L^2}+\frac{b^2}{L^2}+\frac{2ab}{L^2}\cos 2\pi\frac{(a+b)}{\lambda}-\frac{2b}{L}\cos\frac{2\pi a}{\lambda}-\frac{2a}{L}\cos\frac{2\pi b}{\lambda}}$$

$$\left.\begin{array}{l} a = 2{,}6\,m \\ b = 6{,}0\,m \\ L = 8{,}6\,m \end{array}\right\} \text{LEVEL} \qquad \left.\begin{array}{l} a = 4{,}0\,m \\ b = 6{,}0\,m \\ L = 10{,}0\,m \end{array}\right\} \text{ALIGNMENT}$$



measured in a passenger car. The running behavior of the car is known and representative. The lateral suspension forces are also measured on this vehicle. The measurement system is built into a car, which runs in the middle of a train, consisting of three passenger cars. Among other things, this system consists of the following:

- the accelerometer
- measurement equipment for the lateral suspension force
- a magnetic tape storage unit and
- the recording oscillograph (Figure 4.2).

The accelerometer is fixed on the floor of the car, always above the kingpin of the truck which is trailing in the direction of travel. The lateral suspension force is always measured on the leading axle of the trailing truck in the direction of travel.

As long as no trouble spot occurs, it is not necessary to record the measured data continuously. However, in order to be able to record accelerations and forces in the area of a trouble spot, all measured values are briefly stored on magnetic tape - about 10 seconds. When specified limits are exceeded, the recorder switches on and makes a delayed transfer of the tape-recorded information onto the measurement chart. In this way, it is possible to follow the start-up and decay of the defect (Figure 4.3). The experience has confirmed that very short peak accelerations can be disregarded with the

FIGURE 4.2 SYSTEM SKETCH OF THE TROUBLE MEASUREMENT SYSTEM

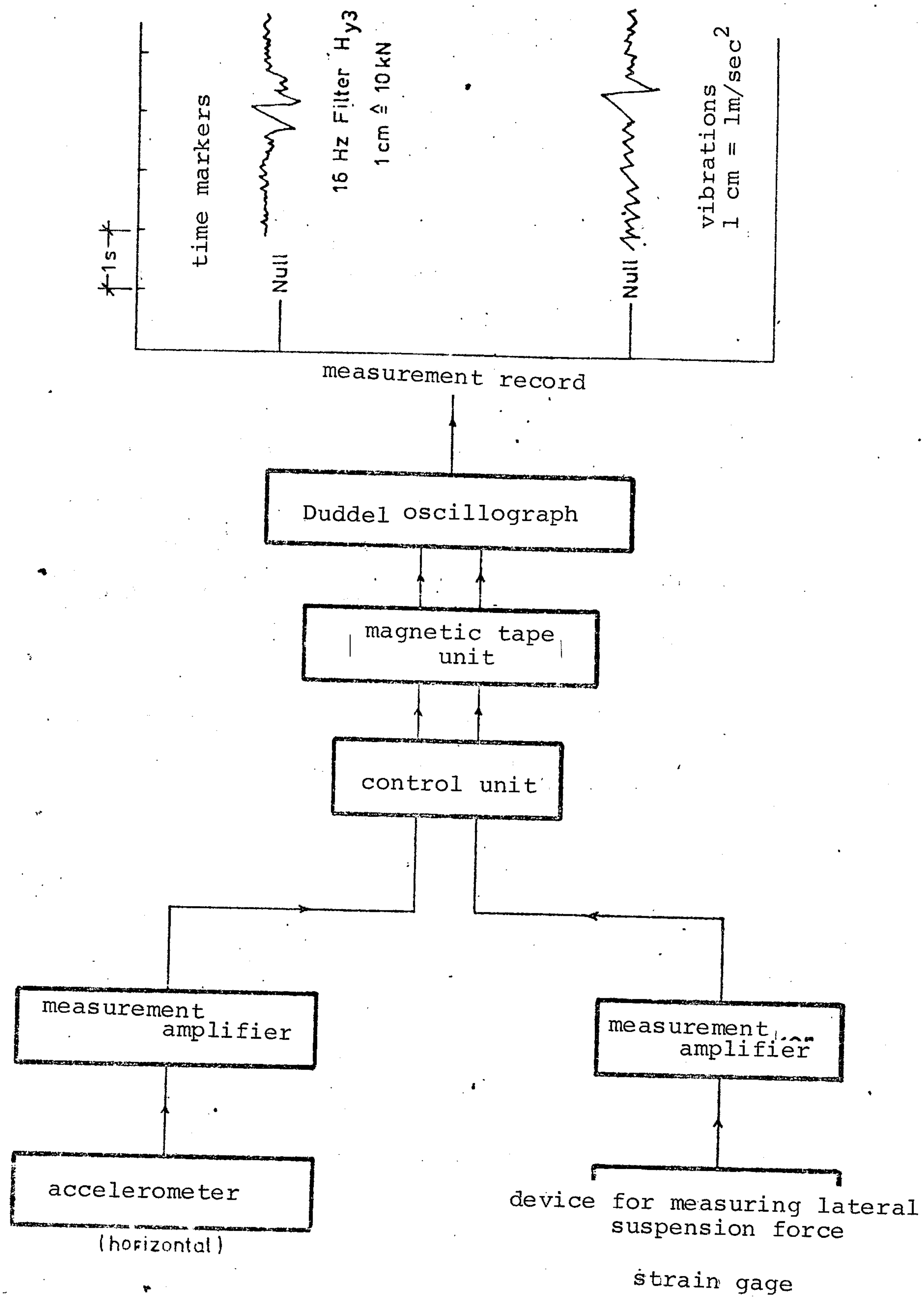
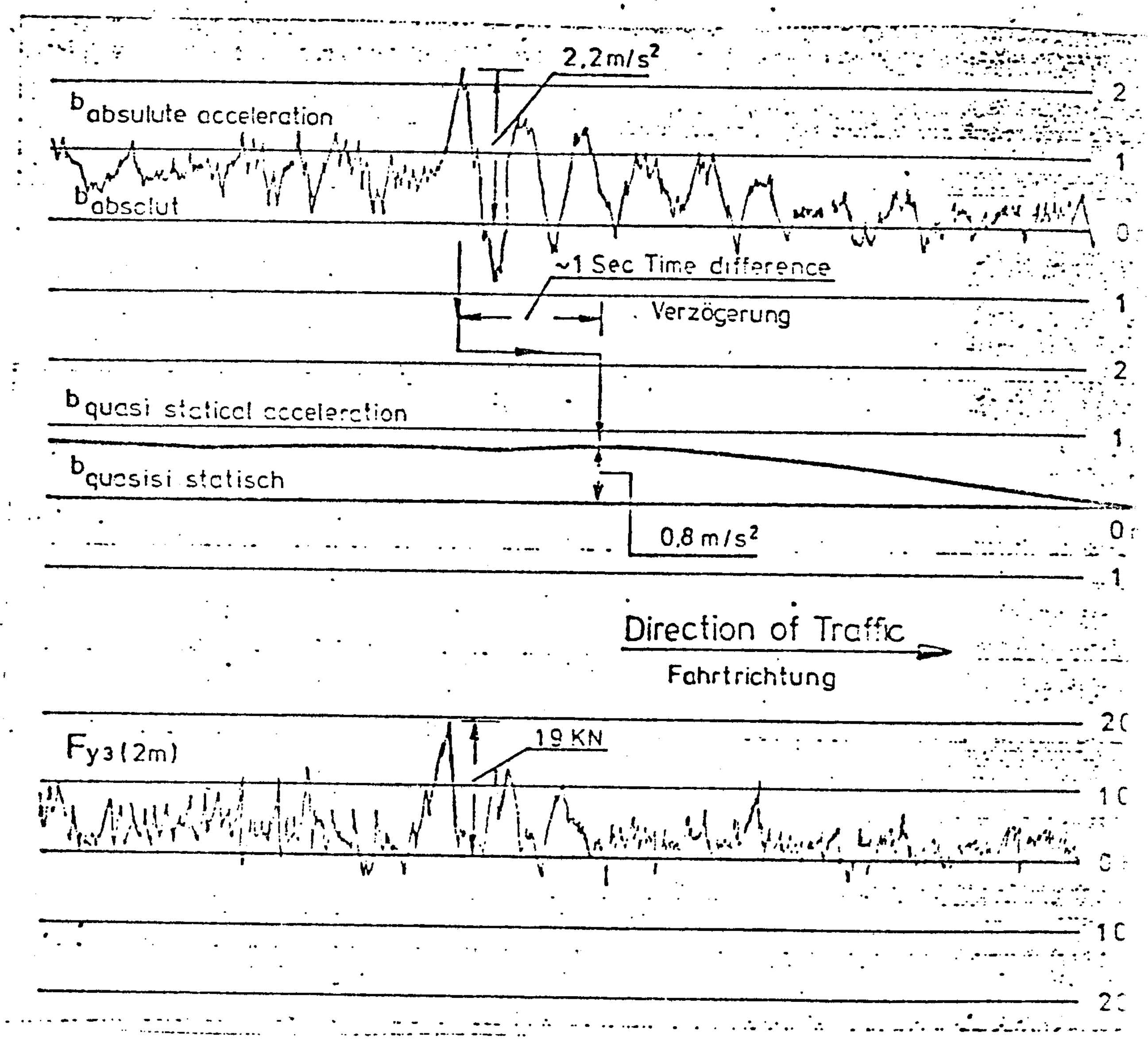


FIGURE 4.3 MEASUREMENT RECORD OF THE ACCELERATION MEASUREMENT TRAIN



measurements. Consequently, frequencies above 16 Hertz are filtered out. The measurement data obtained in this manner are recorded as absolute lateral acceleration and lateral suspension force.

Furthermore, the lateral acceleration are also filtered above 0.5 Hertz, so that the quasi-static and dynamic components of the absolute lateral accelerations can be separated. Taking into account the roll angle of the car body, which enlarges the quasi-static lateral acceleration by 30 percent to 40 percent, the triggering limit values for the dynamic and quasi-static lateral acceleration were each set to 1.2 m/sec^2.

The measurement of the lateral suspension force has in the meantime proven less informative and is no longer used.

The triggering levels for the measurement equipment are chosen in such a manner that incipient track quality defects are recognized early. The measures that need to be initiated are guided by the degree of transgression of the measured values (see Point 1).

The systematic acceleration and force measurements led to a quantifiable improvement in track and switch conditions, since more specific maintenance could be implemented on the basis of these measurements. By far the greatest fraction of trouble spots, and specifically about 85 percent, is caused by constraint points of the line layout. Switches here represent the major fraction, with about 65 percent.

4.2.2.4 Ultrasonic Measurements

Furthermore, heavily loaded tracks are traversed with a new ultrasonic rail testing train twice a year, and the remaining primary tracks once a year, in order to detect rail defects early and to be able to avoid rail ruptures. The annual testing performance is 35,000 kilometers. Cracks and defects in the rail head, -web, and -base area, as well as welding defects are here determined and recorded at a measurement speed of 50 kilometers per hour, after the principle of the pulse echo method. The defects that have been thus determined are subdivided into three groups. The relative frequencies of these groups have the following trend (Figure 4.4).

Groups 1 and S (welding) include defects for which early rupture is expected. These must be immediately eliminated or secured by fishplates. Among these defects belong e.g. defects which include an area of 50 percent of the rail head section, or welding cracks.

Group 2 includes smaller defects and cracks. Defects of this group can be left in the track, but must be identified and observed.

The position of the defects can be specified with an accuracy of ± 10 cm.

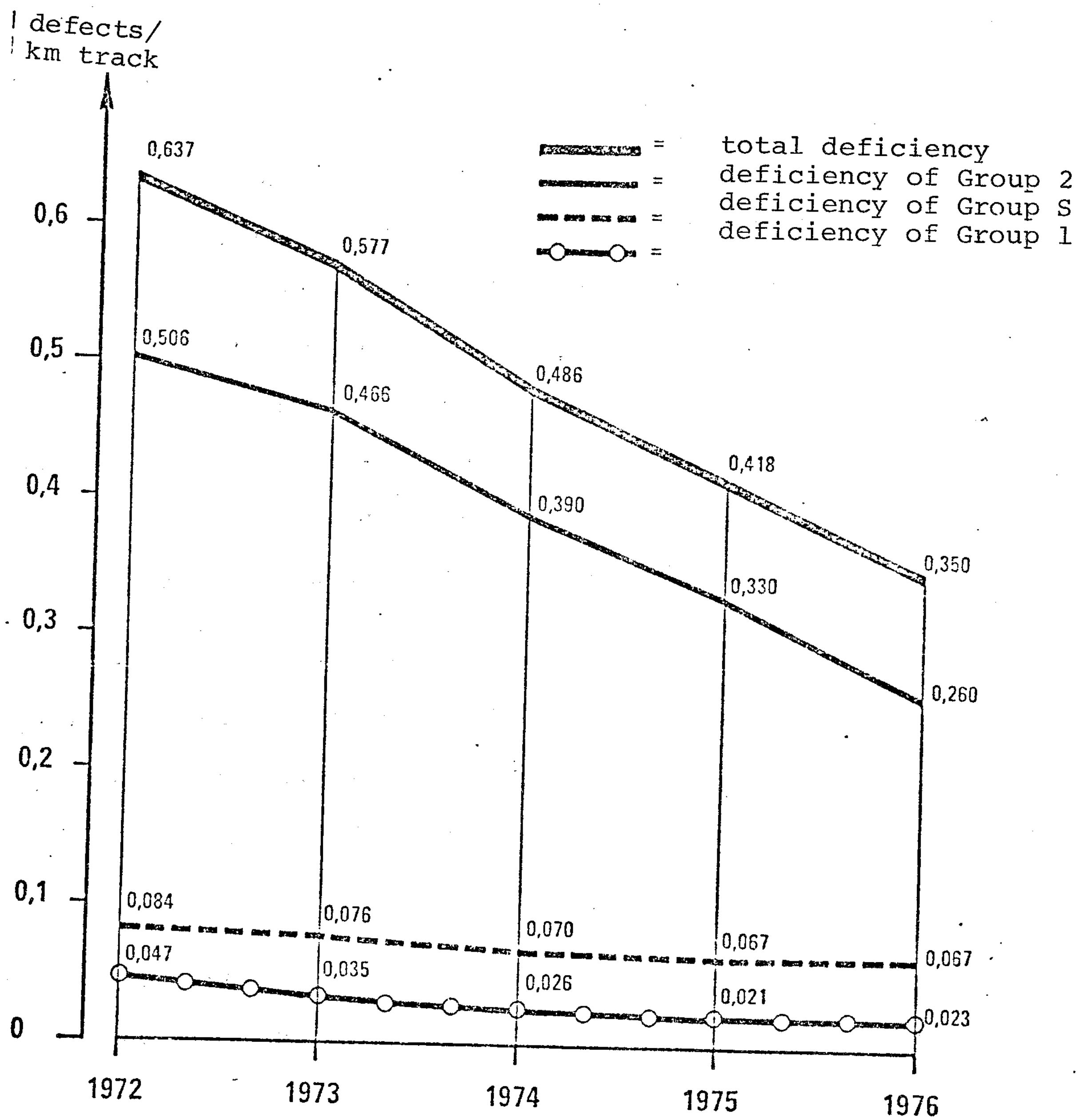


FIGURE 4.4 RELATIVE FREQUENCY OF RAIL DEFECTS (MEASUREMENT RESULTS OF THE ULTRASONIC RAIL TEST TRAIN)

The data are evaluated centrally by especially trained personnel (Figure 4.5). This considerably reduces the personnel involved in the run. This method involves a delay in ascertaining defects, but this delay is accepted, since the rail fastenings customary give no cause to fear derailment even if a rupture occurs. In order to reduce the personnel expended for evaluation, automatic evaluation is being striven for.

Manual devices are used for testing only in special cases as well as for checking the components of switches, especially the tongues of switches for high-speed travel.

4.2.2.5 Rail Surface Measurements

Finally, the German Federal Railway supplements its geometric track surface measurements by means of a rail surface inspection trolley. By means of mechanical sensors and accelerometers, this trolley senses corrugations and undulations up to a wavelength of about 35 cm with a measurement speed of 50 km per hour. No individual values are recorded on the measurement chart (Figure 4.6), but rather running averages, which are formed over several meters of rail. For each rail, the wavelength, as well as the wave depth, are recorded here. Evaluation of the measurement chart forms the basis for determining the extent of rail to be treated by the rail grinding train.

The regular treatment of the running surface of the rails on high-speed and high-load rails is regarded as necessary in order to avoid the higher noise level (increased by about 8-10 dBA), which exists even with minor corrugations. This is motivated by reasons of environmental protection and riding comfort.

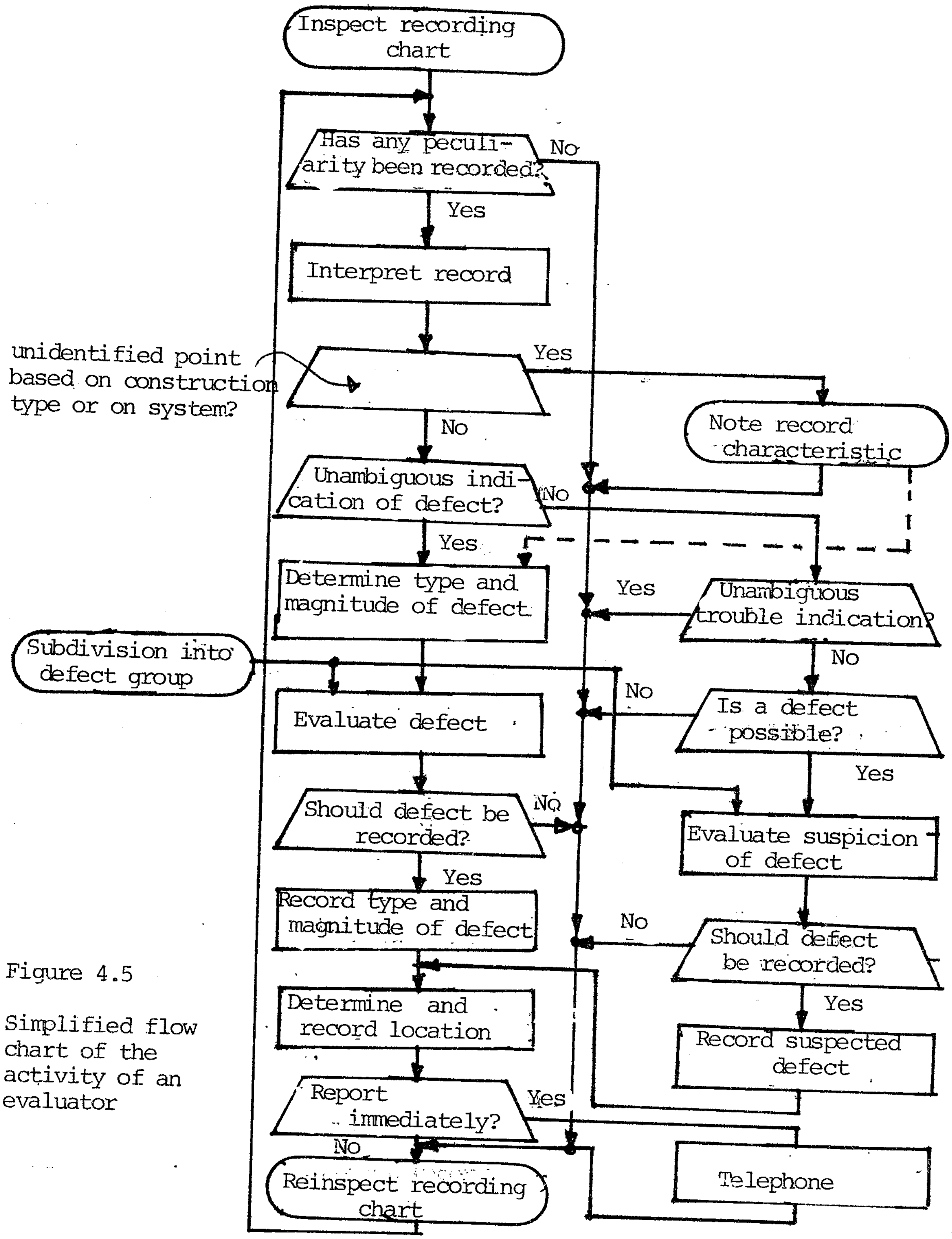


Figure 4.5

Simplified flow chart of the activity of an evaluator

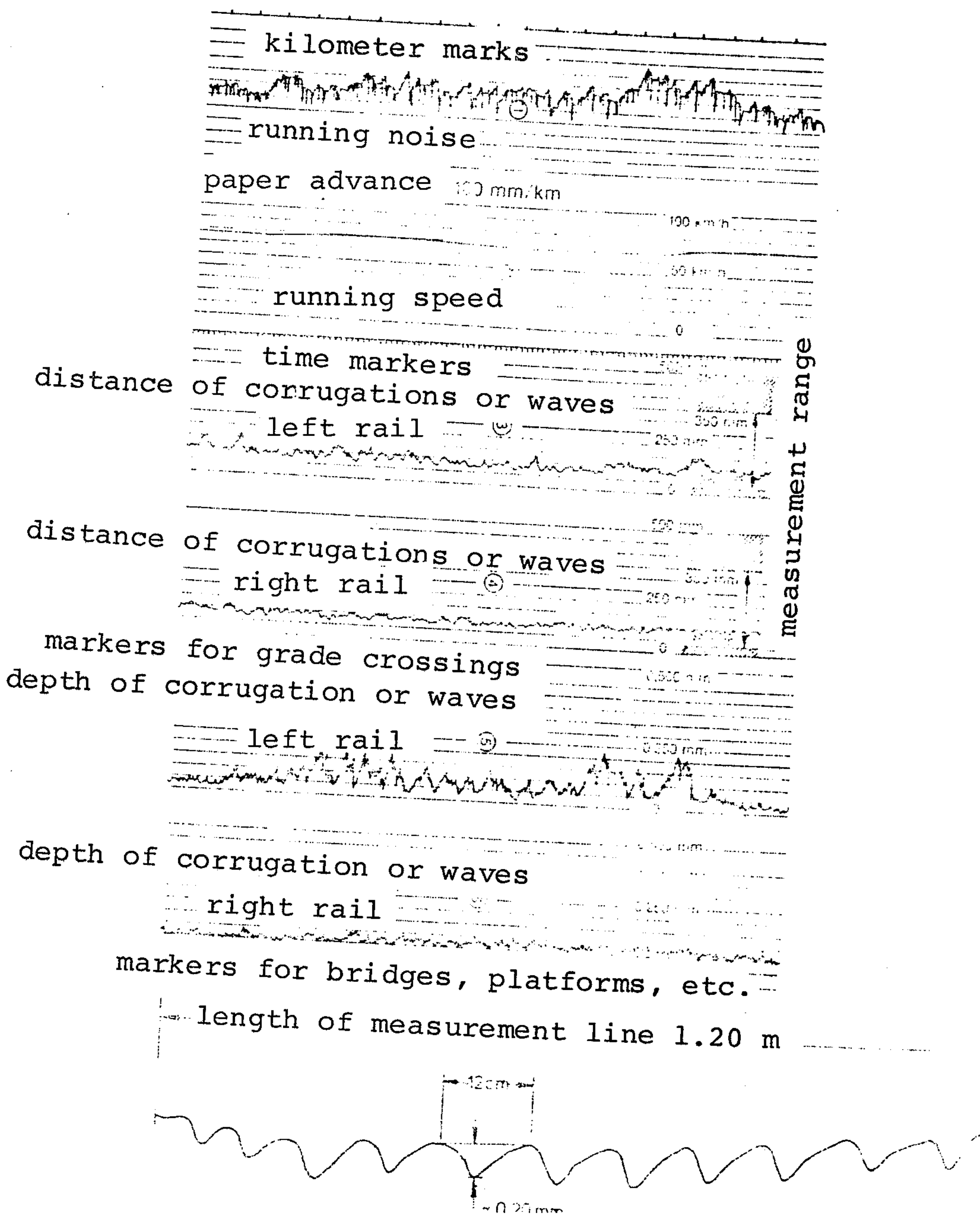


FIGURE 4.6 MEASUREMENT RECORD OF THE RAIL SURFACE INSPECTION TROLLEY

4.2.3 Costs

Costs for track maintenance are affected by the track maintenance methodology, the wage scale, and material costs. A comparison between German and American conditions can therefore lead to wrong conclusions. The authors limit themselves to presenting the cost reduction, which the German Federal Railway was able to achieve by introducing continuously welded rails. They believe that this saving could also be realized by American railway companies.

For a comparative consideration of economy, both one-time and running savings must be accounted for.

4.2.3.1 Tracks

The added costs for fabricating a fishplate joint are about 100 DM. This covers the additional ties and base plate and fastening (the German Federal Railway lays joints only on double ties). Fish plates and fish plate screws require about 120 DM and about 30 DM for installation. Consequently, the resulting additional costsare approximately 250 DM. Relative to 30 m long rails fabricated in steel plants, the additional costs are therefore calculated in the amount of $250 \div 30 = 8.33$ DM per meter of track.

The manufacturer delivers 30 m long rails. The German Federal Railway welds these stationary in a welding plant into 120 m long rails. The costs for this welding are 50 DM per weld. The "Thermit" welds produced at the construction site, on the hand, cost 250 DM per weld. Relative to a 120 m long rail section, the following costs are thus calculated:

6 plant welds, each 50 DM		= 300 DM
2 track welds, each 250 DM		= 500 DM
	Σ	= 800 DM

800 DM ÷ 120 m = 6.66 DM per meter of rail.

Different figures will result for railways with different fish plates and rail fastening constructions. In any case, the savings for material and production costs already more than balance the costs for welding.

More important than the one-time costs are the running costs, which accrue in the maintenance of jointed rails. Both the reduced lifetime of the jointed rail and the different maintenance expenditure must be considered. It can be started from the fact that jointed rails have a service life of only 22 years as compared to 25 years for a continuously welded rail.

One meter of rail, incuding rails, ties, and fastenings, costs about 450 DM, plus 100 DM for track renewal. Because track materials can be reused, their recovery is assumed at 30 percent of investment costs.

Running maintenance costs per rail joint can be set at about 2.50 DM per year. This yields an amount of 2.50 DM x 2 ÷ 30 = 0.17 DM per meter of track.

An estimated total calculation of annual costs (A), with an interest rate of P = 6 percent, for one meter of rail, shows the following:

1. K = 8.33 DM, n = 22 year (K = capital costs)

   A = 8.33 x 0.083 = 0.69 DM

2. K = 6.66 DM, n = 25 year

   A = 6.66 x 0.078 = 0.52 DM

3. K = 550 DM, E = 0.3, n = 22

   A = 0.7 x 5.550 x 0.083 + 0.3 x 550 x 0.06 = 41.85 DM

   K = 550 DM, E = 0.3, n = 25

   A = 0.7 x 550 x 0.78 + 0.3 x 550 x 0.06 = 39.93 DM

4. Maintenance costs = 0.17 DM

This shows that the welded rail has an advantage of (0.69 - 0.52) + (41.85 - 39.93) + 0.17 = 2.16 DM per meter.

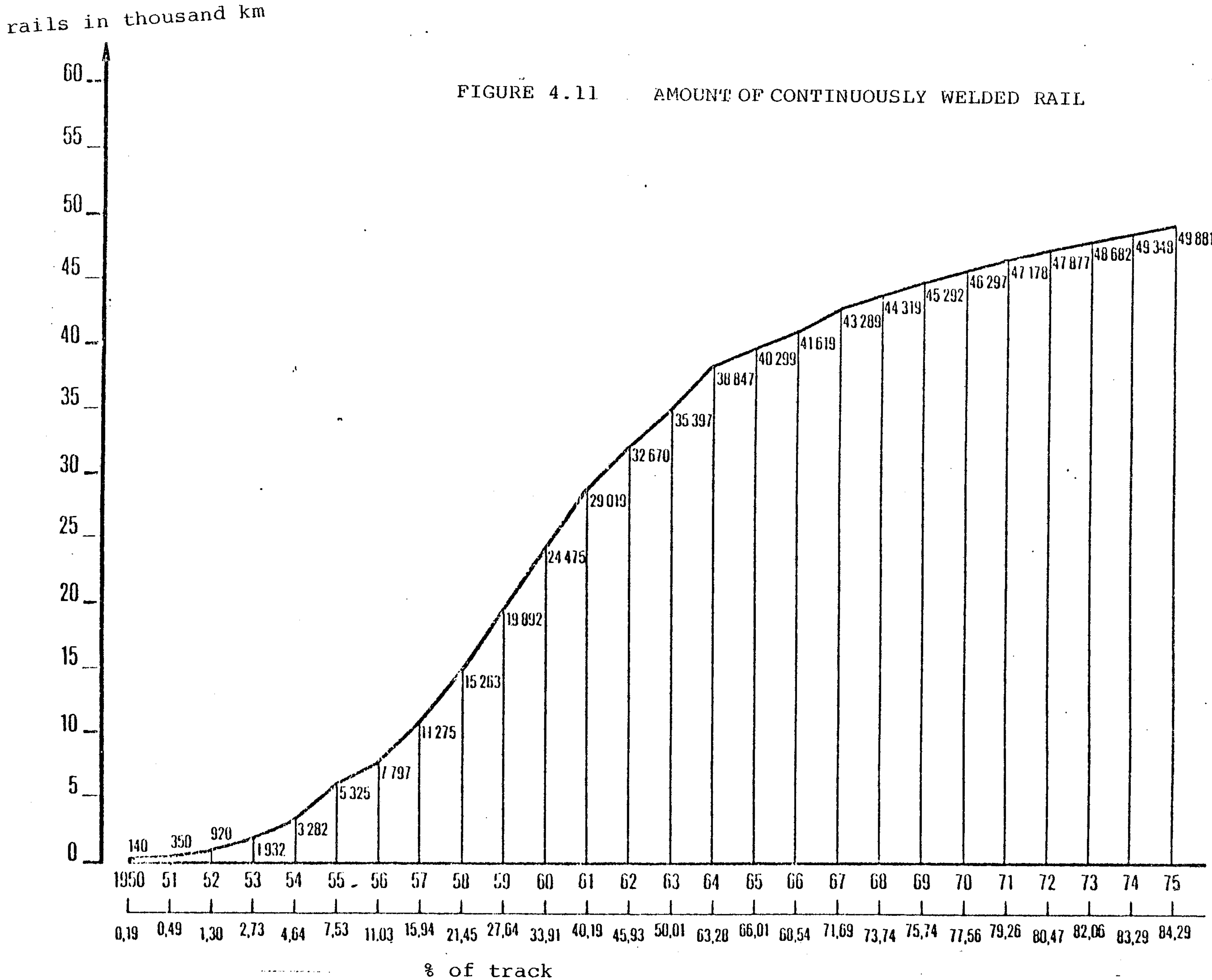
4.2.3.2 Switches

In the meantime, the German Federal Railway also welds all switches continuously with rails. Experience has shown that the lifetime of switches can thereby be increased by 25 percent. From precise studies, it is known that the saving per switching unit is about 700 DM per year.

4.2.3.3 Imponderabilia

Savings for the rolling material, as a consequence of more quiet vehicle running, are significant, but not precisely quantifiable. Non-quantifiable savings in driving power also

FIGURE 4.11 AMOUNT OF CONTINUOUSLY WELDED RAIL



accrue. According to a study of the AREA (American Railway Engineering Association) (Bulletin 549, 1959, pp. 633-653), savings from continuously welded rail were also calculated in the U.S.A. In 1953, these savings already amounted to 1.040 dollars per mile. If the exchange rate of that time is taken into account these data check with our calculations.

4.2.3.4 Summary

At the present time, the German Federal Railway has continuously welded 85 percent of its rails (Figure 4.11). This procedure was stimulated by the expected cost savings, which has indeed fully manifested itself.

4.3 Research Activity on Reducing the Cost of Track Maintenance

In the area of track maintenance, the last two decades have been characterized by the introduction of continuously welded rails, mechanization of track maintenance, and increase of speed to 200 km per hour. These measures were necessary primarily because of economic reasons. They are now being supplemented by specific component studies, which extend to all areas of the track. These studies are partly being carried out on a national level and partly on an international level.

4.3.1 Ballast

On the occasion of revising the delivery conditions for ballast, Munich Technical University is, at the present time, studying the effect of various ballast granulations, i.e. various grades, as well as various ballast strengths, on the settling behavior of the ballast. The most favorable ballast characteristics are being determined by means of laboratory experiments. Comparison sections about 1 km long, will subsequently be installed with standard and with optimized ballast in heavily loaded lines. The track quality of both sections will be measured periodically, both with respect to a fixed base and with the inspection car. The two sections will thus be compared. If it should appear that practical results correspond to laboratory experiments, the technical delivery conditions for ballast will be changed in mutual agreement with the ballast supplier firms.

4.3.2 Ties

On the basis of theoretical investigations, it was determined that enlarging the tie support surface effects a considerable reduction of ballast compression. Track stability could thereby be considerably improved. For this reason, experimental sections were laid with ties of various lengths - concrete ties 2.80 m long, concrete ties 2.60 m long, wooden ties 2.70 m long, and wooden ties 2.60 m long; the behavior of these sections is being observed by means of track quality measurements.

These experiments have almost been concluded and have up to now not led to the expected success. Presumably, this is partly based on the fact that the tamping process was not adapted to the longer ties, i.e. the external tamping units would to have had to be moved somewhat away from the rail.

Furthermore, an experiment was undertaken to improve track quality by installing elastic rubber pads under the base plates, to attenuate dynamical stresses. No improvement of track stability appeared. On the contrary, it appeared that the ties could no longer be perfectly tamped and aligned, since they tilt and turn.

4.3.3 Track Maintenance Methods

Within the framework of the ORE, the precision and durability of the operational methods of various tamping and aligning machines are at the present time being compared with one another. These studies extend over several years. Up to now they have led to the following results:

1. The use of alignment machines alone does not lead to an improvement of track quality. Tamping must be done at the same time.
2. Tamping with the fixed-point method does not yield better results than tamping with the smoothed method.
3. The operational results of the machines are poorer than would be expected in terms of the theoretical transfer function. This means that track defects of certain wavelengths are sometimes improved only little. (Important only for high-speed trains!)

The results of these experiments are contained in Report No. 10 of the ORE Committee D 117.

4.3.4 Studies on the Vehicle-Rail Interaction

In connection with researches on the running behavior of fast vehicles (locomotives and passenger cars), it was determined that larger track defects need not necessarily result in a reaction on the part of the vehicle. This means that the forces measured in the wheel/rail contact area, as well as the horizontal and vertical accelerations measured in the vehicle, remain within the permitted tolerances. On the other hand, higher forces have been determined, even though track quality was essentially free of defects. These statements, however, refer only to the speed range $V > 160$ km/h. At speeds $V \leq 160$ km/h, a relation between the magnitude of the force and the quality of the track can generally be found.

Our studies now extend to the examination of the defects and their consequences which lead to considerable forces or accelerations for fast vehicles, and, inversely, of track defects can be traversed without vehicle reaction. For this purpose, no track maintenance is being performed in a rather long track section of about 30 km. Only those trouble spots are being removed, which lead to substantial vehicle reactions. In this fashion, we hope to be able to limit track maintenance in fast track sections to the absolutely essential scope.

4.4 How the Concrete Tie B70 Has Proven Itself

The concrete tie B70 was developed ten years ago on the basis of long years of experience with the B58 tie and other prototypes.

In its dimensions, it corresponds approximately to the wooden tie usual with the German Federal Railway. Consequently, it can also be laid at the same tie spacing as the latter. This tie has up to now proven itself superbly. No damages have appeared in operation, which could be based on overstress or long term stress. When laying this tie instead of wooden ties, care must be taken that the ballast thickness is still 30 cm underneath the concrete tie. This requires special attention because of the greater height of this tie. This requires cleaning the ballast so that the ballast bottom can be laid deeper, or alternatively requires raising the grade.

The delivery conditions for concrete ties are listed under 4.2.

A construction drawing of the B 70 tie for the rail fastening W and the rail S54 is enclosed as Appendix 4.4.

The concrete tie (monoblock) has an advantage for continuously welded rails, namely that it has a very high track lateral strength. The double block tie of the SNCF (French National Railway) also has this advantage, but they are sensitive to bending at high axle loads. As a result, changes of gage and of rail slope can occur.

4.5 The Rail Fastenings

The German Federal Railway today predominantly uses the following fastenings:

K fastening

This fastening may be assumed as known and will not be described further.

W fastening

This fastening is a standard fastening for concrete ties. The rail here lies directly on the concrete body, while a plastic pad is placed in between. The rail is laterally secured by the shoulder plates. The horizontal forces transmitted to the rail are directly dissipated through the shoulder plates into the concrete body. The friction-type tightening between the rail and the tie is achieved by means of the elastic clip SKL 1. This clip is anchored in the concrete by means of a tie screw and through a plastic plug. The elasticity of the plastic plug here effects an elastic transmission of forces to the concrete body. This is advantageous in view of the existing centering stress. Tbe correct installation position of the elastic clip SKL 1 is reached when the center loop of the elastic clip touches the rim of the shoulder plate. A tightening torque of about 250 Nm is required for this. The tightening force of the spring arm on the rail base is then about 11.5 kN with a spring displacement of about 14.5 mm.

DNA 4 Fastening for Wooden Ties

The dual elastic spike DNA 4 has proven a simple and economic rail fastening. With this fastening, a base plate can be dispensed with. The required rail cant must therefore be milled into the tie surface. The dual elastic spike DNA 4 consists of two square shafts, combined by a web. The lateral guide of the rail is given by the web. The spike shafts are pounded into round predrilled holes. It is here important not to hit the spring clip, but the web. The dual elastic spike has a spring force of about 15-18 kN, with a spring displacement of 10-11 mm. This fastening is today installed in tracks with a maximum load of 15,000 t per day and in curved tracks with radii greater than 1000 m, as well as in all remaining tracks of subordinate significance.

4.6 Ballast and Sub-ballast Material

Reference was made under Point 4.2 to the ballast delivery conditions applicable for the German Federal Railway. Furthermore, under Point 4.3, experiments were mentioned which the German Federal Railway is currently performing relative to the improvement of the use characteristics of ballast.

The delivery conditions are enclosed. They are at this time being revised and are being updated to the most recent technical status. The most important ballast tests are the following:

1. Determining the gradation

Intervals are specified for the percentage fractions of various granulation categories. The gradation must lie within the prescribed range (Figure 4.12).

2. Testing frost resistance

Before the delivery is approved, and at certain time intervals thereafter, the material is investigated for its frost resistance. The testing method corresponds to an industrial standard

(DIN) and is also used in street construction.

3. Determination of the impact strength

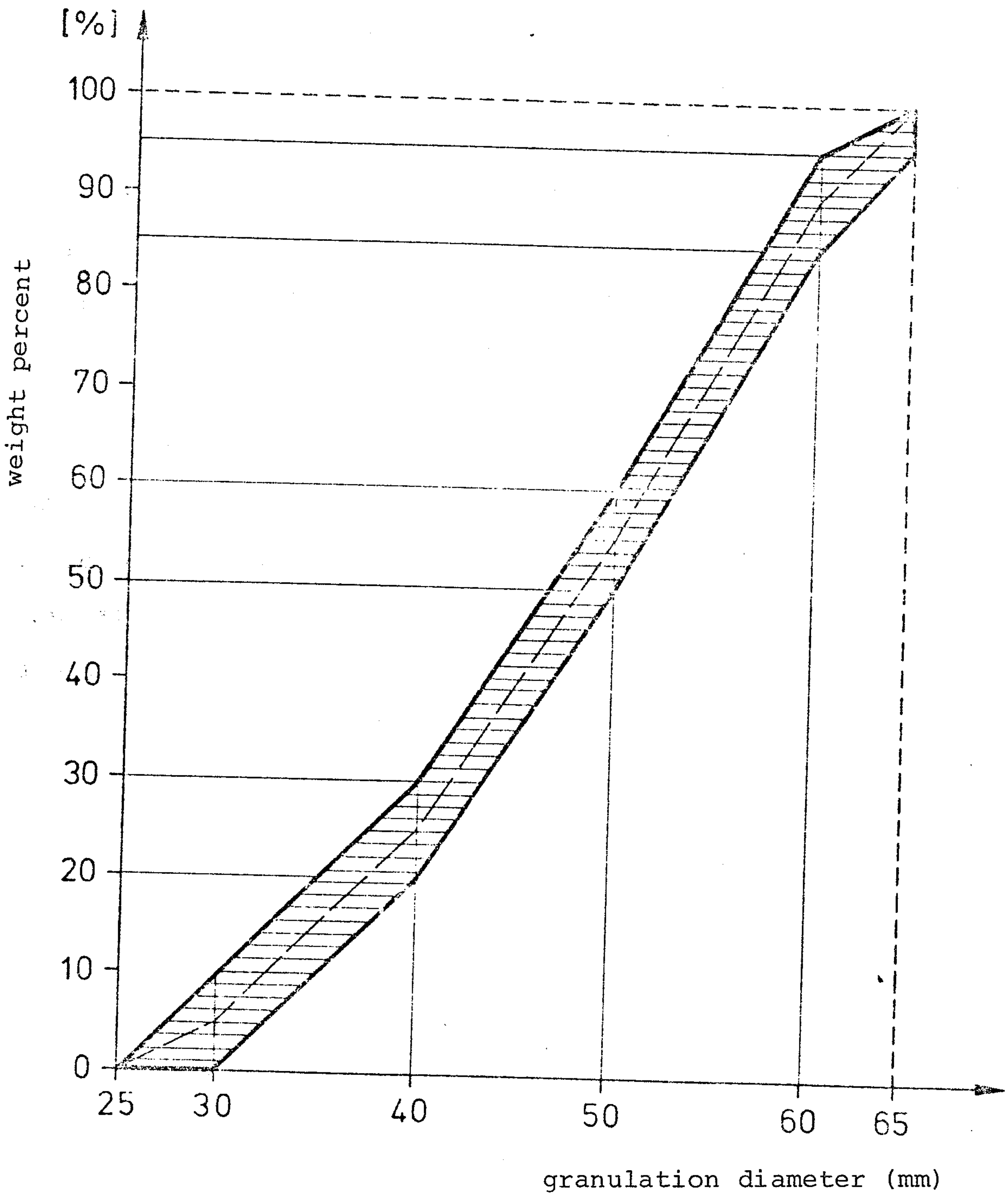
A certain amount of ballast is exposed to a stress from a weight that falls several times from a specified height. The degree to which the ballast shatters is measured. The measurement scale for evaluating the ballast is the degree of shattering ascertained in comparison to the hardest rock. The test device is standardized. The evaluation procedure is handled differently in road construction than with the German Federal Railway.

4. Determination of the ratio of flat/compact material

The ratio of flat/compact material provides information concerning the proportion of plate-like and wedge-like material in the total mass. A certain fraction of this material is regarded as necessary to retain the elastic properties of the ballast. The test method is based on experience. The flat stones present in the individual granulation categories are here placed in proportion to the compacted stones.

The other test methods specified in the technical delivery condition have only subordinate significance or are required only with the initial approval of a supplier firm. The above-mentioned tests are performed regularly, in dependence on the quantity of ballast delivered. They are performed by a central testing agency of the German Federal Railway. It would also be possible to have these tests performed by officially approved, neutral test institutions.

FIGURE 4.12 GRADATION FOR RAIL BALLAST



The German Federal Railway has also set up delivery conditions for the delivery of sub-ballast material. There likewise exist directives for the design of drainages. Both directives can be transmitted if desired.

From experience, a sub-ballast layer at least 30 cm thick is regarded as necessary. The sub-ballast layer must be proof against frost and must prevent the rising of fine material from the subsoil.

5. Final Remarks

For several reasons, summary conclusions will be omitted. On the one hand, only about two months working time were available instead of the originally anticipated three months. Therefore, just for reasons of time, limitations had to be imposed as regards topics and content. On the other hand, the authors have concerned themselves almost exclusively with the running track research program, and have had no direct contact with the U.S. railway industry.

This report therefore primarily treats measurement methods, measurement results, and the guidelines derived therefrom, within the framework of questions arising from the U.S. research program. Furthermore, calculational methods are indicated and construction directives are given, which are based on experience. Their transferability to American circumstances with greater axle forces appears justified. To the extent that recommendations appear to be meaningful, in the view of the authors, such recommendations were made in the individual sections of the report.

It should also be pointed out that the authors were not competent to answer all questions. In these cases, reference was made to sources and their addresses.

The appendices have been translated only to the extent required for understanding the report. For the rest, the complete German version of the appendices is enclosed.

APPENDIX 1.1

SOURCE: Glas. Ann. (Glaser's Annals, Journal for Railway and Traffic Engineering) Volume 96, No. 12, 1972, pp 373-385

TITLE: Determination of Forces Between the Wheel and the Rail from the Bending Strain of the Wheelset Axle

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DK 625.2.03

Summary: A measurement method is described for determining the forces between the wheel and the rail from the bending strains of the wheelset axle. Both the theoretical foundations of this method as well as the devices developed and practical implementation of the method are described. Measurement errors and, as much as possible, their compensation are discussed. Furthermore, reference is made to the possibility of combining this measurement method with the Y-force measurement after the familiar "wheel disc" method. Although the present measurement method has already been developed to such an extent that it can be used without further ado for a large number of measurement tasks, there is no doubt that still further development work must be performed so that the forces between the wheel and the rail can be measured in all cases involving engineering studies under running conditions.

It is necessary to make continuous measurements of all the forces occurring between the wheel and the rail over longer periods, both in order to detect critical special cases with respect to safety as well as to evaluate statistically the vehicle and/or the track.

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1. Introduction

The running safety and riding quality of railway vehicles, as well as the stress and wear on running gears and tracks, depend to a large extent on the forces which result from the vehicle-rail interaction. Consequently, knowledge of these forces is especially significant. For this reason, great efforts have been made at all times to measure the magnitude and temporal course of the forces between the wheel and the rail. But only the appearance of strain gauges after the Second World War opened new possibilities for developing measurement methods to determine the forces between the wheel and the rail. These forces can be measured on the rail as well as on the vehicle. Because of the great effort involved, the course of forces on the rail can be measured only in short segments of several meters. Measurement of the course of the forces over long distances, on the other hand, is possible only from the vehicle.

When making measurements from the vehicle, the forces between the wheel and the rail are derived from the material stress at various points of the wheelset. The material stress is there acquired by measuring the strain. Up to now, prim[illegible] [illegible]ods have been developed and applied, where the force between the wheel and the rail has been derived from the material strains of the wheel discs or wheel spokes. In particular, the force transverse to the rail was measured thereby. Because the measured effects are only small, however, the vertical forces and the forces longitudinal to the rail could not be acquired with the desired accuracy.

Besides these measurement methods, another method has been used in recent years. In this method, the forces between the wheel and the rail are determined from the bending strains of the wheelset axle. Sperling (1)[1], in 1961, specified a system of equations for calculating the lateral guide force Y (Figure 1) and the contact forces Q from the bending stresses in four cross sections of the wheelset shaft. Starting from this, a measurement method was soon developed in the Federal Railway Experimental Institute at Minden (Westphalia), where the bending strains of the wheelset axle were measured with strain gauges and were immediately converted into the wheel forces Y and Q by means of analog computer modules. However, only two usable measured values per wheel revolution were obtained in this manner, and specifically when the measurement plane of the bending moments traversed the perpendicular to the plane of the track. In 1966, Müller and Nefzger (2) expanded this system of equations in such a manner that a continuous measurement of the Y and Q forces and furthermore of T_x became possible. Appropriate measurement and computation equipment was developed for this purpose, was constructed, and has repeatedly been used in engineering experiments under running conditions.

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[1] The numbers in parentheses refer to the list of references at the end of the paper.

2. Theoretical Foundations

Figure 1 shows the external forces applied to a wheelset. The following forces act at the wheel-rail contact point:

In the vertical direction
(z-axis of the wheelset) the wheel contact forces Q

In the transverse direction
(y-axis of the wheelset) the guiding forces Y

In the longitudinal direction
(x-axis of the wheelset) the tangential forces T_x.

In the bearing box, the following forces act on the wheelset:

In the vertical direction the spring forces F_z

In the transverse direction the transverse bearing forces F_y

In the longitudinal direction the longitudinal bearing forces F_x.

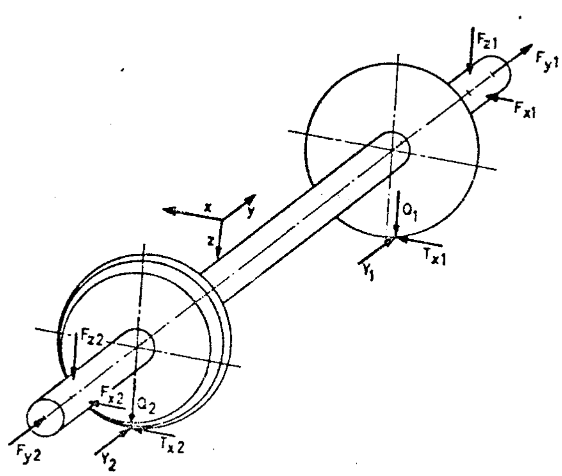


Figure 1: Forces at the Wheelset

The forces on the right side of the wheelset - viewed in the direction of travel - are labeled with the Index 1; those on the left side are labeled with Index 2. This figure also contains the coordinate system with the

x axis in the direction of forward motion of the vehicle,

y axis transverse to the track,

z axis perpendicular to the track.

All motions and forces are referred to this coordinate system.

The direction of the forces is obtained from the coordinate system specified by ORE (3).

The following consideration is the basis for the measurement method: The forces exerted on the wheelset from the outside generate bending moments in the wheelset axle. The bending moment characteristic can be calculated from the external forces, if their magnitudes, directions, and contact points are known. Conversely, from a known bending moment characteristic, the magnitude, direction, and contact point of the external forces can be calculated.

In order to obtain information concerning the forces exerted on the wheelset, it is necessary to know the following two characteristic curves of the bending moments:

a) In the yz-plane, the characteristic generated by the forces Q, Y, F_z, and F_y (if it is not applied to the center of the axle).

b) In the xy-plane, the characteristic generated by the forces T_x and F_x.

These two bending moment characteristics are sufficiently determined by measuring four bending strains each. The simplifying assumption is here made that the forces F_z and F_x are applied at the center of the bearing, that the resulting force F_y is applied at the center of the axle, and that the wheel contact points do not migrate in the y-direction (b_{A1} = const., b_{A2} = const. in Figure 2). Section 5 will discuss in greater detail the measurement errors resulting from this simplification as well as possibilities for compensating them.

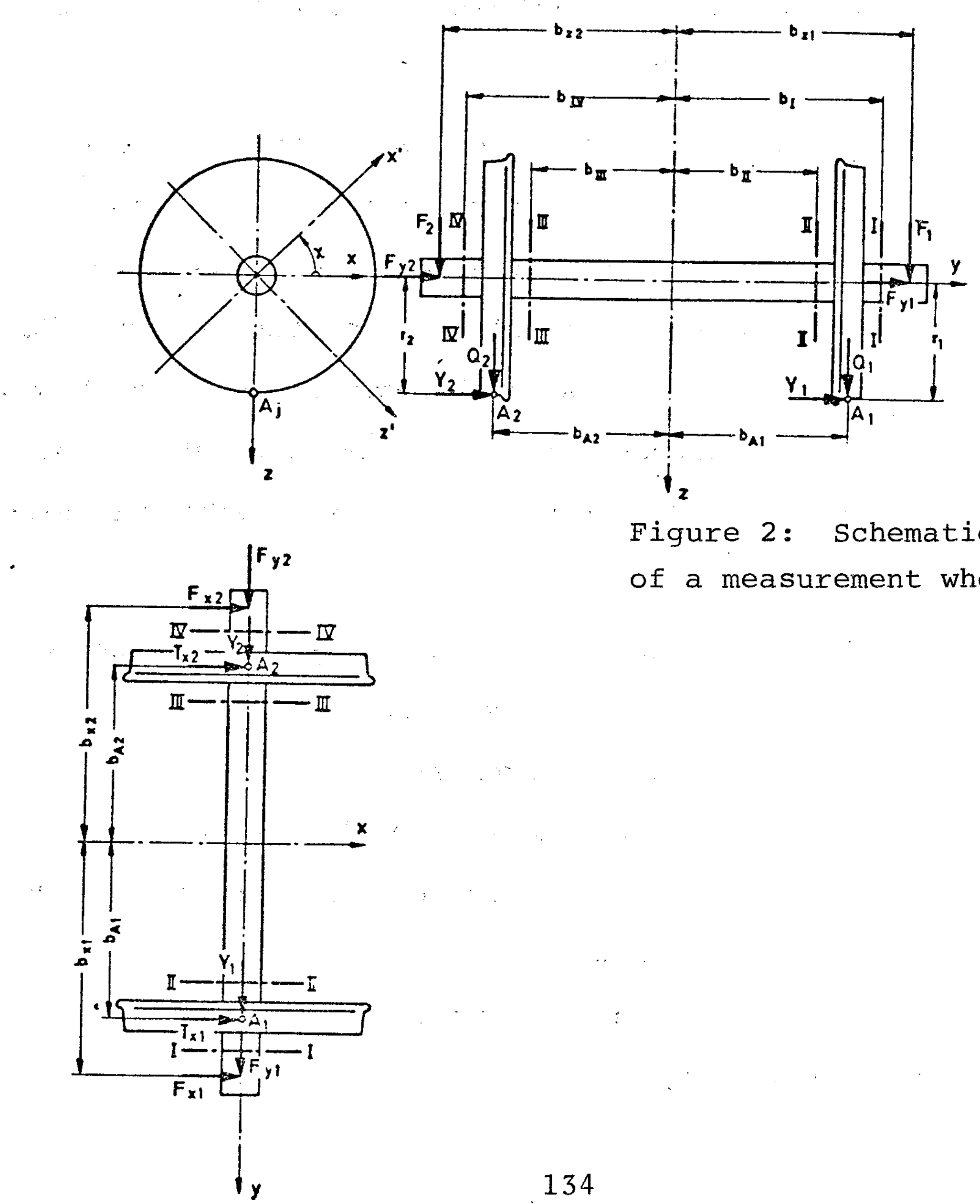


Figure 2: Schematic diagram of a measurement wheelset

Figure 2 contains dimensions and specifications which are required to calculate the relationship between bending stresses and forces under the above-mentioned presuppositions.

The bending moments M_{byz} in the measurement cross sections I, II, III, and IV in the yz-plane are in equilibrium, on the one hand, with the forces F_{z1}, F_{z2}, Q_1, Q_2, Y_1, Y_2, and F_y, which are applied to the wheelset, and on the other hand with the bending stresses σ_z. The following relationships govern this equilibrium:

<u>In measurement cross section I</u>

$$M_{bIyz} = \sigma_{Iz} \cdot W_I \tag{1a}$$
$$= + F_1 (b_{z1} - b_I) \tag{1b}$$

<u>In measurement cross section II</u>

$$M_{bIIyz} = \sigma_{IIz} \cdot W_{II} \tag{2a}$$
$$= F_1 (b_{z1} - b_{II}) + Q_1 (b_{A1} - b_{II}) - Y_1 \cdot r_1 \tag{2b}$$
$$= F_2 (b_{z2} + b_{II}) + Q_2 (b_{A2} + b_{II}) + Y_2 \cdot r_2 \tag{2c}$$

<u>In measurement cross section III</u>

$$M_{bIIIyz} = \sigma_{IIIz} \cdot W_{III} \tag{3a}$$
$$= F_1 (b_{z1} + b_{III}) + Q_1 (b_{A1} + b_{III}) - Y_1 \cdot r_1 \tag{3b}$$
$$= F_2 (b_{z2} - b_{III}) + Q_2 (b_{A2} - b_{III}) + Y_2 \cdot r_2 \tag{3c}$$

In measurement cross section IV

$$M_{bIVyz} = \sigma_{IVz} \cdot W_{IV} \tag{4a}$$

$$= + F_1 (b_{z1} + b_{IV}) \tag{4b}$$

$$+ Q_1 (b_{A1} + b_{IV}) - Y_1 \cdot r_1$$

$$+ Q_2 (b_{IV} - b_{A2}) - Y_2 \cdot r_2$$

$$= + F_2 (b_{z2} - b_{IV}) \tag{4c}$$

In these equations, W is the equatorial resistance moment of the wheelset axle in the respective measurement cross section.

The bending moments M_{bxy} in measurement cross sections I, II, III, and IV in the xy-plane are in equilibrium, on the one hand, with the forces F_{x1}, F_{x2}, T_{x1}, and T_{x2}, which are applied to the wheelset, and, on the other hand, with the bending stresses σ_x. This equilibrium is governed by the following relations:

In measurement cross section I

$$M_{bIxy} = \sigma_{Ix} \cdot W_I \tag{5a}$$

$$= + F_{x1} (b_{x1} - b_I) \tag{5b}$$

In measurement cross section II

$$M_{bIIxy} = \sigma_{IIx} \cdot W_{II} \tag{6a}$$

$$= + F_{x1} (b_{x1} - b_{II}) \tag{6b}$$

$$+ T_{x1} (b_{A1} - b_{II})$$

In measurement cross section III

$$M_{bIIIxy} = \sigma_{IIIx} \cdot W_{III} \tag{7a}$$

$$= F_{x1} (b_{x1} + b_{III}) \tag{7b}$$

$$+ T_{x1} (b_{A1} + b_{III})$$

$$= F_{x2} (b_{x2} - b_{III}) \tag{7c}$$

$$+ T_{x2} (b_{A2} - b_{III})$$

In measurement cross section IV

$$M_{bIVxy} = \sigma_{IVx} \cdot W_{IV} \tag{8a}$$

$$= + F_{x1} (b_{x1} + b_{IV}) \tag{8b}$$

$$+ T_{x1} (b_{A1} + b_{IV})$$

$$+ T_{x2} (b_{IV} - b_{A2})$$

$$= + F_{x2} (b_{x2} - b_{IV}) \tag{8c}$$

Unfortunately, the bending stresses σ_x and σ_z, which are referred to the wheelset-oriented coordinate system x, y, z, cannot be directly measured because of the wheel rotation. Rather, only the bending stresses σ_x' and σ_z' are measured. The latter occur in the rotating, mutually perpendicular measurement planes x'y and z'y of the wheelset axle. As Figure 3 shows, the measurement

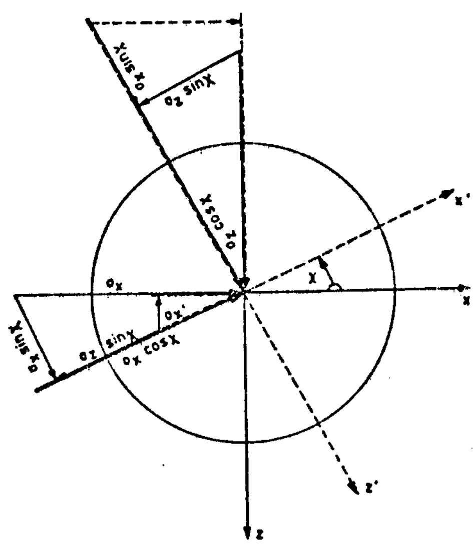


Figure 3: Rotation of a wheelset relative to the x-z reference coordinates

planes x'y and z'y rotate during travel with respect to the reference coordinate system x, z, with a wheel rotation angle χ.

The bending stresses σ_x and σ_z are calculated from the measured bending stresses σ_x' and σ_z' in accord with goniometric relationships as follows:

$$\sigma_x = +\sigma_{x'} \cdot \cos\chi + \sigma_{z'} \cdot \sin\chi \qquad (9)$$

$$\sigma_z = -\sigma_{x'} \cdot \sin\chi + \sigma_{z'} \cdot \cos\chi \qquad (10)$$

The forces, which are applied to the wheelset, as a function of the measurement values σ_x' and σ_z' in measurement cross sections I through IV, and as functions of the wheel rotation angle χ, are obtained by transforming Equations (1) through (10):

$$\begin{aligned} Q_1 = &-\frac{1}{(b_{z1}-b_I)} W_I (-\sigma_{Ix'}\sin\chi + \sigma_{Iz'}\cos\chi) \qquad (11) \\ &-\frac{1}{(b_{II}+b_{III})} W_{II} (-\sigma_{IIx'}\sin\chi + \sigma_{IIz'}\cos\chi) \\ &+\frac{1}{(b_{II}+b_{III})} W_{III} (-\sigma_{IIIx'}\sin\chi + \sigma_{IIIz'}\cos\chi) \end{aligned}$$

$$\begin{aligned} Q_2 = &-\frac{1}{(b_{z2}-b_{IV})} W_{IV} (-\sigma_{IVx'}\sin\chi + \sigma_{IVz'}\cos\chi) \qquad (12) \\ &-\frac{1}{(b_{II}+b_{III})} W_{III} (-\sigma_{IIIx'}\sin\chi + \sigma_{IIIz'}\cos\chi) \\ &+\frac{1}{(b_{II}+b_{III})} W_{II} (-\sigma_{IIx'}\sin\chi + \sigma_{IIz'}\cos\chi) \end{aligned}$$

$$\begin{aligned} Y_1 = &+\frac{(b_{z1}-b_{A1})}{(b_{z1}-b_I)} \frac{W_I}{r_1} (-\sigma_{Ix'}\sin\chi + \sigma_{Iz'}\cos\chi) \qquad (13) \\ &-\frac{(b_{A1}+b_{III})}{(b_{II}+b_{III})} \frac{W_{II}}{r_1} (-\sigma_{IIx'}\sin\chi + \sigma_{IIz'}\cos\chi) \\ &+\frac{(b_{A1}-b_{II})}{(b_{II}+b_{III})} \frac{W_{III}}{r_1} (-\sigma_{IIIx'}\sin\chi + \sigma_{IIIz'}\cos\chi) \end{aligned}$$

$$Y_2 = -\frac{(b_{z2}-b_{A2})}{(b_{z2}-b_{IV})}\frac{W_{IV}}{r_2}(-\sigma_{IVx'}\sin\chi+\sigma_{IVz'}\cos\chi) \qquad (14)$$

$$+\frac{(b_{A2}+b_{II})}{(b_{II}+b_{III})}\frac{W_{III}}{r_2}(-\sigma_{IIIx'}\sin\chi+\sigma_{IIIz'}\cos\chi)$$

$$-\frac{(b_{A2}-b_{III})}{(b_{II}+b_{III})}\frac{W_{II}}{r_2}(-\sigma_{IIx'}\sin\chi+\sigma_{IIz'}\cos\chi)$$

$$T_{x1} = -\frac{W_I}{(b_{x1}-b_I)}(+\sigma_{Ix'}\cos\chi+\sigma_{Iz'}\sin\chi) \qquad (15a)$$

$$-\frac{W_{II}}{(b_{II}+b_{III})}(+\sigma_{IIx'}\cos\chi+\sigma_{IIz'}\sin\chi)$$

$$+\frac{W_{III}}{(b_{II}+b_{III})}(+\sigma_{IIIx'}\cos\chi+\sigma_{IIIz'}\sin\chi)$$

$$T_{x2} = -\frac{W_{IV}}{(b_{x2}-b_{IV})}(+\sigma_{IVx'}\cos\chi+\sigma_{IVz'}\sin\chi) \qquad (16a)$$

$$-\frac{W_{III}}{(b_{II}+b_{III})}(+\sigma_{IIIx'}\cos\chi+\sigma_{IIIz'}\sin\chi)$$

$$+\frac{W_{II}}{(b_{II}+b_{III})}(+\sigma_{IIx'}\cos\chi+\sigma_{IIz'}\sin\chi)$$

Or

$$T_{x1} = -\frac{W_I\,(b_{x1}-b_{II})}{(b_{x1}-b_I)\,(b_{A1}-b_{II})}(\sigma_{Ix''}\cos\chi+\sigma_{Iz''}\sin\chi) \qquad (15b)$$

$$+\frac{W_{II}}{(b_{A1}-b_{II})}(\sigma_{IIx''}\cos\chi+\sigma_{IIz''}\sin\chi)$$

$$T_{x2} = -\frac{W_{IV}\,(b_{x2}-b_{III})}{(b_{x2}-b_{IV})\,(b_{A2}-b_{III})}(\sigma_{IVx''}\cos\chi+\sigma_{IVz''}\sin\chi)$$

$$+\frac{W_{III}}{(b_{A2}-b_{III})}(\sigma_{IIIx''}\cos\chi+\sigma_{IVz''}\sin\chi) \qquad (16b)$$

Or

$$T_{x1} = -\frac{W_I (b_{x1}+b_{A2})}{(b_{A1}+b_{A2})(b_{x1}-b_I)} (\sigma_{Ix'} \cdot \cos\chi + \sigma_{Iz'} \cdot \sin\chi) \quad (15c)$$

$$+\frac{W_{IV} (b_{x2}-b_{A2})}{(b_{A1}+b_{A2})(b_{x2}-b_{IV})} (\sigma_{IVx'} \cdot \cos\chi + \sigma_{IVz'} \cdot \sin\chi)$$

$$T_{x2} = -\frac{W_{IV} (b_{x2}+b_{A1})}{(b_{A1}+b_{A2})(b_{x2}-b_{IV})} (\sigma_{IVx'} \cdot \cos\chi + \sigma_{IVz'} \cdot \sin\chi)$$

$$+\frac{W_I (b_{x1}-b_{A1})}{(b_{A1}+b_{A2})(b_{x1}-b_I)} (\sigma_{Ix'} \cdot \cos\chi + \sigma_{Iz'} \cdot \sin\chi) \quad (16c)$$

Or

$$T_{x1} = -\frac{W_{II} (b_{x1}+b_{III})}{(b_{II}+b_{III})(b_{x1}-b_{A1})} (\sigma_{IIx'} \cdot \cos\chi + \sigma_{IIz'} \cdot \sin\chi)$$

$$+\frac{W_{III} (b_{x1}-b_{II})}{(b_{II}+b_{III})(b_{x1}-b_{A1})} (\sigma_{IIIx'} \cdot \cos\chi + \sigma_{IIIz'} \cdot \sin\chi) \quad (15d)$$

$$T_{x2} = -\frac{W_{III} (b_{x2}+b_{II})}{(b_{II}+b_{III})(b_{x2}-b_{A2})} (\sigma_{IIIx'} \cdot \cos\chi + \sigma_{IIIz'} \cdot \sin\chi)$$

$$+\frac{W_{II} (b_{x2}-b_{III})}{(b_{II}+b_{III})(b_{x2}-b_{A2})} (\sigma_{IIx'} \cdot \cos\chi + \sigma_{IIz'} \cdot \sin\chi) \quad (16d)$$

3. Measurement Equipment for the Wheelset and Converter for Measured Data

Figure 4 schematically shows the measurement wheelset with its measurement equipment and the converter for measured data. This converter transforms the measured data into forces.

The bending stresses, or more precisely the bending strains, are measured by means of strain gauges. The relation between bending

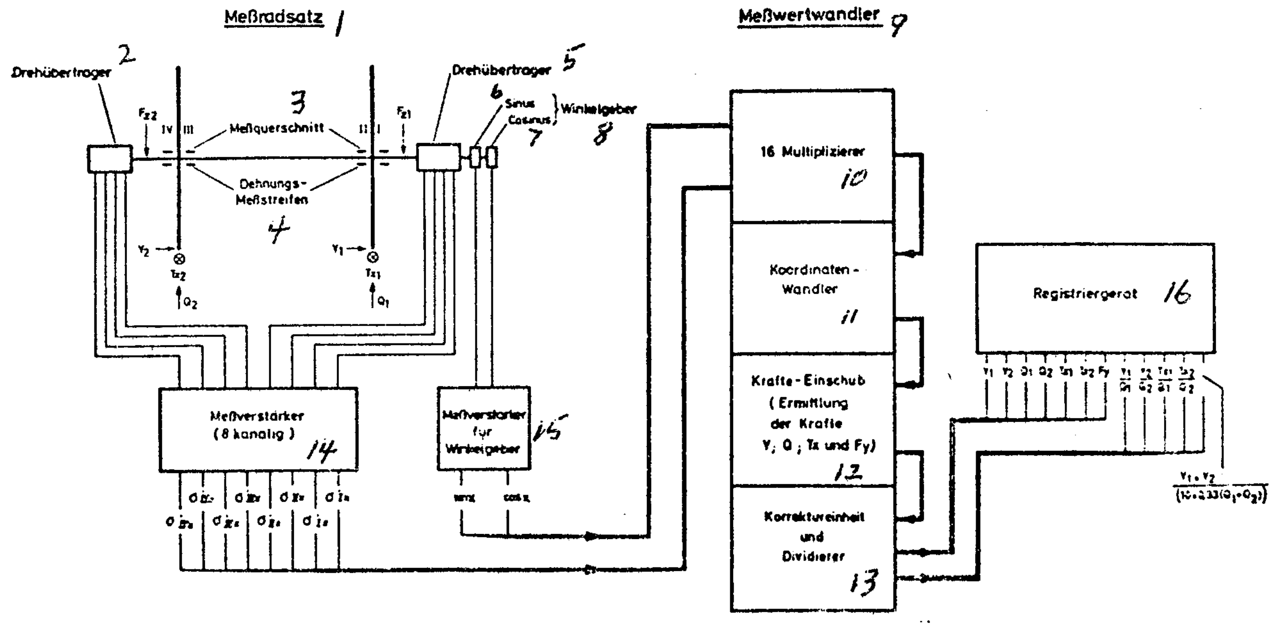


Figure 4: Measurement wheelset with measurement equipment and measurement data converter

1. Measurement wheelset; 2. Rotary transmitter; 3. Measurement cross section; 4. Strain gauge; 5. Rotary transmitter; 6. Sine; 7. Cosine; 8. Angle transducer; 9. Measurement data converter; 10. 16 multipliers; 11. Coordinate converters; 12. Force module (determination of Y, Q, T_x and F_y forces); 13. Correction unit and divider; 14. Measurement amplifier (8 channel); 15. Measurement amplifier for angle transducer; 16. Recording unit

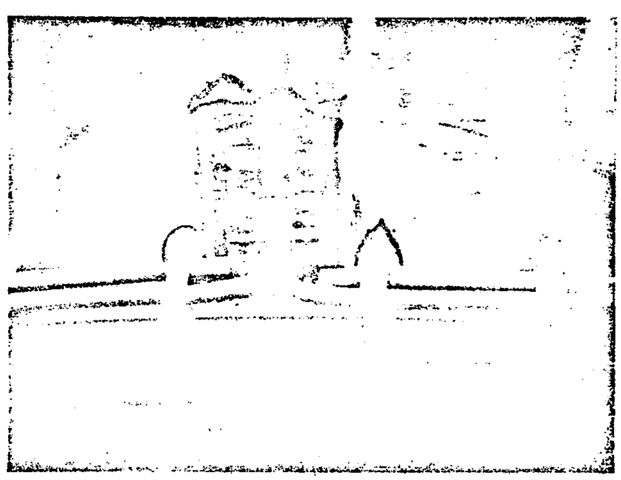


Figure 5: Strain gauge on an axle journal

stress and the measured bending strain is explained in more detail in Section 4.2.2. The strain gauges are glued onto the wheelset axle in the longitudinal direction. Figure 5 shows two strain gauges with their connecting wires, which have been glued to an axle journal of the wheelset axle.

Two flexure measurement points, displaced by 90° with respect to one another, are arranged in each of the four measurement cross sections. Measurement cross sections I and IV each go through one axle journal of the wheelset. Measurement cross sections II and III go through the wheelset axle between the two wheels.

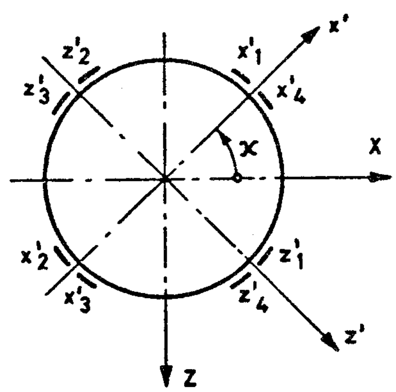


Figure 6: Arrangement of the strain gauges of one measurement cross section

Figure 6 shows the arrangement of the strain gauges of a measurement cross section. The four strain gauges x' are glued on in the following manner: When a bending moment occurs in this measurement plane, either the strain gauges x_1' and x_4' are stretched and strain gauges x_2' and x_3' are compressed, or vice versa - depending on the direction and magnitude of the bending moment in the x'-plane. These four strain gauges x' are connected together in a bridge circuit, in accord with Figure 7. When a bending moment occurs, this bridge is detuned corresponding to the magnitude and direction of the bending stress σ_x'. The like suitably holds for the measurement plane z'.

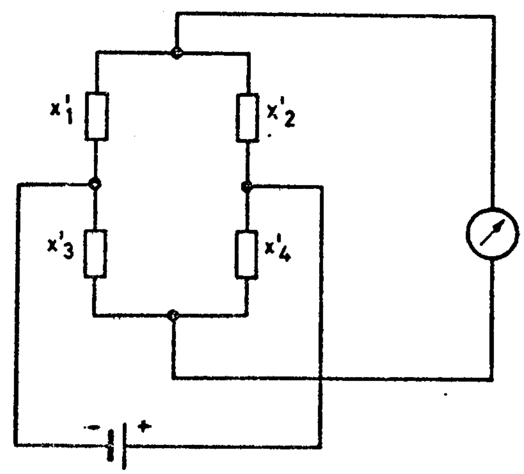


Figure 7: Bridge circuit of a strain gauge point

Each of the eight bending stress measurement points is electrically connected with a measurement amplifier. These measurement amplifiers are located in the measurement section. The bridge detuning amounts to only a few millivolts, and the measurement amplifiers amplify this signal into a measurement signal for further processing. Furthermore, the bridge feed voltages are generated in them.

The electrical connection between the strain gauges on the rotating wheelset and the measurement amplifiers is established through rotary transmitters with slip rings. A rotary transmitter is mounted at each end of the axle shaft. In order to accept the cables between the strain gauges and the rotary transmitters, the wheelset axle has a hollow boring on both ends and is also equipped with radial borings.

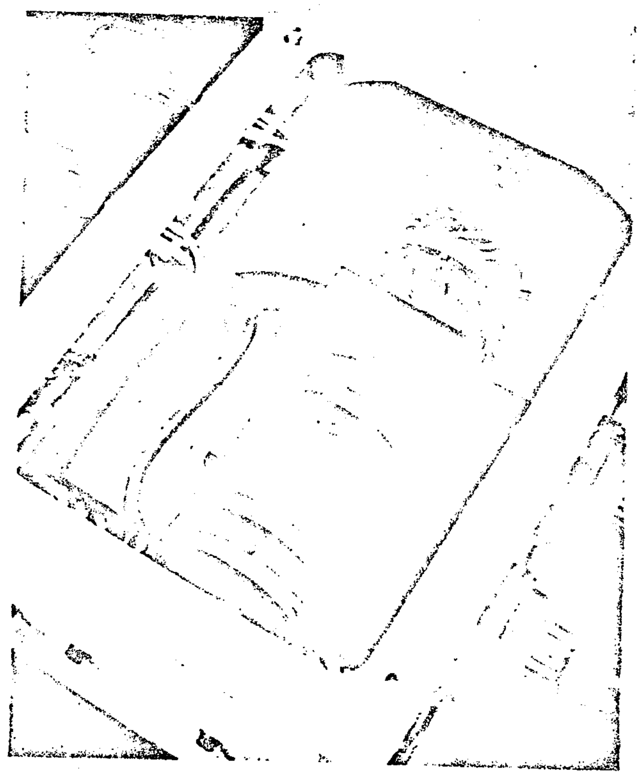


Figure 8: Rotary transmitter and rotation angle transducer

A rotation angle transducer is located in one end of the wheelset. It is coupled with the rotor of the rotary transmitter. The sine and cosine of the wheel rotation angle χ is measured by means of this transducer. One can optionally use a rotation angle transducer with an inductive measurement system or a function potentiometer with a sine-cosine characteristic. Figure 8 shows a rotary transmitter and rotation angle transducer in the transmitter housing, which is flanged onto the bearing box. The rotation angle transducer is connected through cables with a measurement amplifier in the measurement section.

From the measurement amplifiers, the eight measured values of bending stress as well as the sine and cosine values are conducted into the measurement converter (measurement wheelset computer). Here, the forces applied to the wheelset are calculated according to Equations (11) through (16). Furthermore, the quotients Y_1/Q_1, Y_2/Q_2 for derailing safety are here calculated; T_{x1}/Q_1, T_{x2}/Q_2 for non-positive stress are calculated; and the value relevant for the positional stability of the rail, namely

$$\frac{10 + 0{,}33\,(Q_1 + Q_2)}{Y_1 + Y_2}$$

is formed in kilonewtons. This conversion occurs simultaneous with the course of measurement, with a time shift less than 1/10,000 seconds.

Figure 9 left shows the front side of the measurement converter while a measurement wheelset is being calibrated. Its outputs are connected through a matching unit (center) with a UV-recorder (right). The cable rolls are visible in the figure. The converter for the measured data is connected to the measurement wheelset through these cable rolls. The modules of the measured data converter are constructed primarily from analog computer components (4) and have the following functions:

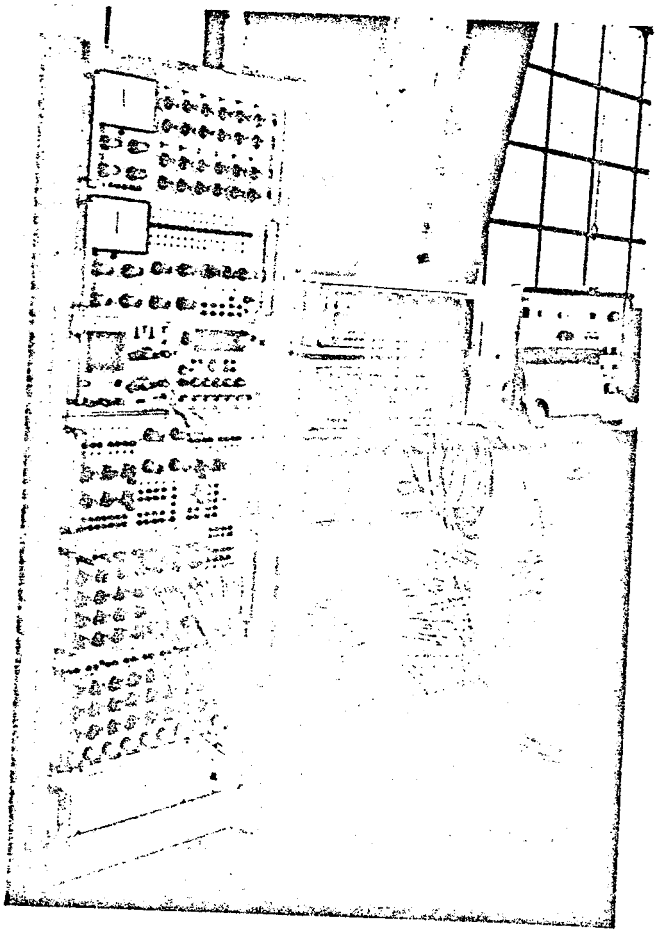


Figure 9: Measurement Data Converter (left) with matching unit (center) and UV oscillograph as recording unit (right)

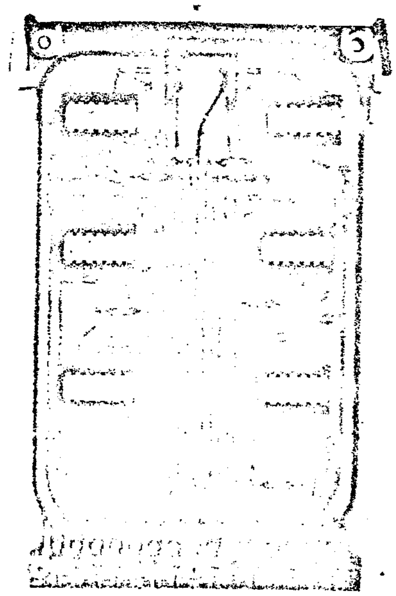


Figure 10: Printed circuit board with 6 operational amplifiers for adding and subtracting measured values

In the multiplier module, the eight measured values of bending stress are multiplied by analog means with the sine and cosine of the wheel rotation angle. The 16 electrical voltages, which are proportional to the products of the measured bending stresses and the angular functions, are then conducted into the coordinate converter and are there added or subtracted in accord with Equations (9) and (10). At the output of the coordinate converter thus appear the bending stresses σ_x or σ_z, relative to the coordinate system x, z. The additions and subtractions are implemented with appropriately wired operational amplifiers. Figure 10 shows six operational amplifiers used for this purpose, and mounted on one printed circuit board. In the force module, the forces Y, Q, T_x, and $(Y_1 + Y_2)$ are calculated in accord with Equations (11) through (16). The results of the coordinate converter are here directly used for the bracketted expressions for σ_x and σ_z. The coefficients resulting from the dimensions of the wheelset and the position of the strain gauges are calculated manually during calibration by means of Equations (11) through (16). They are set by means of potentiometers on the front plate of the force module. The quotients Y/Q etc. are formed in the lower module. Furthermore, this module contains a device, which corrects deviations from the equations resulting from non-central force applications in the bearing boxes, etc. These corrections are formed by adjusting empirically determined correction values.

All the modules contain a number of auxiliary electrical devices, by means of which adjustments, functions, and accuracy can be quickly tested. An auxiliary module also serves this purpose. Among other things, this module contains an oscilloscope and a digital voltmeter.

The calculated forces and force quotients are, as a rule, recorded on chart paper during a run, by means of an analog recording unit, for example a UV-oscillograph. But they can also be made available on an analog magnetic tape for subsequent evaluation, or they can be evaluated immediately in an automatic evaluation device.

4. Adjustment and Calibration of the Measurement Equipment

4.1 Calibration Equipment

Before a measurement wheelset is used for measurement runs, it is necessary to apply various defined forces to it, so that the measurement amplifier and measured data converter can be adjusted in such a fashion that the forces determined by the measurement wheelset correspond as closely as possible to the forces introduced. For this calibration process, a calibration device for the Y- and Q-forces has been developed in the Federal Railway Experimental Institute at Minden. The wheelsets built into the vehicle can be calibrated by means of this device. The device is schematically shown in Figure 11. A device for calibrating the T_x forces is currently being developed.

The vehicle is run on the track so that the measurement wheelset comes at rest on the supports of the calibration device. The supports are mounted on rollers and can be moved transverse to the track. One of these supports, however, is laterally fixed through a gauge pin. Y-forces can be applied on the other support by means of a hydraulic lift (Lukas lift). These Y-forces are likewise conducted over a gauge pin, as can be seen from Figure 12. The gauge pins have strain gauges glued on them; these can measure the magnitudes of the introduced Y-forces and can compare these

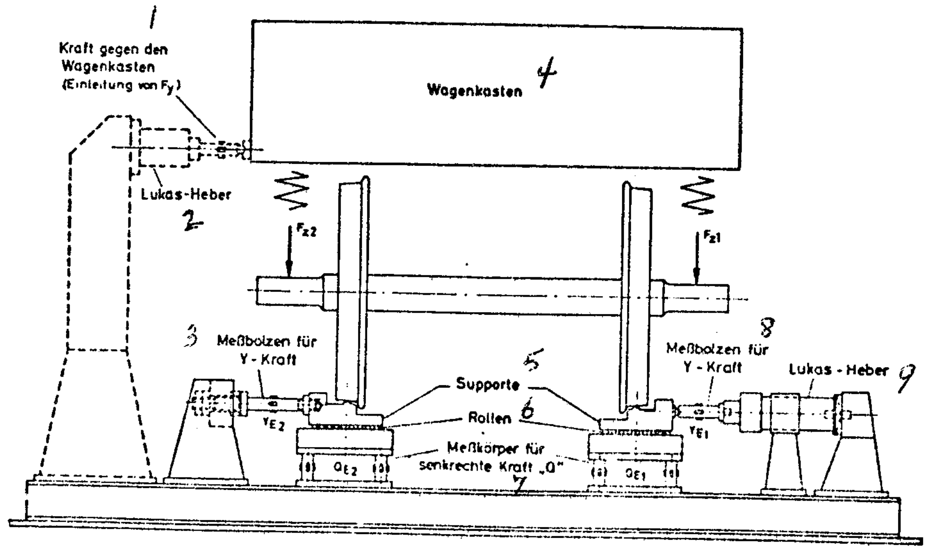


Figure 11: Schematic diagram of the calibration equipment
1. Force against the body (introduction of F_y); 2. Lukas lift; 3. Gauge pin for Y-force; 4. Body; 5. Supports; 6. Rollers; 7. Measurement bodies for vertical force "Q"; 8. Gauge pins for Y-force; 9. Lukas lift

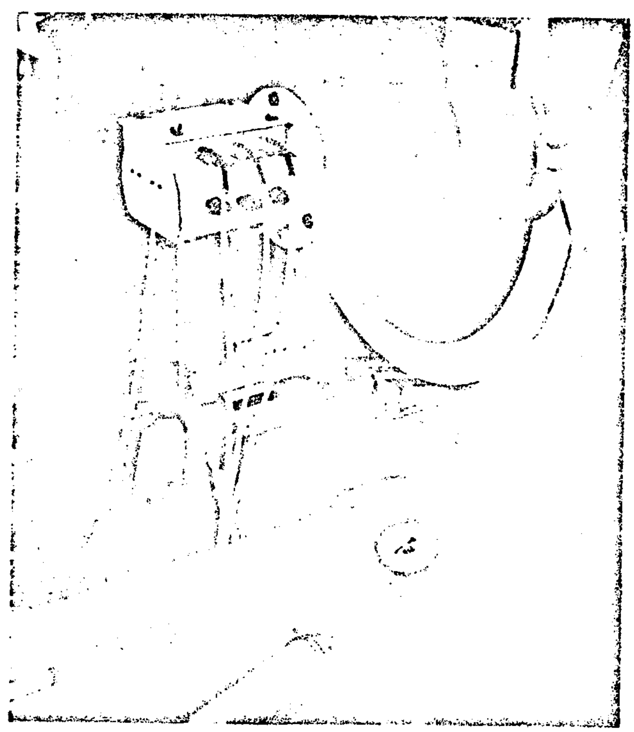


Figure 12: Vehicle on the calibration device

with the forces which are simultaneously determined with the measurement wheelset. Through another hydraulic lift, forces can optionally be applied to the vehicle frame or the truck frame. These forces act wholly or partly as F_y-forces on the measurement wheelset. In the vertical direction, the supports are supported over measurement bodies with strain gauges, by means of which the introduced Q-forces are measured and are compared with the measured Q-values of the wheelset. The Q-forces are varied by lifting and loading the vehicle or the truck; this is done by means of hydraulic lifts or lifting spindle jacks or by applying weights.

4.2 Calibration Procedure

4.2.1 Adjusting the Coefficient Potentiometer

Before beginning calibration, the dimensions of the wheelset corresponding to Figure 2 must be recorded, especially the precise position of the strain gauges. The coefficients must be calculated with these dimensions, after Equations (11) through (16), and the coefficient potentiometer of the measurement data converter must be adjusted.

4.2.2 Conversion of the Measured Bending Strains into Bending Stresses

With this measurement method, it is not the actual bending stresses (= stresses at the surface of the wheelset axle under the strain gauge), but the theoretically expected bending stresses, which result from the action of outside forces and the dimensions of the wheelset. It is these that are involved in the moment equations (1) through (8) and in the equations for the forces (11) through (16) that are derived therefrom. The actual bending stresses σ_{actual} at the gluing point of the strain gauges, however, often deviate considerably from theoretically expected bending stresses, for the following reasons:

a) In areas of strong cross section transitions and in the neighborhood of press fits, such as, for example, at the axle journal, the notch effect causes the force to flow through the cross section with a different distribution than in the case of axles with the same diameter but without a press fit. The path of the forces can then be computed at best approximately.

b) The borings for accepting these measurement cables interfere with the course of the force flux. This may also have an effect on the gluing point of the strain gauges.

Furthermore, the actual bending stress can also be calculated only approximately from the measured values of the bending strain. This calculation proceeds by Hooke's law, by multiplying the strain by the elastic modulus. A precise calculation of the actual bending stress, however, is not possible for the following reasons:

c) The elastic modulus of the axle shaft is known only approximately.

d) The actual strain cannot be derived precisely from the measured value of the strain, because the k-value of the strain gauge is given by the manufacturer only as a representative value of a batch. This value is based on random samples and not as a binding value for individual units. However, this value specifies the relationship of the resistance change to the resistance on the one hand, and of the length change to the length on the other hand ($k = [\Delta R/R] \,/\, [\Delta l/l]$).

Because of these uncertainties, the relationships between the measured values for bending strain and the theoretically expected bending stresses are determined in preliminary experiments and are taken into account by a correction factor β. For this purpose, definite forces are applied to the measurement wheelset, which is installed in the vehicle, and the bending strains are measured. The theoretically expected bending stresses are calculated from the forces, according to the following equations. These equations are derived from Equations (1) through (8):

$$\sigma_{Iz} = -Q_1 \frac{(b_{z1}-b_I)\,(b_{z2}+b_{A1})}{W_I\,(b_{z1}+b_{z2})} - Q_2 \frac{(b_{z1}-b_I)\,(b_{z2}-b_{A2})}{W_I\,(b_{z1}+b_{z2})} + Y_1 \frac{r_1\,(b_{z1}-b_I)}{W_I\,(b_{z1}+b_{z2})} + Y_2 \frac{r_2\,(b_{z1}-b_I)}{W_I\,(b_{z1}+b_{z2})} \qquad (17)$$

$$\sigma_{IVz} = - Q_2 \frac{(b_{z2}-b_{IV})\,(b_{z1}+b_{A2})}{W_{IV}\,(b_{z1}+b_{z2})} \tag{18}$$

$$- Q_1 \frac{(b_{z2}-b_{IV})\,(b_{z1}-b_{A1})}{W_{IV}\,(b_{z1}+b_{z2})}$$

$$- Y_2 \frac{(b_{z2}-b_{IV}).\,r_2}{W_{IV}\,(b_{z1}+b_{z2})} - Y_1 \frac{(b_{z2}-b_{IV}).\,r_1}{W_{IV}\,(b_{z1}-b_{z2})}$$

$$\sigma_{IIz} = - Q_1 \frac{(b_{z2}+b_{II})\,(b_{z1}-b_{A1})}{W_{II}\,(b_{z1}+b_{z2})} \tag{19}$$

$$- Q_2 \frac{(b_{z1}-b_{II})\,(b_{z2}-b_{A2})}{W_{II}\,(b_{z1}+b_{z2})}$$

$$- Y_1 \frac{(b_{z2}+b_{II}).\,r_1}{W_{II}\,(b_{z1}+b_{z2})} + Y_2 \frac{(b_{z1}-b_{II}).\,r_2}{W_{II}\,(b_{z1}+b_{z2})}$$

$$\sigma_{IIIz} = - Q_2 \frac{(b_{z1}+b_{III})\,(b_{z2}-b_{A2})}{W_{III}\,(b_{z1}+b_{z2})} \tag{20}$$

$$- Q_1 \frac{(b_{z2}-b_{III})\,(b_{z1}-b_{A1})}{W_{III}\,(b_{z1}+b_{z2})}$$

$$+ Y_2 \frac{(b_{z1}+b_{III}).\,r_2}{W_{III}\,(b_{z1}+b_{z2})} - Y_1 \frac{(b_{z2}-b_{III}).\,r_1}{W_{III}\,(b_{z1}-b_{z2})}$$

The actual bending stresses σ_{actual} are calculated from the measured values of the bending strains ε_M in the preliminary experiment, according to the extended Hooke's law

$$\sigma_{ist} = \varepsilon_M E \, \frac{1}{n} \cdot \frac{k_B}{k_{DMS}} \tag{21}$$

Here, the k-value of the strain gauge k_{DMS}* is assumed in accord with the manufacturer's specifications, and the elastic modulus

*DMS = strain gauge. Translator

of the axle shaft is assumed at $2.1 \cdot 10^7$ N/cm^2. Errors according to c) and d) are here deliberately accepted and are subsequently taken into account through the correction factor β(see below). In the previous equation, we also have n = number of active strain gauges per measurement bridge = 4 in the present case, and k_B = k-value of the strain gauge amplifier, generally 2.

The theoretically expected bending stresses are determined according to one of the Equations (17) through (20). The bending stresses σ_{actual} are calculated from the measured strains according to Equation (21). The correction factors

$$\beta = \frac{\sigma}{\sigma_{ist}} \tag{22}$$

is formed from these bending stresses. The correction factors of several experiments are averaged, and this is done separately for each measurement point. The amplification factors of the eight

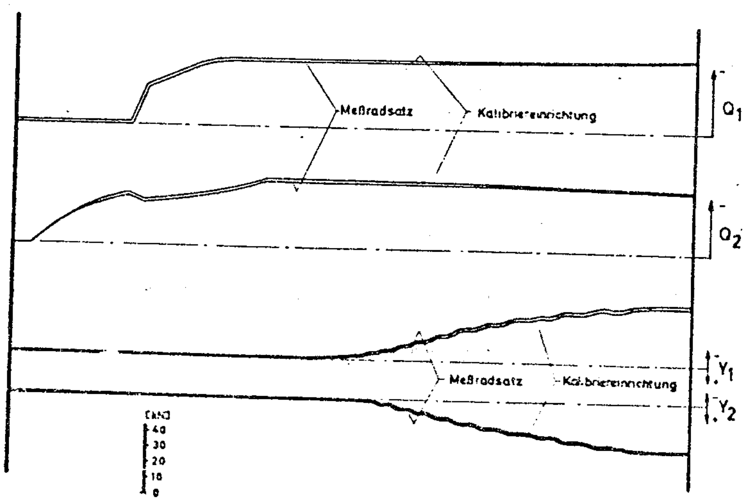


Figure 13: Strip chart with measured values of the measurement wheelset and the calibration device

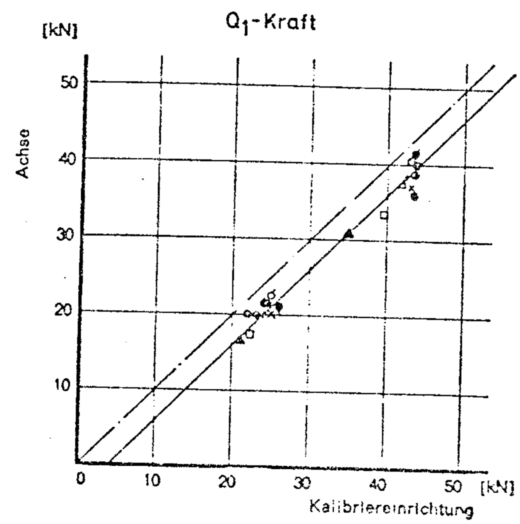


Figure 14: Result of comparative measurements between the measurement wheelsetand the calibration device for various wheel rotation angles and contact points

measurement amplifiers are subsequently adjusted, taking into account the correction factors β, so that all the measurement amplifiers deliver equal electric output voltages for the same values of σ.

The bending strains are consequently converted into the theoretically expected bending stresses according to Hooke's law, taking into account the correction values that were determined in the preliminary experiment.

Remark: The correction factors can also be approximately determined or checked in a roll-out experiment of the vehicle on a good tangent track. The static wheel loads must here be inserted for Q_1 and Q_2 into Equations (17) through (20).

4.2.3 Comparison of Measured Forces with the Forces Actually Introduced

After the measurement amplifier and the coefficient potentiometer are adjusted, forces of various magnitudes and combinations are introduced into the measurement wheelset. They are measured and compared with one another, both with the measurement wheelset and with the measurement equipment of the calibration apparatus. In particular, this is done for various wheel rotation angles and various wheel contact points, which are laterally displaced with respect to one another. For comparison, the measured values are suitably recorded on a strip chart. This provides the best opportunity for detecting deviations, when the traces of the measured values of the measurement wheelset as well as the traces of the calibration equipment are superposed on one another. Figure 13 shows such a recording from the calibration of a measurement wheelset under a passenger car. Loads Q and Y were applied sequentially. For this comparison, the strip chart shows that the measured values of the measurement wheelset deviated from those of the calibration equipment by 2 kN on the average.

Upon conclusion of the comparison measurements, the deviations may show a definite trend. In that case, the adjustments of the measurement amplifier and of the measurement converter are appropriately corrected and are checked by new comparison measurements.

4.2.4 Equalization of the Zero Position before Measurement Runs

Before each measurement run, the zero positions of the eight measurement amplifiers must be carefully equalized. This is done during a run on a good tangent track. Here, the measured values of the bending strains are symmetrically adjusted to zero by means of an oscilloscope. These bending strains occur sinusoidally with wheel rotation if the external forces are equal. This zero adjustment must be performed once or twice daily.

5. Errors in Measured Values and Corrections

As with every other measurement method, here too various effects reduce the accuracy of the measurement. Some of these measurement errors can be compensated in whole or in part. As an example,

Figure 14 shows the results of the Q-force calibration of a wheelset under a passenger car. The scatter of the values was created by various wheel rotation angles and contact points.

5.1 Effect of the Self-Weight of the Wheelset

In Figure 14, the measured values of the wheelset are too small by a constant amount, as compared to those of the calibration equipment, over the entire range. This error is caused by the static self-weight of the wheelset. This weight is not applied at the contact point of the forces F - as is assumed in the equations of Section 2. The component of the wheelset axle weight exists as a distributed load. The component of the wheel body weight causes only slight bending moments in the axle shaft.

The converter for measured data contains a device to compensate the effect of the static self-weight. By introducing an electrical voltage of appropriate magnitude and polarity, this device equalizes the error during the course of computations.

5.2 Measurement Errors Resulting from the Inertial Forces of the Wheelset

No studies have yet been made concerning measurement errors resulting from the inertial forces of the wheelset. Comparison measurements on this point will be performed as soon as possible. These measurements will involve the stationary force measurement points on the rail and measurements by means of a wheelset from the vehicle.

5.3 Measurement Errors from Rotation of the Wheelset

The effect of the wheel rotation angle on the measurement accuracy is primarily determined by the conformity of the sine- and cosine-measurement device. With inductive systems, conformities of $\pm$ 1.5% are achieved. With potentiometric systems, conformities of $\pm$ 0.5% and better are achieved. However, according to present experience, potentiometric angle measurement systems are suitable only conditionally for fast runs.

5.4 Effect of Temperature on the Accuracy of Measurement

As far as can be determined, the temperature of the measurement wheelset does not exert a major effect on the accuracy of measurement. The ambient temperature of the converter for measured data should be held reasonably constant (about $\pm$ $8^{\circ}C$ in the measurement section) - as is generally true for electronic measurement units. If the temperature changes exceed this range, new equalization of certain devices is recommended, especially of the multipliers. Equalization takes about 10 minutes and can also be performed during a run.

5.5 Measurement Errors Resulting from Changes in the Contact Points

A lateral displacement of the wheel contact point on the track may be caused, for example, by the hunting movement of the wheelsets. This causes measurement errors, because the forces Q, Y, and T_x are no longer applied at the theoretically assumed points. The dependence of the measurement error on the contact points was determined for a wheelset of a passenger car, both by measurements as well as by calculations on a digital computer. This dependence involves the following:

a) A lateral displacement of the contact points does not affect the measurement accuracy of the vertical force Q.

b) A lateral displacement of the contact point of wheel 1 by 10 mm causes an error of 2.1% in the measured value of the lateral force Y_1. These 2.1%, however, refer to the magnitude of Q_1, because the measurement error of Y - except for the position of the contact point - depends only on the respective wheel load Q, and not on Y itself. In this example, the measurement error of Y_1, for a wheel load Q_1 of 100 kN and a lateral displacement of the contact point by 10 mm, amounts to 2.1 kN, regardless of the magnitude of Y_1 itself. This measurement error appears in the measured value Y as an apparent pressure force between the wheel flange and the rail, when the contact point is displaced in the direction of the wheel flange, and inversely in the opposite direction. Within the relevant range, the measurement error increases linearly with displacement. A lateral displacement of the contact point on wheel 2 causes a measurement error only for the Y-measurement value of wheel 2, but not for the measurement values of wheel 1.

The measurement error due to a change in the contact points can at this time not yet be compensated. When evaluating the measured results, however, it is known whether the wheelset runs on the right or left on the rail. In these cases, approximate corrections can be made during evaluation.

5.6 Mutual Effects of the Measurements of Q and Y

The effects on the Y-measurements through Q lie below 4%. The effects on the Q-measurements through Y lie below 2%.

5.7 Measurement Errors Due to Tilting of the Bearing Box as well as Off-center and Eccentric Force Application in the Bearing Box

With several types of running gear, considerable error measurements result when a force F_y acts between the wheelset and the vehicle or truck. For example, when a freight car runs along a curved track, its body is deflected towards the outside of the curve, relative to the wheelset, because of centrifugal force. In that case, the supporting leaf springs no longer press vertically on the housing of the bearing box. In contrast to swivel-joint roller bearings, the rigid roller axle bearings of this vehicle cannot avoid edge* pressures: One additional force couple is always introduced into the wheelset shaft in the area of the bearing box.

With some running gears, the spring load F no longer is applied at the center of the bearing, when F_y-forces appear; the distances

*The German word also means "tilt". Translator

b_{z1} and b_{z2} (Figure 2) are no longer constant. Furthermore, for various running gears, error measurements result when the force F_y is not applied concentrically to the axle center.

In all these cases, additional bending moments occur in the wheelset axle and thus enter the measurement. However, the bending moments are converted into forces, in the measurement data converter, only according to Equations (11) through (16). These equations do not consider these cases with additional bending moments, so that the calculated forces are wrong.

In principle, it is possible to avoid such measurement errors by increasing the amount of measurement and calculation, as will be shown below:

5.7.1 Compensation of the Measurement Errors by Additional Measurement Cross Sections

Each new variable force and each new force couple and each variable application point can be accounted for by introducing, for each case, one additional measurement cross section and by appropriately expanding the system of equations and the measured data converter. In this way, the measured values of the forces Y, Q, and T_x correspond to the actual forces.

For example, if the spring loads F of a running gear do not always act on the same point during a run, but act with variable distances b_{z1} and b_{z2}, these new variables can be eliminated by setting up an additional measurement cross section on each axle journal.

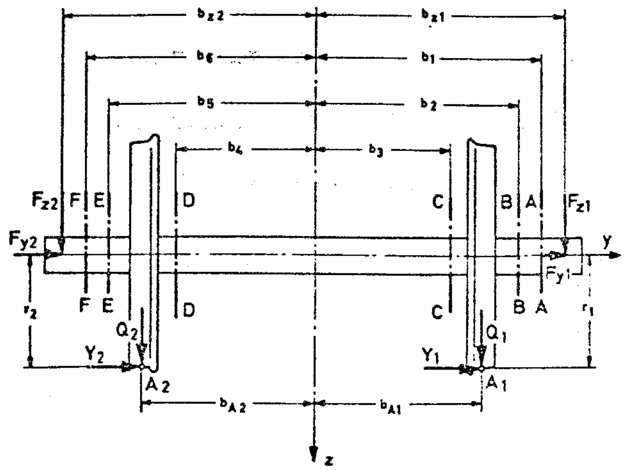


Figure 15: Schematic diagram of a measurement wheelset with 6 measurement cross sections, to avoid errors due to a migrating contact point of the spring loads F_{z1} and F_{z2}

Figure 15 schematically shows the wheelset shaft with its six measurement cross sections. The equilibrium conditions for each measurement cross section A through F yield the equations for the forces (without taking into account the wheel rotation). These equations are suitably set up like Equations (1) through (8):

$$Q_1 = \sigma_{Az}\frac{W_A}{(b_1-b_2)} - \sigma_{Bz}\frac{W_B}{(b_1-b_2)} - \sigma_{Cz}\frac{W_C}{(b_3+b_4)} + \sigma_{Dz}\frac{W_D}{(b_3+b_4)} \tag{23}$$

$$Q_2 = \sigma_{Fz}\frac{W_F}{(b_6-b_5)} - \sigma_{Ez}\frac{W_E}{(b_6-b_5)} - \sigma_{Dz}\frac{W_D}{(b_3+b_4)} + \sigma_{Cz}\frac{W_C}{(b_3+b_4)} \tag{24}$$

$$Y_1 = -\sigma_{Az}\frac{W_A(b_2-b_{A1})}{r_1(b_1-b_2)} + \sigma_{Bz}\frac{W_B(b_1-b_{A1})}{r_1(b_1-b_2)} - \sigma_{Cz}\frac{W_C(b_{A1}+b_4)}{r_1(b_3+b_4)} + \sigma_{Dz}\frac{W_D(b_{A1}-b_3)}{r_1(b_3+b_4)} \quad (25)$$

$$Y_2 = \sigma_{Fz}\frac{W_F(b_5-b_{A2})}{r_2(b_6-b_5)} - \sigma_{Ez}\frac{W_E(b_6-b_{A2})}{r_2(b_6-b_5)} + \sigma_{Dz}\frac{W_D(b_{A2}+b_3)}{r_2(b_3+b_4)} - \sigma_{Cz}\frac{W_C(b_{A2}-b_4)}{r_2(b_3+b_4)} \quad (26)$$

However, with the present state of measurement technology, measurements based on the preceding considerations can in practice be performed only with considerable inaccuracies. This is the case when the distances of the measurement cross sections from one another can only be small for reasons of space. For example, the distances of the two measurement cross sections on an axle journal suffer from this limitation. This compensation method with additional measurement cross sections is scarcely used at all at this time.

5.7.2 Compensation of the Measurement Errors by means of Function Generators

The utilization of function generators was able to achieve better results in compensating errors due to the introduction of undefined forces into the axle bearing. During calibration, it is empirically determined how large are the measurement errors of Y, Q, and T_x in dependence on F_y. More precisely: in dependence on the already erroneous measured value $(Y_1 + Y_2)$. Figure 16 shows such dependencies for the measured value Q_1. This value was measured when calibrating a wheelset under a two-axle freight car. In this example, the measurement error Q_1 is only a function of $(Y_1 + Y_2)$, but not of Q_1 itself. The measurement error $(Q_1 - Q_{1\ calibration})$ consequently can simply be represented as a function of $(Y_1 + Y_2)$, see Figure 17. The function generator equalizes the measurement error approximately according to the plotted straight line segments.

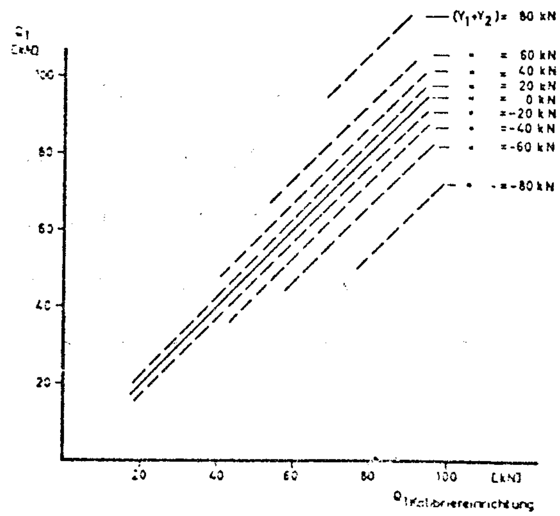
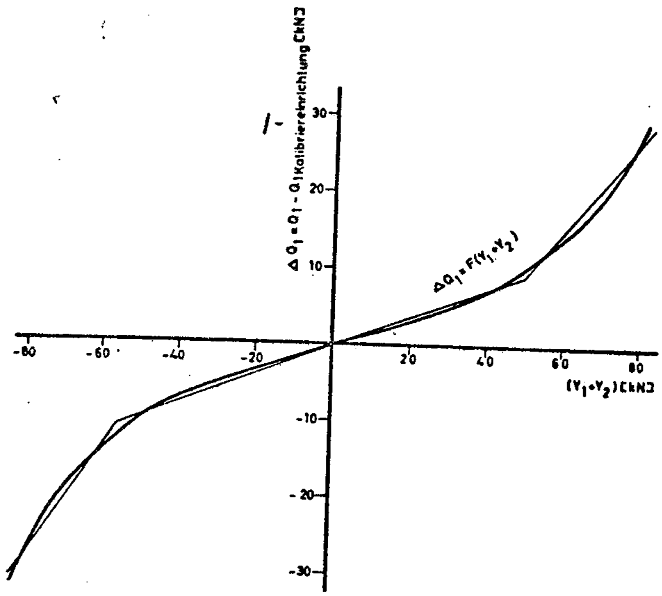


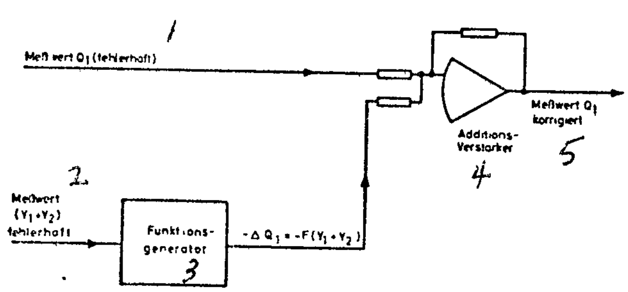
Figure 16: Ratio of the measured values Q_1 to the introduced forces $Q_{1\ calibration}$ in dependence on $(Y_1 + Y_2)$, when undefined forces are introduced into the bearing box of a freight car wheelset



1. Calibration device

Figure 17: Measurement error (Q_1 - $Q_{1\ \text{calibration}}$) in dependence on ($Y_1 + Y_2$), when undefined forces are introduced into the axle bearing of a freight car wheelset.

The measurement data converter contains function generators by means of which the measurement errors are compensated, as shown in Figure 18.



1. Measured value Q_1 (erroneous);
2. Measured value ($Y_1 + Y_2$) erroneous;
3. Function generator;
4. Addition amplifier;
5. Measured value Q_1 corrected

Figure 18: Correction of the erroneous measured value Q_1 by $-F(Y_1 + Y_2)$

5.8 Measurement Errors through Transmission of the Measurement Data from the Rotating Wheel

Many measurement errors reduce the accuracy of the measurement in such a measurement method. Among these errors, especially those should be mentioned which result from the transmission of the strain measurement data from the axle shaft to the vehicle. The currently used rotary transmitters achieve fairly satisfactory values, but improvements would be desirable. Particularly welcome would be a transmission method - whether through slip rings or wireless - where the axle shafts need no longer be equipped with borings in order to accept cables.

6. Forces Which Exert an Additional Bending Stress on the Axle Shaft

In Section 2, we listed the forces that act on the wheelset, namely the forces between the wheel and the rail and the forces in the axle bearings (together with the "misguided" forces after 5.7). But sometimes additional forces act on the wheelset: Acceleration forces on the driving wheelsets, braking forces from shoe brakes or disc brakes, forces from the shock absorbers, etc. Equations (11) through (16) then no longer adequately describe these circumstances. If these forces are neglected, erroneous measurements would result. If these additional forces can be measured directly, it is possible either to include them in the system of equations or to determine empirical correction values as a function of these supplementary forces, and to introduce these values into the calculations.

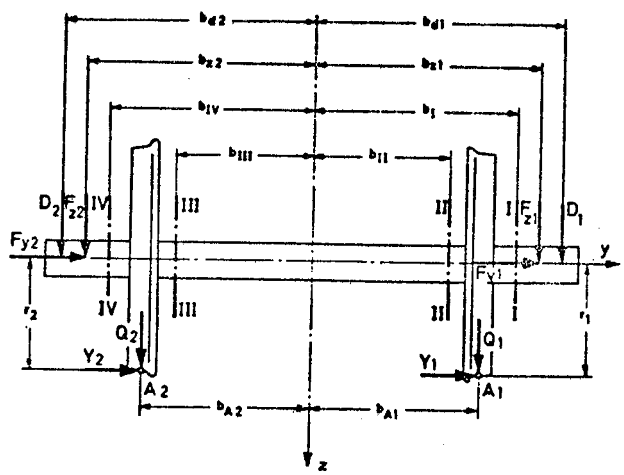


Figure 19: Schematic diagram of a measurement wheelset subject to damper forces D_1 and D_2

As an example of this type, it will be shown how to take into account the forces which come from the hydraulic vibration dampers and reach the axle shaft of a Minden-Deutz truck. The vibration dampers are linked to the bearing box, outside the contact points of the spring loads F. They conduct the damping forces D_1 and D_2 into the axle shaft, as is shown schematically in Figure 19. The magnitude and direction of the damper forces were measured by means of strain gauges on the amount of these vibration dampers. The damper forces D_1 and D_2 were included in the moment equations (1) through (4). By appropriate transformation, the following equations were obtained for the forces between the wheel and the rail:

$$Q_1 = - \sigma_I \frac{W_I}{(b_{z1} - b_I)} - \sigma_{II} \frac{W_{II}}{(b_{II} - b_{III})} \qquad (27)$$

$$+ \sigma_{III} \frac{W_{III}}{(b_{II} + b_{III})} + D_1 \frac{(b_{d1} - b_{z1})}{(b_{z1} - b_I)}$$

$$Q_2 = - \sigma_{IV} \frac{W_{IV}}{(b_{z2}-b_{IV})} - \sigma_{III} \frac{W_{III}}{(b_{II}+b_{III})} \tag{28}$$

$$+ \sigma_{II} \frac{W_{II}}{(b_{II}+b_{III})} + D_2 \frac{(b_{d2}-b_{z2})}{(b_{z2}-b_{IV})}$$

$$Y_1 = + \sigma_{I} \frac{W_{I} (b_{z1}-b_{A1})}{r_1 (b_{z1}-b_{I})} \tag{29}$$

$$- \sigma_{II} \frac{W_{II} (b_{A1}+b_{III})}{r_1 (b_{II}+b_{III})}$$

$$+ \sigma_{III} \frac{W_{III} (b_{A1}-b_{II})}{r_1 (b_{II}+b_{III})}$$

$$- D_1 \frac{(b_{I}-b_{A1}) \cdot (b_{d1}-b_{z1})}{r_1 (b_{z1}-b_{I})}$$

$$Y_2 = - \sigma_{IV} \frac{W_{IV} (b_{z2}-b_{A2})}{r_2 (b_{z2}-b_{IV})} \tag{30}$$

$$+ \sigma_{III} \frac{W_{III} (b_{A2}+b_{II})}{r_2 (b_{II}+b_{III})}$$

$$- \sigma_{II} \frac{W_{II} (b_{A2}-b_{III})}{r_2 (b_{II}+b_{III})}$$

$$+ D_2 \frac{(b_{IV}-b_{A2}) \cdot (b_{d2}-b_{z2})}{r_1 (b_{z2}-b_{IV})}$$

The equations for T_{x1} and T_{x2} (15) through (16) in this case hold unchanged, because the damper forces act only in the yz-plane.

The measured values of the damper forces were introduced into the measured value converter and were processed according to the above equation

In similar fashion it should be possible, and will be tried out in the near future, to sense acceleration forces by torsion measurements and braking forces by appropriate measurements (different ones depending on the construction type). Their effects on the measured values between the wheel and the rail will be eliminated in the measured data converter.

7. Combination of the "Wheelset Axle" Method and the "Wheel Disc" Method

With different driving wheelsets it may sometimes not be possible, for reasons of space, to affix strain gauges between the two wheel discs. In this case, the "wheelset axle" measurement method can be combined with the "wheel disc" measurement method (5). The forces Y_1 and Y_2, which are directly measured at the wheel disc, the measured data for the bending strains from measurement cross sections I and IV, and the sine and cosine measurement data of the wheel rotation angle are here conducted to the measurement data converter. The measurement cross sections II and III are omitted. From the above-mentioned measured values, the measurement data converter calculates the vertical forces Q according to the following relations (compare with Figure 2):

$$\begin{aligned} Q_1 = &- \sigma_{Iz} \cdot W_I \frac{(b_{z1}+b_{A2})}{(b_{z1}-b_I)\cdot(b_{A1}+b_{A2})} \\ &+ \sigma_{IVz} \cdot W_{IV} \frac{(b_{z2}-b_{A2})}{(b_{z2}-b_{IV})\cdot(b_{A1}+b_{A2})} \\ &+ Y_1 \cdot \frac{r_1}{(b_{A1}+b_{A2})} + Y_2 \frac{r_2}{(b_{A1}+b_{A2})} \end{aligned} \quad (31)$$

$$\begin{aligned} Q_2 = &- \sigma_{IVz} \cdot W_{IV} \frac{(b_{z2}+b_{A1})}{(b_{z2}-b_{IV})\cdot(b_{A1}+b_{A2})} \\ &+ \sigma_{Iz} \cdot W_I \frac{(b_{z1}-b_{A1})}{(b_{z1}-b_I)\cdot(b_{A1}+b_{A2})} \\ &- Y_2 \cdot \frac{r_2}{(b_{A1}+b_{A2})} - Y_1 \cdot \frac{r_1}{(b_{A1}+b_{A2})} \end{aligned} \quad (32)$$

The tangential forces T_x are then calculated according to Equations (15c) and (16c).

The measurement data converter at the Minden Experimental Institute are set up for combining both methods.

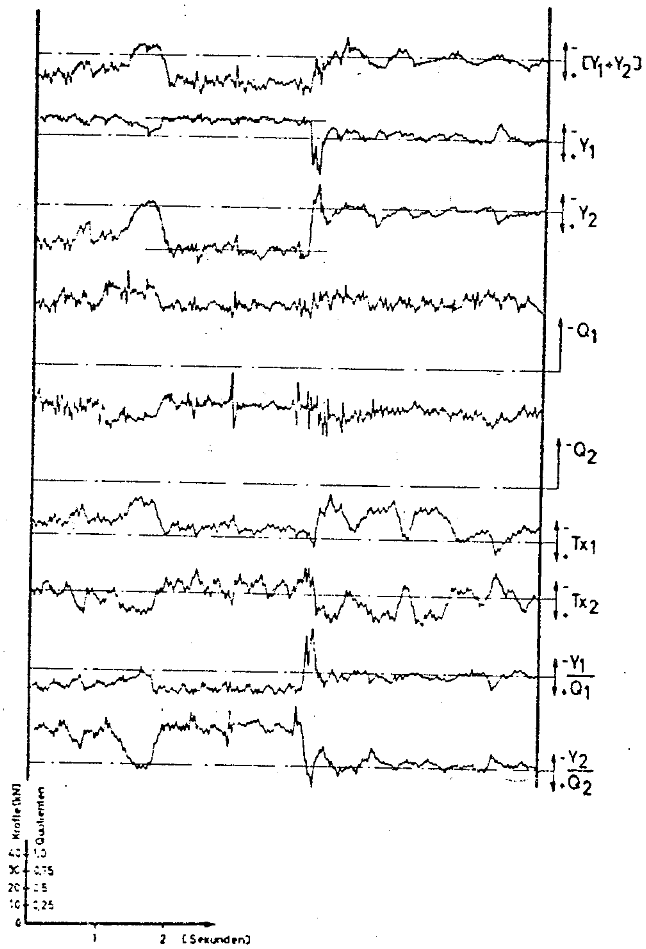


Figure 20: Section of a measurement strip chart

8. Measurement Runs with Measurement Wheelset

Figure 20 shows a section of a measurement strip. The measurement wheelset was the leading wheelset in the truck under a passenger car. What is shown is the run-out from a switching curve (right curve).

In the switch, wheel 2 runs at an angle to the outside rail. The guiding force between the wheel and the rail was here Y_2 = 26 kN on the average. The wheelset is here pushed towards the inside of the curve, counter to the guiding force of wheel 1, by Y_1 = - 10 kN on the average. In the switching curve, the wheel load Q_1 is about 5 kN smaller than in the subsequent tangent section, while the wheel load Q_2 is larger. The quotient Y_1/Q_1 in this range is about - 10 kN/-35 kN = 0.28. The quotient Y_2/Q_2 was about +26 kN/-46 kN = -0.57. The tangential forces T_x run approximately opposite with wheels 1 and 2.

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APPENDIX 1.2

SOURCE: ZEV Glas. Ann. (Glaser's Annals, Journal for Railway and Traffic Engineering) 102 (1978) No. 2, February, pp 53-61

TITLE: The Present State of Development of the "Wheelset Axle Method" for Determining the Forces Between the Wheel and the Rail

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DK 625.1.03:629.4

Summary: The "wheelset axle method" was presented to the public five years ago. In the meantime, this measurement method has been used to determine the forces between the wheel and the rail in many technical studies under running conditions. In order to make application of the method more reliable and more simple, the various components had to be developed further. The present paper presents typical influential quantities for this measurement system, estimates influences on the accuracy of the force measurements, and indicates how these influences can be prevented or reduced. Furthermore, a new type of calibration is presented. The possibilities are discussed for adapting the measurement method to other problems involving force measurements.

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1. Introduction

After the wheelset axle method was published in ZEV Glaser Annalen of December 1972 (1)[1], it was used more and more frequently to determine the T_x-, Q-, and Y-forces between the wheel and the rail. Passenger cars, freight cars, motor train units, and a locomotive were studied. Among these were also included the best-known objects of study, the six-axle Fad 150 ore car for heavy transport (2) between the North Sea harbors and the ore-processing industries in the Salzgitter-Braunschweig area, and the electric motor train unit 403, which is used in IC traffic and charter traffic of the German Federal Railway. The method was used not only within the German Federal Railway, but also in joint studies with other railway adminstrations.

Economic considerations raised the questions of higher speeds and higher axle loads. Consequently, it is absolutely necessary to learn the effects of these requirements on the vehicle and on the track. The forces between the wheel and the rail here play an essential role. To understand the mechanical stress on the track, it is not sufficient to measure the forces relatively. Rather, with these basic studies, it is necessary to know the absolute value of the forces at the contact point between the wheel and the rail. The errors which enter into this measurement method will be discussed and estimated below. As a conclusion from these considerations, means will be indicated for

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[1]The numbers in parentheses refer to the list of references at the end of the paper.

eliminating or reducing these errors. This will provide a survey concerning the present state of the "wheelset axle method".

2. Influential Quantities and How They Are Taken into Account

In (1) Section 5, the mechanical and electrical influential quantities were presented in detail, their effects on accuracy of measurement were discussed, and possibilities for compensating them were indicated. Because of the requirement mentioned in the Introduction, namely to determine the forces between the wheel and the rail absolutely with high accuracy, it is necessary to determine the essential influential quantities, to perform a theoretical error estimate, and to find possibilities for reducing the effect of these influential quantities as much as possible.

2.1 Influential Mechanical Quantities

Various motional experiments utilized measurement wheelsets according to the wheelset axle method. With these experiments, it has appeared that bearing box tilt*, undefined contact points of the F-forces, and changes of the wheel contact point are the most important influential mechanical quantities.

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*The German word can also mean "edges". Translator

2.1.1 Effects on the Accuracy of Measurement

The wheelset axle method was first successfully used with Minden-Deutz trucks and with running gear where the contact point of the F-forces was constant and where the tilt of the bearing boxes was negligible. In cases where the contact points of the forces at the bearing box were undefined, as well as in cases involving severe tilt of the bearing boxes, such as occur e.g. with freight car running gears with leaf springs, the resulting measurement error was compensated with function generators. Measurement with such a compensation is no longer possible, when the mass acceleration of the wheelset in the Y-direction is no longer negligibly small compared to the Y-force.

As can be seen from reference (1) equations 13 and 14, the wheel contact points affect the guide force Y. An estimate for the influence of the wheel contact point on Y is obtained by calculating the largest absolute error (reference 3), under the assumption that the remaining quantities of the system of equations are free of error. This yields the following maximum errors $|\Delta Y_1|$ and $|\Delta Y_2|$:

$$|\Delta Y_1| = |\Delta b_{A1}| \cdot \left(- \frac{1}{b_{z1} - b_I} \frac{W_I \cdot \sigma_{Iz}}{r_1} - \frac{1}{b_{II} + b_{III}} \frac{W_{II}\, \sigma_{IIz}}{r_1} + \frac{1}{b_{II} + b_{III}} \frac{W_{III}\, \sigma_{IIIz}}{r_1} \right) \qquad (1)$$

$$|\Delta Y_2| = |\Delta b_{A2}| \cdot \left| \left(- \frac{1}{b_{z2} - b_{IV}} \frac{W_{IV} \cdot \sigma_{IVz}}{r_2} + \frac{1}{b_{II} + b_{III}} \frac{W_{III} \cdot \sigma_{IIIz}}{r_2} - \frac{1}{b_{II} + b_{III}} \frac{W_{II} \cdot \sigma_{IIz}}{r_2} \right) \right| \qquad (2)$$

Comparisons with equations (11) and (12) from reference (1) yields

$$|\Delta Y_1| = |\Delta b_{A1}| \cdot \frac{|Q_1|}{r_1} \tag{3}$$

$$|\Delta Y_2| = |\Delta b_{A2}| \cdot \frac{|Q_2|}{r_2} \tag{4}$$

The following fact thus becomes apparent: When the wheel contact point is changed ($\Delta b_A \neq 0$), the error of the measured guide force, as compared to the guide force calibrated for an ideal wheel contact point, depends on the currently measured wheel contact force Q and wheel radius r. The wheel contact point can easily change by 0.03 m in a curve. With a wheel radius of 0.5 m, the effect of this error would be 6% of the wheel contact force.

2.1.2 Compensation of Influential Mechanical Quantities

Reference (1) indicates how it is possible to compensate the forces which exert an additional bending stress on the axle shaft. This is done by using more measurement planes.

An unbraked idler wheelset will be used to demonstrate that the introduction of each additional measurement cross section on the axle journal can expand the system of equations to such an extent that the measured values Y, Q, and T_x, which are provided by the wheelset computer, correspond to the actual forces at the contact points A_1 and A_2 in the X-, Y-, and Z-direction.

Figure 1 schematically shows a wheelset. The forces F_{k1} and F_{k2} result from a tilt of the bearing boxes. First of all, they introduce a bending moment into the axle shaft. The tilt pressure F_k is

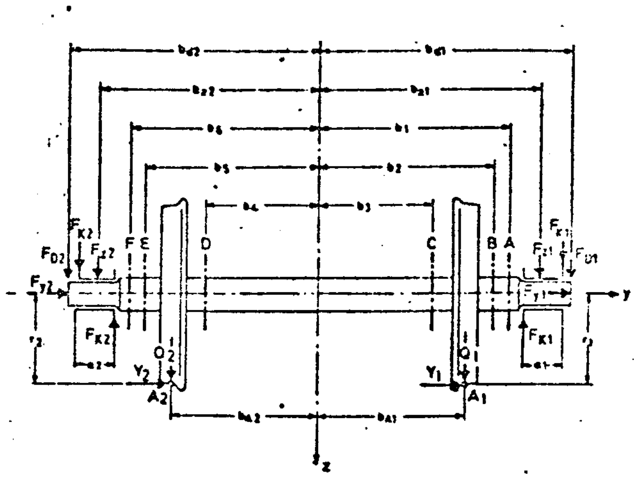

Figure 1: Schematic representation of a measurement wheelset with 6 measurement cross sections

of equal magnitude in each bearing, both inside and outside, but is directed in the opposite sense.

F_{D1} and F_{D2} take into account the forces from hydraulic vibration dampers. Such devices are present with nearly all European passenger car trucks. The additional non-positive connection of the truck frame to the bearing box housing introduces a further moment into the axle shaft.

These load assumptions correspond to reality. Under these assumptions, the following bending moments result in the measurement cross sections A through F for the YZ-plane:

In measurement cross section A

$$M_{Ayz} = \sigma_{Az} \cdot W_A \tag{3a}$$

$$= F_{D1}\,(b_{d1} - b_1) + F_{K1z}\,(b_{z1} - b_1 + \frac{a_1}{2}) - - F_{K1z}\,(b_{z1} - b_1 - \frac{a_1}{2}) + F_{z1} \cdot (b_{z1} - b_1). \tag{3b}$$

the following holds:

$$F_{K1z}\,(b_{z1} - b_1 + \frac{a_1}{2}) - F_{K1z}\,(b_{z1} - b_1) = F_{K1z} \cdot a_1 \qquad (3c)$$

and correspondingly:

$$F_{K2z}\,(b_{z2} - b_6 + \frac{a_2}{2}) - F_{K2z}\,(b_{z2} - b_6 - \frac{a_2}{2}) =$$
$$= F_{K2z} \cdot a_2$$

In measurement cross section B

$$M_{BYz} = \sigma_{Bz} \cdot W_B \qquad (4a)$$
$$= F_{D1}\,(b_{d1} - b_2) + F_{K1z} \cdot a_1 + F_{z1} \cdot (b_{z1} - b_2). \qquad (4b)$$

In measurement cross section C

$$M_{CYz} = \sigma_{Cz} \cdot W_C \qquad (5a)$$
$$= F_{D1}\,(b_{d1} - b_3) + F_{K1z} \cdot a_1 + F_{z1}\,(b_{z1} - b_3) +$$
$$+ Q_1\,(b_{A1} - b_3) - Y_1 \cdot r_1 \qquad (5b)$$

$$= F_{D2}\,(b_{d2} + b_3) + F_{K2z} \cdot a_2 + F_{z2}\,(b_{z2} + b_3) +$$
$$+ Q_2\,(b_{A2} + b_3) + Y_2 \cdot r_2. \qquad (5c)$$

In measurement cross section D

$$M_{DYz} = \sigma_{Dz} \cdot W_D \qquad (6a)$$
$$= F_{D1}\,(b_{d1} + b_4) + F_{K1z} \cdot a_1 + F_{z1}\,(b_{z1} + b_4)$$
$$+ Q_1\,(b_{A1} + b_4) - Y_1 \cdot r_1 \qquad (6b)$$
$$= F_{D2}\,(b_{d2} - b_4) + F_{K2z} \cdot a_2 + F_{z2}\,(b_{z2} - b_4)$$
$$+ Q_2\,(b_{A2} - b_4) + Y_2 \cdot r_2. \qquad (6c)$$

In measurement cross section E

$$M_{EYz} = \sigma_{Ez} \cdot W_E \qquad (7a)$$
$$= F_{D2}\,(b_{d2} - b_5) + F_{K2z} \cdot a_2 + F_{z2}\,(b_{z2} - b_5). \qquad (7b)$$

In measurement cross section F

$$M_{Fyz} = \sigma_{Fz} \cdot W_F \qquad (8a)$$

$$= F_{D2}(b_{d2} - b_6) + F_{K2z} \cdot a_2 + F_{z2}(b_{z2} - b_6) \qquad (8b)$$

(σ = bending stress, W = resistance moment).

The bending moments for the measurement cross sections A through F in the XY-plane serve to acquire the tangential forces T_{x1} and T_{x2} at the contact points A_1 and A_2. The equations for these bending moments are equivalent to the bending moment equations of the YZ-plane. In setting up the equations for the XY-plane, it must be taken into account that the lateral suspension forces F_Y, which correspond to the guide force, enter into the moments with zero lever arm, and that no damping forces act in the X-direction, i.e. $Y_1 = Y_2 = F_{D1} = F_{D2} = 0$. In addition, the forces Q_1 and Q_2 must be replaced by T_{X1} and T_{X2}, and the index z must be replaced by x.

The forces Y_1, Y_2, Q_1, Q_2, T_{X1} and T_{X2} are exerted on the wheelset. They are obtained, as functions of the measured bending moments in the measurement cross sections A through F, by transforming the equations (3) through (8):

$$Y_1 = -\frac{b_2 - b_{A1}}{(b_1 - b_2)\, r_1} M_{Ayz} + \frac{b_1 - b_{A1}}{(b_1 - b_2)\, r_1} M_{Byz} - \frac{b_{A1} + b_4}{(b_3 + b_4)\, r_1} M_{Cyz} + \frac{b_{A1} - b_3}{(b_3 + b_4)\, r_1} M_{Dyz} \qquad (9)$$

$$Y_2 = \frac{b_5 - b_{A2}}{(b_6 - b_5)\, r_2} M_{Fyz} - \frac{b_6 - b_{A2}}{(b_6 - b_5)\, r_2} M_{Eyz} + \frac{b_{A2} + b_3}{(b_3 + b_4)\, r_2} M_{Dyz} - \frac{b_{A2} - b_4}{(b_3 + b_4)\, r_2} M_{Cyz} \qquad (10)$$

$$Q_1 = \frac{1}{b_1 - b_2} M_{Ayz} - \frac{1}{b_1 - b_2} M_{Byz} -$$

$$- \frac{1}{b_3 + b_4} M_{Cyz} + \frac{1}{b_3 + b_4} M_{Dyz} \tag{11}$$

$$Q_2 = \frac{1}{b_6 - b_5} M_{Fyz} - \frac{1}{b_6 - b_5} M_{Eyz} -$$

$$- \frac{1}{b_3 + b_4} M_{Dyz} + \frac{1}{b_3 + b_4} M_{Cyz} \tag{12}$$

$$T_{x1} = \frac{1}{b_1 - b_2} M_{Axy} - \frac{1}{b_1 - b_2} M_{Bxy} -$$

$$- \frac{1}{b_3 + b_4} M_{Cxy} + \frac{1}{b_3 + b_4} M_{Dxy} \tag{13}$$

$$T_{x2} = \frac{1}{b_6 - b_5} M_{Fxy} - \frac{1}{b_6 - b_5} M_{Exy} -$$

$$- \frac{1}{b_3 + b_4} M_{Dxy} + \frac{1}{b_3 + b_4} M_{Cxy}. \tag{14}$$

Among the geometrical dimensions in Equations (9) through (14), only the quantities b_{A1} and b_{A2} are not constant. Consequently, the Y-forces from Equations (9) and (10) depend on the wheel contact point. The following procedure can be used to determine the wheel contact point:

In Equations (13) and (14), the T_x-forces from the planes A, B, C, D and respectively C, D, E, and F were determined without the quantity corresponding to the wheel contact point. But T_x can already be determined solely from the planes A, B, and C or respectively D, E, and F. The following hold:

$$\begin{aligned}
M_{Axy} &= F_{K1x} \cdot a_1 + F_{x1} (b_{x1} - b_1) \\
M_{Bxy} &= F_{K1x} \cdot a_1 + F_{x1} (b_{x1} - b_2) \\
M_{Cxy} &= F_{K1x} \cdot a_1 + F_{x1} (b_{x1} - b_3) + T_{x1} (b_{A1} - b_3) \\
M_{Dxy} &= F_{K2x} \cdot a_2 + F_{x2} (b_{x2} - b_4) + T_{x2} (b_{A2} - b_4) \\
M_{Exy} &= F_{K2x} \cdot a_2 + F_{x2} (b_{x2} - b_5) \\
M_{Fxy} &= F_{K2x} \cdot a_2 + F_{x2} (b_{x2} - b_6).
\end{aligned}$$

From these equations one obtains

$$T_{x1} = \frac{1}{b_{A1} - b_3} M_{Cxy} - \frac{b_1 - b_3}{(b_1 - b_2)(b_{A1} - b_3)} M_{Bxy} +$$

$$+ \frac{b_2 - b_3}{(b_1 - b_2)(b_{A1} - b_3)} M_{Axy} \qquad (15)$$

$$T_{x2} = \frac{1}{b_{A2} - b_4} M_{Dxy} - \frac{b_6 - b_4}{(b_6 - b_5)(b_{A2} - b_4)} M_{Exy} +$$

$$+ \frac{b_5 - b_4}{(b_6 - b_5)(b_{A2} - b_4)} M_{Fxy}. \qquad (16)$$

The following therefore hold for the wheel contact points b_{A1} and b_{A2}:

$$b_{A1} = b_3 + \frac{1}{T_{x1}} M_{Cxy} - \frac{b_1 - b_3}{T_{x1}(b_1 - b_2)} M_{Bxy} +$$

$$+ \frac{b_2 - b_3}{T_{x1}(b_1 - b_2)} M_{Axy} \qquad (17)$$

$$b_{A2} = b_4 + \frac{1}{T_{x2}} M_{Dxy} - \frac{b_6 - b_4}{T_{x2}(b_6 - b_5)} M_{Exy} +$$

$$+ \frac{b_5 - b_4}{T_{x2}(b_6 - b_5)} M_{Fxy}. \qquad (18)$$

One thus obtains the wheel contact points b_{A1} and b_{A2}, if the forces T_{x1} and T_{x2} are determined according to Equations (13) and (14). These equations are linearly independent of Equations (15) and (16) and are independent of the wheel contact point. In order to solve Equations (17) and (18) electronically, the moments must be divided by the forces T_{x1} and T_{x2}. Since the forces pass through zero, the calculated wheel contact points become undefined, even though the moments simultaneously tend to zero and the quotient tends to a finite limit value.

Various possibilities for electronic solution are at this time still under study. An attempt is being made to determine continuously the wheel contact point according to Equations (17) and (18). By inserting such a solution into Equations (9) and (10), a measurement of the guide force Y becomes possible, which would be error-free with respect to the wheel contact point.

2.2 Influential Electrical Quantities

The wheelset axle method requires an extensive measurement chain to determine the forces between the wheel and the rail. There are several critical points within this measurement chain. These have the dominating responsibility for the electrical accuracy of the measured quantities. These points primarily comprise the strain gauge bridge itself, the transmission of electrical signals, proportional to the strains, from the axle shaft, by means of a data transmitter, the sine-cosine function generator, the amplifier, and the multiplier.

2.2.1 Influential Electrical Quantities

As is apparent from Section 2.1.2, the forces and the wheel contact points are determined by moments from various measurement planes and corresponding lever arms (geometrical dimensions). It will be shown below in what form the electrical representation of the mechanical moments is afflicted with errors.

Figure 2 shows the arrangement of the strain gauge of a cross section and the associated electrical circuit of a Wheatstone

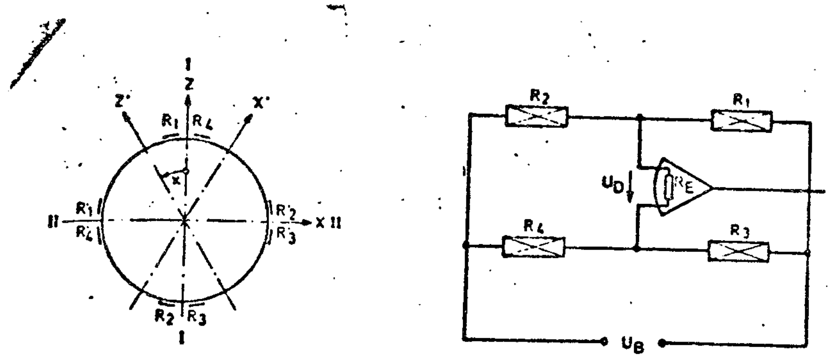

Figure 2: Arrangement of the strain gauges on the wheelset axle and the associated bridge circuit

bridge. The following explanations will clarify why this application of the strain gauge makes sense and makes it possible to calculate the moments. For the diagonal voltage U_D of the bridge (Reference 4), the following holds:

$$U_D = U_B \frac{R_1 \cdot R_4 - R_3 \cdot R_2}{(R_1 + R_2)(R_3 + R_4) + \frac{(R_1 \cdot R_2 (R_3 + R_4) + R_3 \cdot R_4 (R_1 + R_2))}{R_E}}$$

For $R_E > R_1$, R_2, R_3, and R_4, this equation can be simplified

$$(R_1 \approx R_2 \approx R_3 \approx R_4 \approx 600\Omega \text{ und } R_E \approx 10^6\Omega$$

$$U_D = U_B \frac{R_1 \cdot R_4 - R_3 \cdot R_2}{(R_1 + R_2) \cdot (R_3 + R_4)}. \qquad (19)$$

Two conditions are required for further calculation:

- The forces in the YZ-plane cause a resistance change ΔR_z in bridge I ($\Delta R_x = 0$),
- The forces in the XY-plane cause a resistance change ΔR_x in bridge II ($\Delta R_z = 0$).

This results in the following formula for the individual resistances of the bridge with resistances R_0 in the unstressed state

$$
\begin{aligned}
R_{1,4} &= R_{01,4} + \Delta r_z \cdot \cos\kappa + \Delta R_x \cdot \sin\kappa \\
R_{2,3} &= R_{02,3} + \Delta R_z \cdot \cos(\kappa + 180°) + \\
&\quad + \Delta R_x \cdot \sin(\kappa + 180°) \\
R'_{1,4} &= R'_{01,4} + \Delta R_z \cdot \cos(\kappa + 90°) + \\
&\quad + \Delta R_x \cdot \sin(\kappa + 90°) \\
R'_{2,3} &= R'_{02,4} + \Delta R_z \cdot \cos(\kappa + 270°) + \\
&\quad + \Delta R_x \cdot \sin(\kappa + 270°).
\end{aligned}
$$

Because of trigonometric relationships, and after insertion in (19), the following is obtained for the diagonal voltages U_D of bridges I and II:

$$
U_{DI} = U_{BI}\left[\frac{(R_{10} + \Delta R_z \cdot \cos\kappa + \Delta R_x \cdot \sin\kappa)\cdot(R_{40} + \Delta R_z \cdot \cos\kappa + \Delta R_x \cdot \sin\kappa)}{(R_{10} + R_{20})\cdot(R_{30} + R_{40})} - \frac{(R_{30} - \Delta R_z \cdot \cos\kappa - \Delta R_x \cdot \sin\kappa)\cdot(R_{20} - \Delta R_z \cdot \cos\kappa - \Delta R_x \cdot \sin\kappa)}{(R_{10} + R_{20})\cdot(R_{30} + R_{40})}\right]
$$

$$
U_{DII} = U_{BII}\left[\frac{(R'_{10} - \Delta R_z \sin\kappa + \Delta R_x \cos\kappa)\cdot(R'_{40} - \Delta R_z \sin\kappa + \Delta R_x \cos\kappa)}{(R_{10} + R_{20})\cdot(R_{30} + R_{40})} - \frac{(R'_{30} + \Delta R_z \sin\kappa - \Delta R_x \cos\kappa)\cdot(R'_2 + \Delta R_z \sin\kappa - \Delta R_x \cos\kappa)}{(R'_{10} + R'_{20})\cdot(R'_{30} + R'_{40})}\right].
$$

The strain gauges are selected and equalized in such a manner that the individual resistances in the unstressed state are of equal magnitude. Consequently, the resistance R can be used for these individual resistances in further calculations. The following diagonal voltages U_{DI} and U_{DII} are thus obtained:

$$
U_{DI} = U_{BI}\left(\frac{\Delta R_z}{R}\cos\kappa + \frac{\Delta R_x}{R}\sin\kappa\right) \tag{22}
$$

$$
U_{DII} = U_{BII}\left(-\frac{\Delta R_z}{R}\sin\kappa + \frac{\Delta R_x}{R}\cos\kappa\right). \tag{23}
$$

The following relationship (Reference 5) exists between a moment, which effects bending of a body, and the resistance change of a strain gauge, which is glued to the body under bending stress:

$$\frac{\Delta R}{R} = K \cdot \epsilon = K \cdot \frac{\sigma}{E} = K \cdot \frac{M}{W \cdot E}. \qquad (24)$$

Here the symbols have the following significance:

K = Strain gauge constant

ε = Strain

E = Elastic modulus of the body

W = Resistance moment

σ = Bending stress

M = Bending moment.

After suitable transformation of Equations (22) and (23), and under the condition of equal bridge feed voltages $U_{BI} = U_{Bii} = U_B$, the individual bending moments are obtained as follows:

$$M_{yz} = \frac{W \cdot E}{K \cdot U_B} (U_{DI} \cdot \cos\kappa - U_{DII} \cdot \sin\kappa) \qquad (25)$$

and

$$M_{xy} = \frac{W \cdot E}{K \cdot U_B} (U_{DI} \cdot \sin\kappa + U_{DII} \cdot \cos\kappa). \qquad (26)$$

We assume that the bridge feed voltage U_B, the factor $W \cdot E/K$ and the formulas (U_{DI} sin k + U_{DII} cos k) and respectively (U_{DI} Cos k - U_{DII} sin k) are subject to error and that they enter into the moments together with their errors. Under this assumption, the error calculation yields the following maximum relative errors for the moments M_{yz} and M_{xy}:

$$\frac{|\Delta M_{yz}|}{|M_{yz}|} = \frac{|\Delta U_B|}{|U_B|} + \frac{|\Delta\left(\frac{E \cdot W}{K}\right)|}{\left|\frac{E \cdot W}{K}\right|} + \frac{|\Delta\, (U_{DI} \cdot \cos\kappa - U_{DII} \cdot \sin\kappa)|}{|U_{DI} \cdot \cos\kappa - U_{DII} \cdot \sin\kappa|} \tag{27}$$

with $U_{DI} \cdot \cos k - U_{DII} \cdot \sin k \neq 0$

$$\frac{|\Delta M_{xy}|}{|M_{xy}|} = \frac{|\Delta U_B|}{|U_B|} + \frac{|\Delta\left(\frac{E \cdot W}{K}\right)|}{\left|\frac{E \cdot W}{K}\right|} + \frac{|\Delta(U_{DI} \cdot \sin\kappa + U_{DII} \cdot \cos\kappa)|}{|U_{DI} \cdot \sin\kappa + U_{DII} \cdot \cos\kappa)} \tag{28}$$

with $U_{DI} \cdot \sin k + U_{DII} \cdot \cos k \neq 0$.

The first two expressions in the above equations are negligible, since the relative error of the feed voltage can easily be kept below 1 ‰, and since the deviation of the characteristic mechanical data E, W, and K are determined for each measurement plane by calibration, and consequently do not contribute to the error. The remaining relative errors from the products of the diagonal voltage and the cosine or respectively sine of the wheel rotation angle are caused on the one hand by the conformity error of the sine-cosine transducer and, on the other hand, by the error of the analog multiplier.

The two residual errors are mutually dependent. Consequently, it is difficult to perform a detailed theoretical error estimate.

The diagonal bridge voltages are of the order of 0 mV to ± 5 mV. These small voltages are currently being transmitted through rotary transmitters and cables, some of them more than 100 meters long. They are transmitted to the measurement car for further processing - such as amplification, multiplication, addition, etc. This transmission superposes on the measurement signal thermal voltages and other noise voltages from stray inductance. Alternating voltages are suppressed only by a high common mode rejection of about 60 dB in the input amplifier. Depending on the magnitude of the noise signals in relation to the useful signal, this can possibly lead to a garbling of the measurement signal.

2.2.2 Compensation of the Influential Electrical Quantities

In analog computer technology, the multiplier is known to be a component with relatively great error. Its accuracy depends on the amplitudes of the input signals. For this reason, it is necessary to perform all the multiplications in the high voltage range. Present commercially available analog multipliers have a percentage error of at least 0.15% relative to $\pm$ 10 V. Depending on the measured quantities, the multipliers are used in individual quadrants. Furthermore, in particular equations, multiplications in the same quadrant are sometimes subtracted from one another. Consequently, it makes sense to select the multipliers according to their inaccuracy with respect to particular quadrants and to deploy them correspondingly pairwise. A test procedure was set up to test the multipliers, according to Figure 3. This procedure uses a digital computer. By means of this procedure, it is possible to reduce the error by a factor of 10 during multiplication.

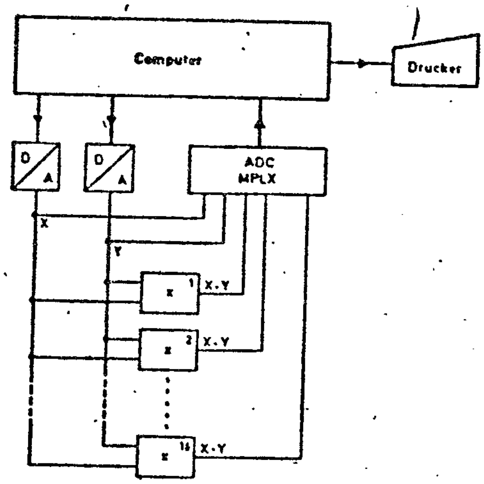

Figure 3: Circuit diagram for automatic determination of characteristic error curves of multipliers
1. Printer

The possibility exists of dispensing with analog multiplication, by using a sine or cosine-shaped feed voltage instead of the constant bridge feed voltages U_{BI} and U_{BII}, i.e. with

$$U_{BI} = U_0 \cdot \cos\kappa \qquad (29a)$$

(U_0 = peak value of the alternating voltage)

and

$$U_{BII} = -U_0 \cdot \sin\kappa \qquad (29b)$$

$$M_{yz} = \frac{W \cdot E}{K \cdot U_0}(U_{DI} + U_{DII}) \qquad (30)$$

and correspondingly with

$$U_{BI} = U_0 \cdot \sin\kappa \qquad (31a)$$

and

$$U_{BII} = U_0 \cdot \cos\kappa \qquad (31b)$$

one obtains

$$M_{xy} = \frac{W \cdot E}{K \cdot U_0}(U_{DI} + U_{DII}). \qquad (32)$$

But the conformity error of the sine-cosine transducer now appears even more strongly.

This possibility of measuring the moments has another advantage in addition to reduced measurement error: By giving up the multiplier, the costs for the measurement chain can be substantially reduced. Its disadvantage lies in the fact that only the Q- and Y-forces or the T_x-forces can be determined with two bridges per measurement plane. The determination of all the forces can be achieved only by gluing on two additional bridges, displaced from one another by 90°.

The preceding explanations make clear that the conformity error of the sine-cosine transducer plays an essential role in the wheelset axle method. Consequently, it has always been attempted to keep this error as small as possible. Figure 4 shows the error curves for different transducers, where the transducer was calibrated to zero-volt output voltage for zero degrees. It can be clearly seen that currently used resistance potentiometers have a percentage error of maximum 1.5%.

A possibility for reducing the effect of noise voltages, arising from data transmission, consists in amplifying the measurement signal already on the axle and only then conducting it over the transmission chain. Figure 5 shows the built-in amplifier in combination with the transmitter for the measurement current and the sine-cosine transducer. A difference amplifier with a common mode rejection of 120 dB and a 200-fold amplification is used for each

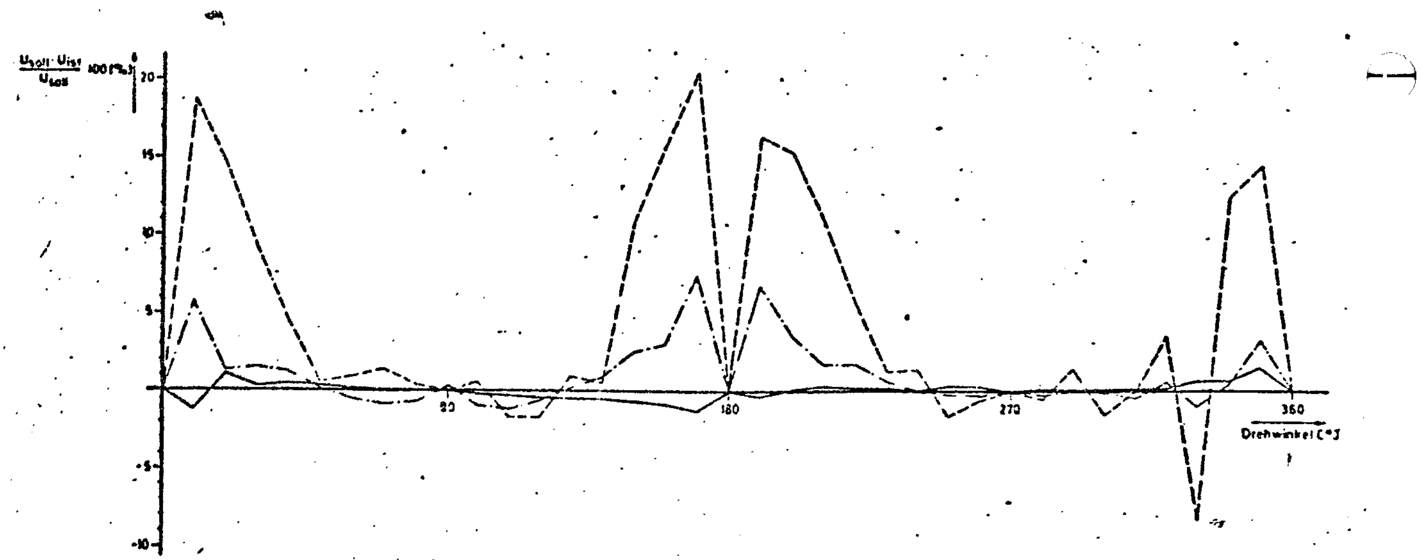

Figure 4: Error curves for various transducers

——— Resistance Potentiometer

------- Inductive transducer (without correction)

-·-·-·- Inductive transducer (with correction)

1. Rotation angle (°)

bridge. Noise signals in the μV range no longer affect measurement signals amplified in this fashion. An FET switch was provided, to be able to detune the bridges in a defined manner for checking purposes. By means of this FET switch, a resistor can be switched in parallel with the strain gauge. Figure 6 shows a board equipped for six channels. Another advantage of these amplifiers on the axle is that the amplified measured quantities can be transmitted unsymmetrically with respect to ground. This leads to reduction of the transmission paths and consequently to a shortening of the mechanical construction at the axle journal collar.

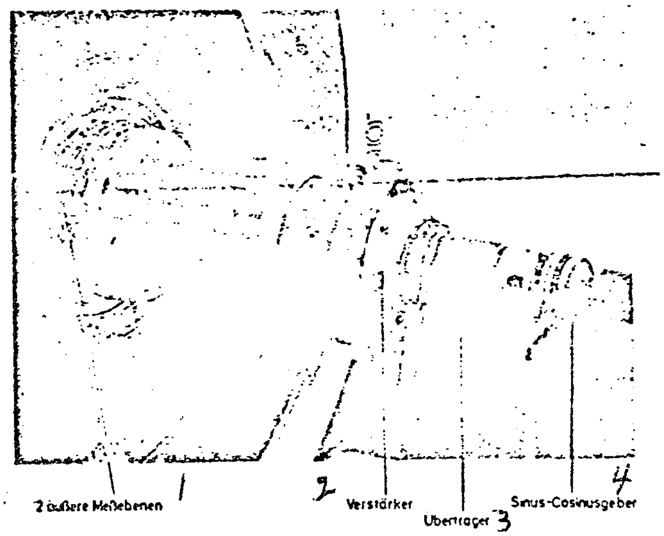

Figure 5: Axle-journal collar with 2 external measurement planes, flanged-on amplifier, transmitter, and sine-cosine transducer

1. External measurement planes
2. Amplifier
3. Transmitter
4. sine-cosine transducer

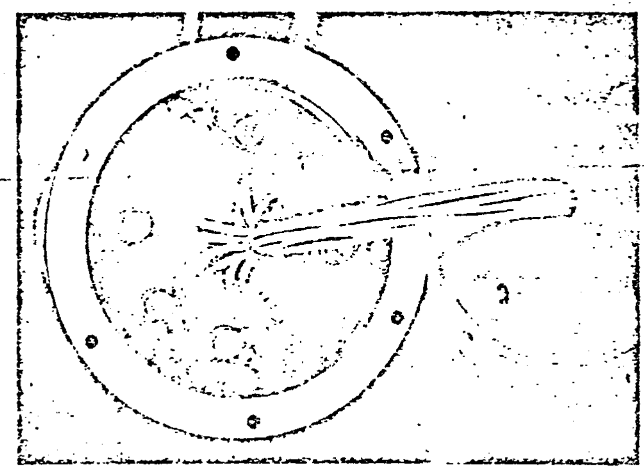

Figure 6: Plate and housing of the measurement amplifier are fixed to the wheelset axle

The measurement amplifier on the axle shaft additionally guarantees a more precise adjustment of the bridge excursion and simplified zero-point control of the entire measurement chain. At the beginning of each measurement day, each of the 12 measurement amplifiers is first adjusted for 0 Volts. With the above-mentioned FET switch, a detuning resistor is switched in parallel to a branch of the bridge. This is now no longer done over a long cable path, but directly at the glued-on bridge. Consequently, the excursion of the individual bridge is controlled at the output of the measurement amplifier.

As described in Section 2.2.1, the resistance value of the strain gauge is measured in the unglued state, and four strain gauges with approximately the same resistances are connected together into a complete bridge. Gluing the strain gauges causes a change of the resistance R_0. This is included in the calibration process during the unloaded state of the axle shaft. For each channel, this is recorded as an electric voltage, proportional to the applied bridge feed voltage. These voltages must be adjusted immediately after adjusting the bridge excursion, by means of a null potentiometer. This is done for each of the 12 measurement amplifiers. Subsequently, the bridge feed voltage is short-circuited and is placed at zero potential, by means of a change-over unit in the measurement amplifier. In this way, the zero position of the entire measurement chain, from the Wheatstone bridge through the measurement amplifiers, multipliers, adders, up to the output of the measured data converters for the forces Y, Q, and T_x, can be adjusted in such a manner that no additional zero-point error any longer occurs, except for the above-mentioned error of the analog multiplier. This check is possible at any time and no longer depends on the track position and speed of the vehicle being measured.

2.3 Summary of the Total Error which is Caused by the Influential Quantities

The discussion of the effects of influential mechanical and electrical quantities on the measurement accuracy of the wheelset axle method has shown that essentially only three errors still affect the measured result: Among the mechanical quantities, these errors include the non-constant wheel contact points. On the other hand, among the electrical quantities, these errors include the error from the analog multiplier and the sine-cosine transducer. According to the explanations in Section 2.2, the entire electrical error can be specified to maximum 2% of the respectively measured force. For the Y-forces, the effect of the wheel contact point must also be added to this error. According to Equations (3) and (4), in the case of heavy wheel loads, i.e. large Q and curves, i.e. large b_{A1}, this error has a quite significant effect. When the wheel contact points change to a greater extent, the currently used remedy is that the wheel contact point on curves is determined by labeling the rail with chalk. The Y-force is then corrected according to Equations (3) and (4).

3. Conclusions for Using the Wheelset Axle Method

The various considerations and insights have as their primary objective to facilitate use of the wheelset axle method with routine studies on railway vehicles, to exclude error sources, and to reduce labor. This has effects both on the calibration process and on the measurement chain.

3.1 Effects on the Calibration of Wheelsets

Up to now, calibration of a measurement wheelset according to reference (1) point 4.1 and 4.2 was very laborious and timeconsuming. In addition, recent experiments on the calibrating stand have shown that a wheelset, installed underneath a vehicle, is exposed to other unknown influential quantities - such as the bearing box tilt and undefined contact point of the spring loads. The resulting β-values ($\beta = \sigma_{theoretical}/\sigma_{actual}$) are consequently falsified by these unknown influential quantities. The calibration can be improved by precalibrating the wheelset in its uninstalled state. After it is installed in the vehicle, only follow-up calibration is thus necessary. This also makes it possible to shorten the working times of the experimental vehicles for the precise determination of β-values and of the self-weight of the wheelset.

Precalibration will be described below, by means of a six-plane measurement wheelset:

As Figure 7 shows, the glued measurement wheelset rests on two dollies. It can therefore be turned into any arbitrary angular position. The wheelset is now lifted in the desired angular position 0^{o}, 90^{o}, 180^{o}, 270^{o} at the contact points A_1 and A_4. This is done with hydraulic lifters with gauge pins on top. The forces F_{1k} and F_{3k} are caused by the self-weight G of the wheelset. These forces are thus introduced into the wheelset, through in-house fabricated semi-rings with point-like supports, at the contact points A_1 and A_4. The measured voltages σ_{AZ} and σ_{BZ} must now be proportional to the force F_{1k}; the voltages σ_{EZ} and σ_{FZ} must be proportional to the force F_{3k}.

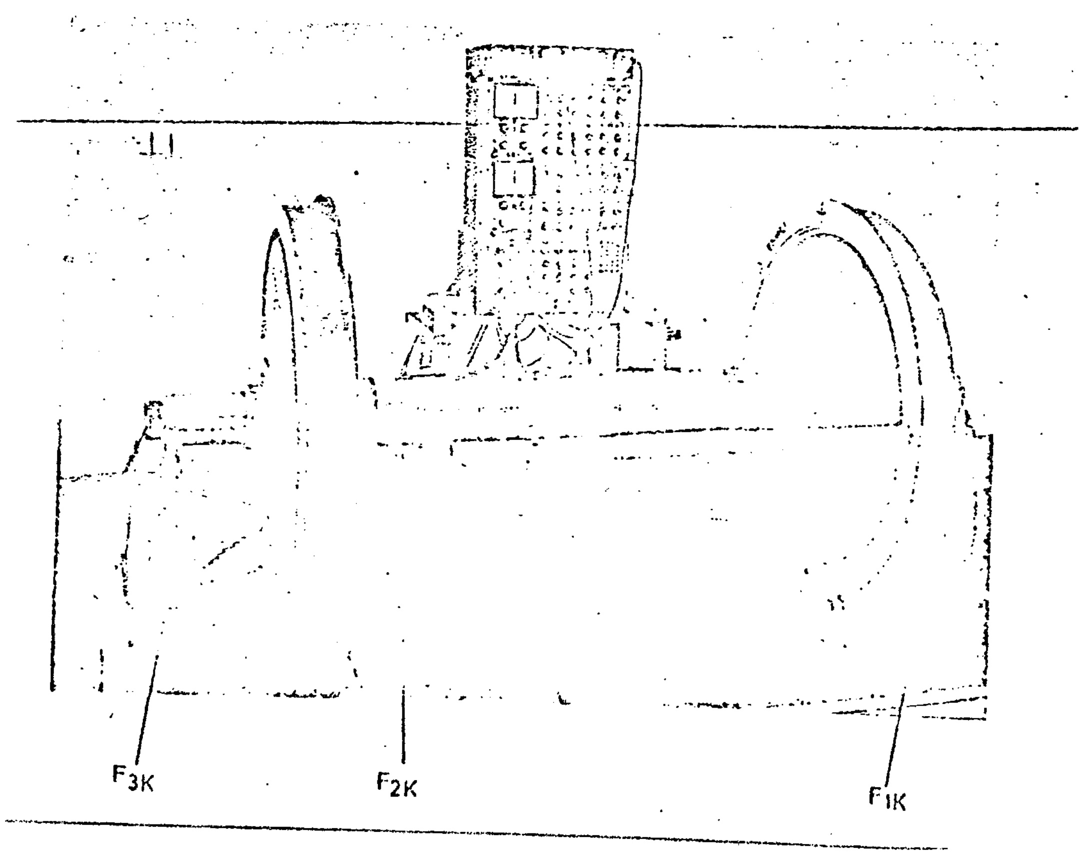

Figure 7: Wheelset with associated measurement setup, mounted on rollers, for acquiring β-values

Punched measurement points are built in on the inside of each wheel disc, at a distance a_3. There are always 4 of these, displaced by 90^0.

The wheelset stays in its lifted position in order to determine the β-values of planes C and D. The forces F_{2k} in the XY-plane are now introduced into the wheel discs (Figure 8). This is done by means of a hydraulic lift, through a spindle with gauge pins between the punching points A_2 and A_3. The forces F_{1k}, F_{2k}, and F_{3k}, which are introduced into the wheelset, must now be in equilibrium with the forces from the bending stresses of the

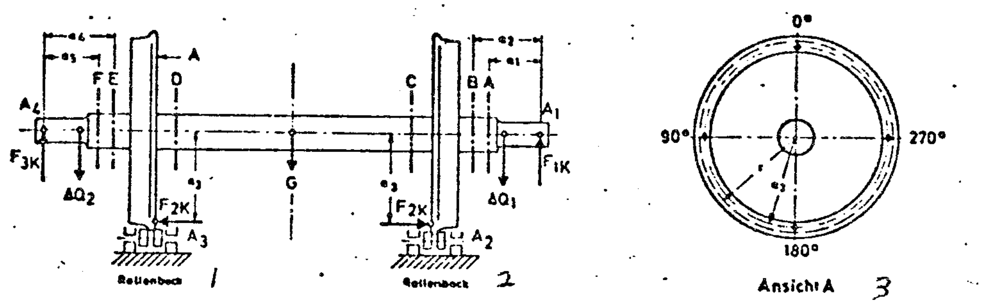

Figure 8: Schematic representation of forces acting on the wheelset during acquisition of -values

1. Dolly; 2. Dolly; 3. View A

measurement cross section. The latter forces are determined by means of the measurement chain. The equilibrium relationship is as follows:

For the measurement cross section A, the following holds:

$$F_{1k} = \frac{M_{Ayz}}{a_1} = \frac{W_A}{a_1} \sigma_{Az}. \tag{33}$$

For the measurement cross section B, the following holds:

$$F_{1k} = \frac{M_{Byz}}{a_2} = \frac{W_B}{a_2} \sigma_{Bz}. \tag{34}$$

For the measurement cross section C, the following holds:

$$F_{2k} = \frac{M_{Cxy}}{a_3} = \frac{W_C}{a_3} \sigma_{Cx}. \tag{35}$$

For the measurement cross section D, the following holds:

$$F_{2k} = \frac{M_{Dxy}}{a_3} = \frac{W_D}{a_3} \sigma_{Dx}. \tag{36}$$

For the measurement cross section E, the following holds:

$$F_{3k} = \frac{M_{Eyz}}{a_4} = \frac{W_E}{a_4} \cdot \sigma_{Ez}. \qquad (37)$$

For the measurement cross section F, the following holds:

$$F_{3k} = \frac{M_{Fyz}}{a_5} = \frac{W_F}{a_5} \sigma_{Fz}. \qquad (38)$$

If the wheelset is now rotated by 90°, the β-values of planes A_x, B_x, C_z, D_z, E_x, and F_x can be determined with the same calibration run.

If the wheelset is pressed upwards at contact points A_1 and A_4, the sum of the forces $F_{1k} + F_{3k}$ corresponds to the force of the self-weight of the wheelset, which is applied at the center of the wheelset. The Q-forces, formed from Equations (9) and (10), are too small by the amounts ΔQ_1 and ΔQ_2. The reason for this is that, as distributed loads, from the respective end of the axle shaft to the measurement planes A, B, and F, E, they cause a moment directed opposite to the forces F_{1k} and F_{3k}.

The remaining static components of the self-weight for the forces Q_1 and Q_2 now are as follows:

$$Q_{1\mathrm{Eigen}} = F_{k1} - \Delta Q_1 \qquad (39)$$
$$Q_{2\mathrm{Eigen}} = F_{k2} - \Delta Q_2 \qquad (40)$$

This precalibrated measurement wheelset can now be installed into the experimental vehicle.

A continuing need for a calibration stand also exists in order to check the β-values and for follow-up calibration in the installed state. The calibration stand described in reference (1) has the disadvantage that, first of all, no T_x-forces can be calibrated or checked and, secondly, that the force necessary to overcome friction in the transverse direction is about 2% of the wheel load. For these reasons, a new calibration stand was designed and constructed. This stand makes possible calibration of all the forces between the wheel and the rail. By means of roller elements, it reduces the displacement force by a factor of 10. Figure 9 shows the calibration stand with a cross roller slide for the Y- and $T_x$forces and the associated force transducer with gauge pins. Because the roller slide can be displaced in the lengthwise direction, braking and accelerating forces can be simulated in a defined manner, and their effects on the forces between the rail and wheel and on the suspension forces can be investigated.

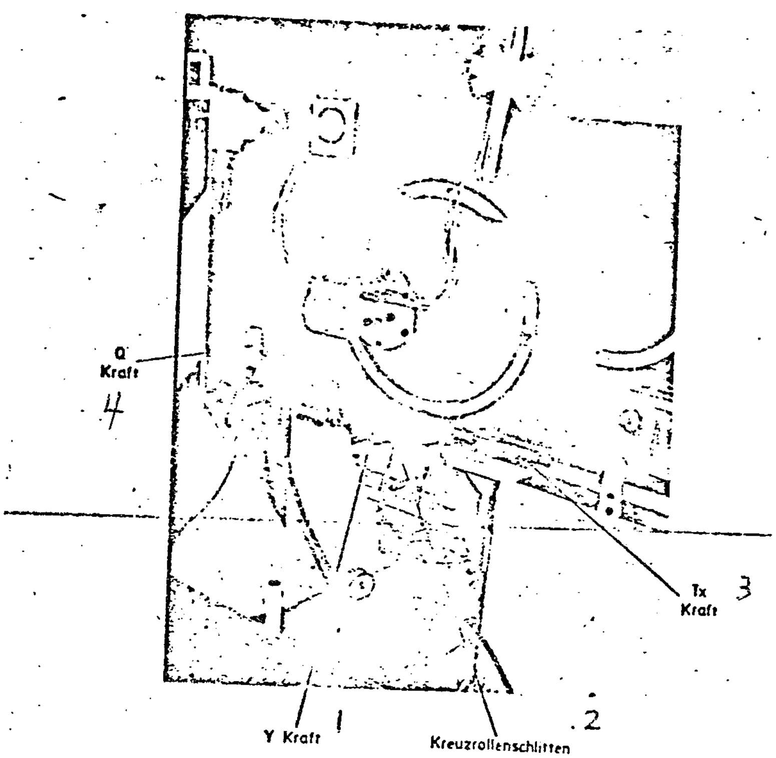

Figure 9: Calibration stand with cross roller slides and force transducers for Q-, Y-, and T_x-forces

1. Y force
2. Cross roller slide
3. T_x force
4. Q force

3.2 Effects on the Measurement Chain

In order to avoid uncertainties and noise effects in measuring the forces between the wheel and the rail, it is necessary to use the six-plane measurement wheelset. As a result, the previously used measurement chain (measurement wheelset computer) had to be designed anew. Furthermore, these additional planes, as already explained above, eliminate the necessity of correcting the measurement values and consequently obviate a correction unit.

The new measurement wheelset computer can be used universally in combination with measurement wheelsets which have 4 or 6 measurement cross sections.

A space-saving modular system makes it possible to house the entire measurement wheelset computer in a double module, despite expanded electronics for 6 measurement cross sections (Figure 10). All electronic components are located on plug-in subassemblies. A built-in test device can quickly determine defects which may possibly occur in the subassemblies. Consequently, defective plug-in units can be exchanged for replacement units. Essentially the following functions are separtely housed in plug-in subassemblies:

- A test unit with an analog display instrument, and a switch for measurement range, testing, and selecting,
- Electronics for adjustable and stabilized feed voltage of the sine and cosine angle transducer,
- A sine and cosine amplifier part with zero equalization and factor potentiometer,

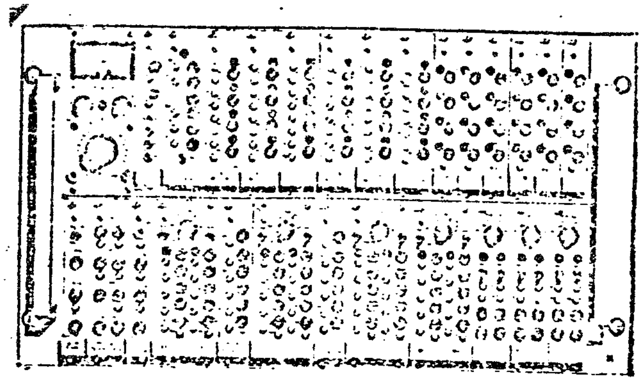

Figure 10: 6-plane measurement wheelset computer in a modular system

- A sine and cosine amplifier part with zero equalization and factor potentiometer,*
- 6 plug-in units with analog multipliers, which multiply the 12 bending stress measurement values from the measurement axle by the sine or respectively cosine of the wheel rotation angle,
- 6 more plug-in units use the product of the measured bending stresses and the angle functions to form the bending stresses σ_x and respectively σ_z, relative to the coordinate system x,z,
- 3 plug-in units determine the forces Y, Q, T_x according to Equations (9) through (14), whereby the lever arms can be adjusted with 4 coefficient potentiometers. Another potentiometer serves to set the zero point of the operational amplifiers. By means of the "self-weight" potentiometer, a constant voltage is set for the static load of the self-weight of the wheelset, as determined during calibration.

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*The repeat is in the original - translator

Besides other modules, which simplify operation of the computer, such as various test voltages, filters, etc., several more modules are present in order to form various operations on the forces, which are important from an arithmetic processing perspective.

4. Prospects, Further Possibilities of Application

The system of equations in Section 2.1 shows that the forces are uniquely determined by the position of the measurement cross sections and by the geometry of the axle. Consequently, the possibility exists of choosing the number and position of measurement planes and of thereby affecting the theoretical relationships between the forces applied to the wheelset and the measured moments.

Besides the forces listed in Section 2.1, other accelerating or braking forces can be applied to the wheelset. In this case, these additional forces can be eliminated by the suitable application of measurement planes. In this way, the forces Q, Y, and T_x, which appear at the contact points, can be measured without falsification.

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APPENDIX 1.3

September 1978

No longer valid!

Draft (not approved)

INTERNATIONAL RAILWAY ASSOCIATION

UIC-KODEX

This memorandum is to be inserted	0 0 0 - 0
into the volume:	V E
V - Vehicles	First Edition 0.0.00

Passenger Car Running Gear

Binding directives are labelled with an asterisk:*

REVISIONS

Memorandum Number 000-0, 00 Edition of 0-0-0000			
Change Notice		Change Notice	
No.	of	No.	of

Remark

This memorandum is part of a problem area to which belong the following memoranda, among others:

- Memorandum 505-2 - VI: Kinematic boundary lines for passenger and baggage cars used in international traffic
- Memorandum 510-2 - VE: Cars - conditions for using wheels with different diameters in running gears of different construction type
- Memorandum 541-1 - VE: Brakes - directives for constructing various brake components
- Memorandum 567-1 - VE: Standard passenger cars of construction types X and Y, approved for international traffic - Characteristics
- Memorandum 567-2 - VE: Standard passenger cars of construction type Z, approved for international traffic - Characteristics

Table of Contents

5. Specifications for simplifying maintenance

5.1 Parts subject to wear

5.2 Wheel set couplings

5.3 Suspensions

5.4 Fastening the hydraulic dampers

5.5 Brakes

5.6 Truck coupling at the car body

5.7 Adjusting the height of the buffers and height adjustment for wear

1. <u>Classification of Trucks</u>

1.1* Trucks with a standard design, for an operating speed of 160 km/h (economical basic construction form)

- with disc brake (single axle brake), optionally with additional shoe brake, or shoe brake alone
- with/without hand brake
- with/without magnetic rail brake
- with special equipment in appropriate circumstances (anti-skid device according to UIC Memorandum 541-1, Chapter IX and X, current return, speed indicator, sanding equipment, cow catcher, load weighing equipment, brake control indicator)
- without supplementary equipment for higher speeds

1.2* Trucks for an operating speed of 160 to 200 km/h

Design according to Number 1.1, but with the following conditions:

- Equipment for yaw damping
- In appropriate circumstances, more powerful brakes

1.3* Trucks for an operating speed above 200 km/h

(still open)

2. <u>Technical Characteristics</u>

2.1 <u>Running Speed</u>

2.1.1* Operating speed = running speed on a standard maintained track

For data, see Number 1.

2.1.2 Maximum speed (experimental speed)

Maximum speed = operating speed plus 10 percent

2.2 Running Characteristics

The truck must guarantee stable running of the vehicle.

The subsequent test of riding quality and running safety is to be performed according to the experimental conditions and measurement methods described in Appendix 1.

2.2.1* Riding Quality - Travel Comfort

Reserving introduction of an ISO-standard recognized by the UIC, provisional limit values are specified for the riding quality perpendicular and horizontal transverse, according to the Wz-method with Wz $\leq$ 3.25, and according to a method still to be specified (similar to ISO 2631), with approximately the following values:

longitudinal:	$\geq$ 20 h
transverse:	$\geq$ 10 h
vertical:	$\geq$ 10 h

2.2.1.1 It is recommended that the value of Wz = 2.5 be maintained for riding quality.

2.2.2 Running Safety

2.2.2.1* Experiments must demonstrate that the limit values for acceleration and H^+-forces are not exceeded.

Limit values for accelerations horizontal-transverse depend on the construction type of the trucks.

$^+$H - horizontal forces between truck and wheel set

The essential evaluation criterion here is the condition that a sudden increase of acceleration remains intact during a longer time.

The wheel set roller bearing transverse forces ($\equiv$H) may not exceed the following limit values:

$$H_{max\ (2\ m)} \leqq 0.85\ (10 + 2Q/3) \quad (kN)$$
$$H_a + s \leqq 0.5\ (10 + 2Q/e) \quad (kN)$$

Appendix 1 contains the experimental conditions for this.

<u>Definitions</u>:

$H_{max\ (2\ m)}$ = maximum transverse force, measured at the level of the bearing box, which acts on a running distance $\geqq$ 2 m, in kN

H_a = arithmetic average of the transverse forces, measured at the level of the bearing box, in kN

s = standard deviation of the H-forces in kN

Q_o = average static wheel force in kN

2.2.2.2* If unique limit values, specific to the vehicle or to the truck, for accelerations and wheel set roller bearing transverse forces, cannot be associated with the measured values according to Appendix 1, and if the running behavior approaches critical values (unstable run[1]), the following further conditions must also be fulfilled:

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[1]A wheel set runs unstable if the running of its shaft is no longer characterized by friction lock but by positive lock (striking of the flange against the rail head).

The maximum transverse forces exerted on the track may not exceed the following limit values:

$\Sigma Y_{max\ (a\ m)}$	$\leq 0.85\ (10 + 2Q/3)$	(kN)	at track defect (geometry or track modules) points and track instability
$\Sigma Y_a + s$	$\leq 0.5\ (10 + 2Q/3)$	(kN)	average running behavior

<u>Definitions</u>:

$\Sigma Y_{max(2\ m)}$ = largest sum of the transverse forces Y, in kN, which acts from the wheel set on the rail along a running length $\geq$ 2 m

ΣY_a = arithmetic average of the sum of the transverse forces, in kN, which act from the wheel set on the rail

s = standard deviation of ΣY in kN

Q = average static wheel load in kN

2.2.3* Derailing safety in warping rails

Experiments must demonstrate that derailing safety is guaranteed on warped rails.

The ratio of transverse forces to vertical forces at the guiding wheel may not exceed 1.2[1].

Derailing safety on warped rails must be guaranteed

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[1]Definitive regulation by ORE/UIC being reserved.

- with a rail warp, on the base of the truck wheelbase, of 7 0/00
  and simultaneously
  with a rail warp, on the base of the truck-center distance
  of 4.1 0/00
  (distance between pins 19.0 m)

- also with a rail warp, on the base of the truck wheelbase,
  of 10 0/00
  and simultaneously
  with a rail warp, on the base of the truck-center distance
  of 3.1 0/00
  (distance between pins 19.0 m).

2.3 Noise Reduction

2.3.1* By means of constructive measures, the excitation and transmission of body sounds must be kept at a minimum in the areas of

- wheel set linkage
- secondary suspension stage
- truck linkage at the car body and
- brake

2.3.2* Sound radiation is to be reduced by suitable measures, for example by installing diaphragms; operational monitoring and maintenance may not be impaired thereby.

Measurements and evaluations must be performed on unbraked vehicles.

Appendix 3 contains the experimental conditions.

2.4 <u>Maximum Clearance Envelope, Boundaries, Slant Coefficient, Turn-Out Angle</u>

2.4.1 Maximum Clearance Envelope

(Still to be specified by the Studies Group.)

2.4.2* Boundary profile according to UIC Memorandum 505-2

2.4.3* Construction height from RS+ to lower edge of the body cross-bar - 1000 mm, with the possibility of allowing the secondary suspension to penetrate into the car body

2.4.4* Slant coefficient s $\leq$ 0.40, if possible $\leq$ 0.20

2.4.5* Smallest runnable track radius with 19 m pin distance - 80 m

2.5* <u>Total Mass</u>

Total mass of a truck with equipment for speeds up to 200 km/h corresponding to the Number 1.2, $\leq$ 7 t

2.6 <u>Linkage Car Body-Truck</u>

2.6.1* It should be possible to adapt the linkage of the truck to various types of passenger cars within one construction group (according to Appendix 2, Number 1). The cars have appropriate contact surfaces for this purpose.

The car body must be supported on the truck at a height range of 1000 mm above RS.

The car body-truck linkage must not be prone to vibration.

2.6.2* Lift-off protection must be present.

+RS = Rail Surface

2.6.3* For the truck-car body linkage, the following reference loads shall be applicable:

Vertical: $P_v = 2 \cdot$ maximum axle load $- G_{truck}$

$$\boxed{P_{v\ stat.} = 1.3 \cdot P_v}$$

Transverse: $\boxed{P_{Q\ qstat.} = 0.3 \cdot P_v}$

Longitudinal: Longitudinal forces arise from braking and impact shocks. Impact shocks are here decisive.

$P_{L\ qstat.} = 0.16 \cdot P_v$ (as a result of braking)

$\boxed{P_{L\ qstat.} = 3 \cdot G_{truck}}$ (as a result of impact shocks)

Definitions:

P_v = vertical rated load on the truck at a maximum axle load (for one truck)

G_{truck} = self-weight of a truck

$P_{v\ stat.}$ = vertical static force, to be used as the basis of calculation

$P_{Q\ qstat.}$ = quasi-static transverse force, to be used as the basis of calculation

$P_{L\ qstat.}$ = quasi-static longitudinal force, to be used as the basis of calculation

2.7 Limit Values for Fabrication, Refurbishing, Operation and Operating Time and Running Performance

2.7.1* Limit values for the fabrication, refurbishing and operation of wheels: see UIC Memorandum 510-2.

2.7.2* Depending on its permitted speed and track lines, the truck should have an operating time of 1.5 - 4 years between overhauls, or a running performance of 350,000 - 600,000 km, without the occurrence of significant defects which would require putting the car out of operation.

2.8.3 It is desirable that the running range of the truck without profile correction should be

-350,000 km on lines with few curves

-200,000 km on lines with many curves

3. <u>Special Characteristics for Individual Construction Groups</u>

3.1 <u>Wheel Sets and Wheel Set Bearings</u>

3.1.1* Axle loads	7-16 t
3.1.2* Distance of wheel sets	2.5 - 2.75 m
3.1.3* Wheels	
3.1.3.1* Wheel diameter	The running circle of wheels must belong to diameter state 920/840 mm
3.1.3.2 Type of wheels	It is recommended to use Monobloc-wheels
3.1.4* Wheel profile	UIC/ORE standard wheel profile according to Appendix 2 of UIC Memorandum 510-2.

3.1.5 Wheel set bearing

3.1.5.1* Two-system wheel set roller bearings according to Number 4.2 of this memorandum and according to the recommendations in RP 8 of the ORE Committee B 136.

3.1.5.2* Because hot boxes are taken care of by wayside installations, only external wheel set bearings may be used, where a scanning space must be kept free below the bearing housings.

3.1.5.3 In order to avoid electric current damage in the wheel set roller bearings, it is recommended to use a return current contact in combination with shunts. The roller bearings must here be electrically insulated.

3.1.5.4 It is recommended that the construction of the wheel set bearings be designed so that profile correction on underfloor lathes is not hindered.

3.1.6* Wheel set bearing housing

Material: Steel or Sphero-cast metal

Light metal, inasmuch as adequate guidance of the wheel set is guaranteed in case it runs hot.

3.1.7* Wheel set bearing center distance 2000 mm

3.1.8 Wheel set linkage

The spring stiffnesses c_x and c_y must be chosen so that

- on the one hand a sufficiently large interval is guaranteed between the operating speed and the beginning of unstable running of the wheel set,
- and on the other hand, adequate detuning is guaranteed from proper frequencies by the running of the wheel set and proper bending frequencies horizontally transverse to the body.

3.1.9* Lift-off protection

Locking devices must be present, or devices which prevent the wheel set bearing housing from losing its coupling to the truck frame, or which, in case of derailing, make it possible to suppress deflection of the spring.

3.2 Damper Suspensions

3.2.1* The mass of unsuspended parts must be as small as possible, taking into account the dynamics of the truck and car body.

3.2.2* Suspension displacement empty-loaded on a tangent track, $\leq$ 85 mm

3.2.2.1 In some circumstances, equalization must be provided for wear of the suspensions, for wheel rim wear, and for the support height tolerances of springs.

3.2.3 The suspension (primary suspension and secondary suspension) should preferably be implemented by helical springs. Alternative solutions, air or rubber springs.

3.2.3.1 It is recommended that the primary and secondary suspension stages be designed so that simple adaptation of the suspension is possible when the individual corners of the car have different weights.

This holds for all car weights within the framework of the axle loads cited in Number 3.1.1.

3.2.3.2 In the case of helical spring supports, it is recommended that the centering of the car body be guaranteed by an oriented installation of the suspensions.

3.2.3.3 It is recommended that hydraulic vibration dampers be provided for the primary and secondary suspensions. The action of these vibration dampers is proportional to the speed.

3.2.3.4 If damping is provided for the secondary suspension, separate horizontal and vertical dampers should be supplied.

3.2.4* The division between the vertical primary and secondary suspension must be made according to technical criteria regarding running and derailing perspectives, and taking into account the respective vertical proper vibrational frequencies of the car body.

3.2.5* In case the suspension breaks, intercept devices or emergency supports must guarantee operating safety.

3.2.6* Total spring displacement horizontal transverse (q + w) at least $\pm$ 60 mm.

3.2.6.1 A car body-transverse play control, which depends on the curvature of the track, should to the greatest possible extent be independent of the longitudinal play of the truck.

3.2.6.2 The transverse reset force of the car body support should as much as possible be proportional to the load.

3.3 Additional Specifications for Trucks with Air Suspension

3.3.1 Roll of the car body should lead to the smallest possible consumption of air.

3.3.2* In case the air suspension fails in normal operational use, the following must be guaranteed:

- running safety at maximum speed and
- a reasonable riding quality up to 160 km/h.

In case of automatic load braking, a braking weight at brake position P must be reached at the level of the self-weight of the car, and at brake position R at the level of 1.5 the self-weight of the car.

3.3.3* The evaluation of operating capability must be indicated and the operating range must be identified (notches, pointers, identifying color fields). The state of the air suspension in operational use must be easily recognized by the operating personnel.

3.3.4* Leakage losses in the air suspension system must be so small that the loss at the height of the bumpers is less than 5 mm, when the system is locked for two hours.

3.3.5.1 Air is supplied through the main container line or respectively through a compressed air generating system on the vehicle itself.

3.3.5.2* The compressed air may not be withdrawn from the compressed air brake equipment.

3.3.5.3 It is recommended that containers for additional air be installed beneath the car body.

3.3.6 If possible, the air suspension should have its own damping for vertical oscillations, so that additional damping elements will be obviated.

3.4 Truck Frames

3.4.1* Welded and cast steel designs are permitted for the truck frames. The types of steel utilized must guarantee a minimum tensile strength of 370 N/mm^2, and must be easily weldable.

From a technical welding perspective, the truck frame should be constructed in such a fashion that subsequent strain annealing is not necessary.

3.4.2* Truck frames, for which ten years' operating experience does not yet exist, must be subjected to load experiments.

For this purpose, vertical as well as horizontal forces must be applied at suitable points, with a minimum number of load alternations of $6 \cdot 10^6$. Instead of a load collective, it is permissible to apply only one load stage.

In this experiment, the program, set down in Appendix 4 of the Memorandum, for testing frames of passenger car trucks, is to be used.

3.5 Technical Braking Equipment

3.5.1* The standard equipment for trucks, according to Number 1.1, consists of two shaft brake discs per wheel set, optionally with an additional shoe brake, or four shoe brakes by themselves.

3.5.1.1 In special cases, wheel disc brakes instead of shaft disc brakes are permitted at all wheels, or special constructions of shaft disc brakes.

3.5.2* It must be possible to install an additional disc brake and/or additional shoe brake in addition to the standard disc brake with two shaft brake discs.

3.5.2.1* In the case of additional shoe brakes, besides the disc brake, the brake shoes utilized must conform to UIC Memorandum 541-1, Chapter III, Number 1.

3.5.3* For cars with higher loads (ratio of permissible total weight to self-weight $\geq$ 1.2), a load-dependent brake must be provided.

3.5.4* In the case of trucks for speeds above 140 km/h, it must be possible to install magnetic rail brakes suspended overhead.

3.5.5* At each wheel set, it must be possible to build in a speed indicator for the anti-skid system according to UIC Memorandum 541-1, Chapter X.

3.5.6 Installation of safety shackles for the brake rigging: see UIC Memorandum 541-1, Chapter III, Number 10.

3.5.7* Both wheel sets of the truck must be equipped with brake cylinders, which possess a handbrake connection.

3.5.7.1* The handbrake is connected to the car body through flexible pulls with a pressure jacket. Isolation must be possible at an easily accessible point.

3.5.8* The pneumatic and electrical connections of the car body must be designed flexibly and must be easily accessible.

3.5.9* A tension lock must be present in the brake rigging for initially adjusting the installation dimensions of the braking cylinder.

3.5.9.1 In the case of pin joints in the brake rigging, it is recommended that a basic play of 0.5 mm and the fitting H 11/c 11 be provided. The basic play must be taken into account in the boring.

3.5.10* With a built-in rod setting device, the brake cylinder must be able to accept the following, with the largest possible transmission in the brake rigging:

- play of the brake lining
- stretch excursion under maximum piston force
- wear of brake linings or respectively brake shoe soles
- in the case of disc brakes, wear of the brake discs
- in the case of shoe brakes, wear of the wheel rims
- wear of bolts and bushings

3.6* Vehicle Liftability

The construction of the truck must permit the complete truck to hang nearly vertically from the vehicle in a safe position and to be transported without risk of accident.

4.* Standard Construction Parts - Interchangeability

The subsequent standard or uniform construction parts are to be used:

4.1 Standard Wheel Set

(Still open)

(Conditions will be specified by ORE-SVA B 136.)

4.2 Wheel Set Bearings

4.2.1* Standard truck frames must be equipped with roller bearing construction types, which are utilized by the member railways.

4.2.2 Conditions for roller bearing boxes with two bearing systems for passenger cars.

For the roller bearing boxes of passenger cars, the same bearing systems should as much as possible be utilized as for the roller bearing boxes of freight cars.

With respect to interchangeability of the entire "wheel set with its bearing boxes", no specific directives are prescribed.

4.2.3* Basic values for constructing roller bearing boxes with two bearing systems:

- axle load (on the rails)	15 t
- running speed	140 km/h
- annual running performance	150,000 km
- lifetime	40 years for 75 percent of the bearing systems and 20 years for 90 percent of the bearing system

4.2.4* Two axle journals are permitted: one with 120 mm diameter and one with 130 mm diameter; these are specified in Table 5.

4.2.5* For the bearing box on axle journals with 120 mm diameter, the external diameter of the bearing systems shall be 220 mm or 240 mm.

4.2.6* For the bearing box on the axle journals with 130 mm diameter, the external diameter of the bearing systems shall be 220, 230, 240, or 250 mm.

4.2.7* The width of the two roller bearings in a wheel set roller bearing shall be 160 mm maximum. No specific directives are prescribed for the construction type of the rollers and for their installation.

4.2.8* The bearing boxes must have a lubrication interval of at least 24 months and must make possible a total running performance up to 150,000 km, under the presupposition that the greasing directives of the manufacturer are followed.

4.2.9 Wheel set bearing housing (still open).

5. Specifications for Simplifying Maintenance

5.1 Parts Subject to Wear

5.1.1 It is recommended that the number of parts subject to wear be kept as low as possible.

5.1.2* It must be possible to check without difficulty all wear points, which are present to maintain operating safety and running quality.

5.1.3* Parts subject to wear must be easily interchanged by easily loosened connections.

5.2 Wheel Set Linkage

5.2.1* To install and remove the wheel sets, it must only be necessary to fasten or respectively loosen the wheel set holder, the lift-off safety device, and the connections for the anti-skid device, grounding contact, etc.

5.2.2* Damper and pull-rod must be designed in such a manner that they need not be loosened when the wheel set is replaced.

5.2.3.1* It must be guaranteed that the wheel sets can be set parallel in simple fashion and that their parallelism can be retained.

5.2.3.2* If bearing play is necessary, it must be possible to adjust it in simple fashion.

5.3 Suspension

After being unmounted, secondary suspension sets must be capable of being again installed precisely in their old orientation.

5.4 Fastening of Hydraulic Dampers

5.4.1* It must be possible to replace dampers for the vertical vibration direction without a pit.

5.4.2* Dampers for the horizontal-transverse vibration direction, must be easily accessible from the pit, and must be exchangeable without the car body being lifted.

5.5 Brakes

5.5.1 All parts of the wheel set brake rigging (including the brake cylinder) should be removable from the truck frame as a compact unit.

5.5.2* The brake cylinder must be easily accessible to adjust the piston stroke or the brake lining play.

5.5.3* It must be possible to change the brake lining by opening or closing simple interlocks, so that only a few manual operations are required for this purpose.

5.5.3.1 It is recommended that screws not be used for this purpose.

5.5.4* In the case of wheel disc brakes or supplementary shoe brakes, the lining (brake shoe soles) must be replaceable without an additional external pit.

5.6 Truck Linkage to the Car Body

5.6.1 It is recommended that the number of connection points be kept as small as possible.

5.6.1.1* The connection points must be easily accessible to facilitate simple and quick installation and removal.

5.6.2 It is recommended that supplementary equipment for high speeds, e.g. swivel (turn) restraints and dampers for lateral damping motions, be designed so that they can, to the greatest possible extent,

- be checked during operation from the vehicle side
- be replaced without a pit

5.7 Adjusting the Buffer Level and Equalizing the Height for Wear

5.7.1 It is recommended that easily replaceable construction parts with a secure seat be used as supports.

5.7.2 It is recommended that construction parts for adjusting the height of buffers, e.g. sway braces or valve riggings for the air suspension, be designed so that they are easily accessible and so that they can be quickly and reliably adjusted.

5.7.2.1 It is recommended that these construction parts be protected against dirt and corrosion.

Appendix 1

Program and Conditions for Implementing the Running Experiments

(Edition: September 1978)

1. In General

The purpose of the running experiments is to determine riding quality (comfort and quiet running) and running safety.

1.1 Quiet running in particular includes:

- travel comfort for the passengers

Quiet running is evaluated by the following criteria:

1.1.1 - vertical accelerations, transverse and longitudinal accelerations in the car body, with respect to travel comfort

1.1.2 - accelerations of the truck frame.

1.2 Running safety is evaluated by the following criteria:

- stresses of the running gear and of the track.

1.3 With these experiments, the following tests are especially important:

1.3.1 On a tangent section

1.3.1.1 - quiet running at high speed

1.3.1.2 - reaction of the vehicle to track defects, with various track gauges, as well as crossings and switches, grade crossings and bridges.

1.3.2 <u>In curved tracks</u>

1.3.2.1 - Quiet running at maximum cross level deficiency

1.3.2.2 - The reaction of the vehicle at points which have defects.

1.4 In order to obtain this information concerning quiet running and running safety, the experiments must comprise runs on short line sections on a track, which has the parameters that have been selected for determining various quantities in dependence on speed and on cross level deficiency during curved runs and for v_{max} + 10 percent on tangent tracks.

2. Performance of Systematic Experiments

2.1 Formation of Experimental Train

The car being tested will be arranged at the end of the experimental train, normally short-coupled, presupposing that the preceding vehicle is of the same construction type. Otherwise, the experiments will be performed with loose coupling.

2.2 Condition of the Cars

2.2.1 During the experiments, the car must be operationally functional.

2.2.2 The experiments will be performed with the car empty.

2.2.3 The trucks are to be equipped with wheels with an unworn running profile - but with tool marks worn off - according to UIC Memorandum 250-2, Appendix 2a.

<u>Remark</u>: The report of the experiment must include the following:

- the track width of the wheel set
- a profile designation of the wheels of the measured vehicle

2.3 Experimental Lines

Length of the line section about 25 km, of this

2.3.1 - One section for a maximum speed of 220 km/h (tangent track and curved tracks with a radius of about 2000 to 5000 m).

2.3.2 - A section with curved tracks having radii between 300 and 1500 m.

2.3.3 The sections must have the most important parameters of track layout (gauge 1432 and 1435 mm, rail cant 1:20 and 1:40). They must be in an average maintenance condition corresponding to the permitted speed.

Remarks

The following must be obtained from the report of the experiment:

- the gauge
- canting of the rail
- standard deviation of the gauge
- standard deviation of the track surface
- standard deviation in the alignment

2.4 Running Speeds

The run will be performed at the following speeds:

2.4.1 - On the section with a tangent track and on curved track with $r \geqq 2000$ m: 160, 180, 200, and 220 km/h,

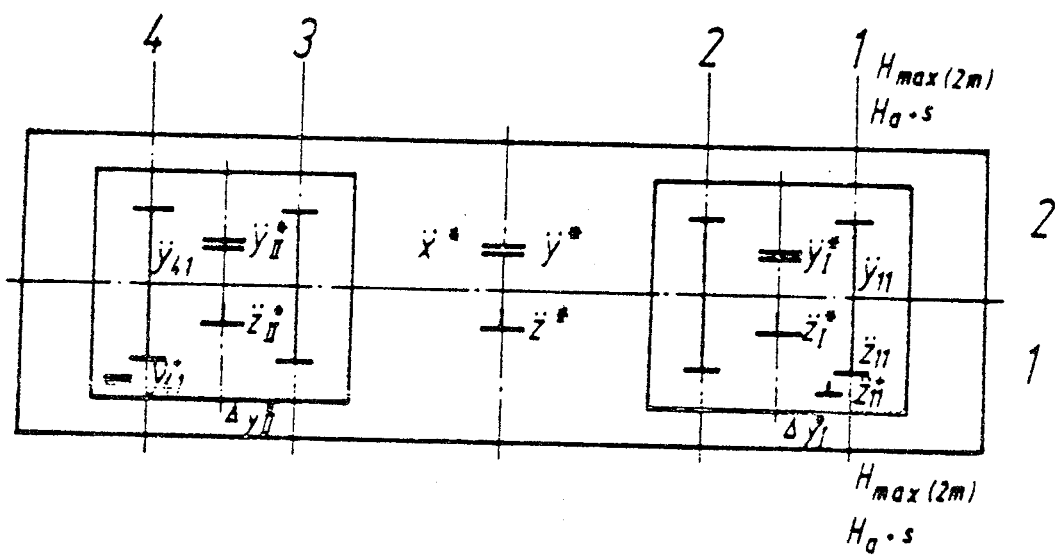
2.4.2 - On the section with the track curvature between 300 and 1500 m, with speeds corresponding to the cross level deficiency permitted by the railways (100, 130, and 160 mm).

2.4.3 The experimental runs must absolutely be performed on dry tracks for measurements relating to the dynamics in the transverse direction.

3. Measured Parameters and Arrangement of Accelerometers

3.1 Diagram of Measurement Points

The position of the accelerometers on the vehicle is apparent from the diagram of measurement points. The numbers written in the direction of the front wall correspond to the numeration of the car sides. The wheel sets have numbers 1-4 in the direction of travel.



The wheel set roller bearing transverse forces are to be measured at the wheel sets at which the largest forces are expected.

As a rule, the largest forces are expected:

- in the case of curved runs and on clear trouble spots in tangent tracks: at the leading wheel set of the leading truck.
- when resonances and instabilities occur on tangent tracks: at the trailing wheel set of the trailing truck.

If not all the wheel sets are equipped with force measurement devices, the vehicle may be turned around for these experiments.

3.2 Measured Quantities for Running Quality and Running Safety

Designations	Measured Quantities	Filters: Recording Paper	Filters: Magnetic Tape Recording
$\ddot{z}_{II}^{*}$ and $\ddot{z}_{I}^{*}$	Vertical acceleration of the car body above the kingpin of the two trucks	16 Hz	40 or 125 Hz
$\ddot{y}_{II}^{*}$ and $\ddot{y}_{I}^{*}$	Transverse acceleration of the car body above the kingpin of the two trucks	16 Hz	40 or 125 Hz
$\ddot{z}^{*}$	Vertical acceleration at the body center	16 or 20 Hz	40 or 125 Hz
$\ddot{y}^{*}$	Transverse acceleration at the body center	16 Hz	40 or 125 Hz
$\ddot{x}^{*}$	Longitudinal acceleration of the car body at the car center	16 or 20 Hz	40 or 125 Hz
η^{*}	Angle of roll motion (calculation of slant coefficient)		
Δy^{*}	Transverse distances car body - truck		

Designations	Measured Quantities	Filter: Recording Paper	Filter: Magnetic Tape Recording
$\ddot{z}\,\overset{+}{11}$ or $\ddot{z}\,\overset{+}{12}$	Vertical acceleration of the truck frame above the first wheel set	20 or 32 Hz	40 or 125 Hz
$\ddot{y}\,\overset{+}{41}$ or $\ddot{y}\,\overset{+}{42}$	Transverse acceleration of the truck frame above the fourth wheel set (check of quiet running)	16 or 32 Hz	40 or 125 Hz
	Pitching motion of the truck	-	
	Turning motion of the truck	-	
$\ddot{z}$ 11 or $\ddot{z}$ 12	Vertical acceleration of an axle box	32 Hz	40 or 125 Hz
$\ddot{y}$ 11 or $\ddot{y}$ 12	Transverse acceleration of wheel set 1	32	125 Hz
$\ddot{y}$ 41 or $\ddot{y}$ 42	Transverse acceleration of wheel set 4	32	125 Hz
ΣY max (2 m)	Maximum transverse wheel set force		
$\Sigma Y_a + s$	Average transverse wheel set force plus standard deviation		
H_{max} (2 m)	Maximum wheel set roller bearing transverse force	-	125 Hz
$H_a + s$	Average wheel set roller bearing transverse force plus standard deviation	-	125 Hz

3.3 Filtering

Limit frequency of the filters = - 3 dB

Filter characteristic $\geq$ 24 dB per octave

4. Recording and Evaluation of Measured Quantities, Presentation of Results

4.1 In order to reduce the amount of labor and to increase the direct and detailed evaluation possibilities, it is necessary to use a magnetic tape recording method. Oscilloscope records on paper are used as a supplement for the direct observation and preliminary evaluation of the measured results.

Comfort evaluation grades and a general evaluation of running safety must be immediately apparent from the evaluation. The further processing to determine the spectrum of the power density and the arithmetic and/or square mean value, however, should be performed only according to need and, where appropriate, for parts of the track.

The accelerations $\ddot{z}\,\overset{*}{I}I$ and $\ddot{z}\,\overset{*}{I}$ and $\ddot{y}\,\overset{*}{I}I$ and $\ddot{y}\,\overset{*}{I}$ are recorded simultaneously on the oscillogram and possibly simultaneously on magnetic tape.

The acceleration $\ddot{y}^{*}$ is required only where necessary.

The accelerations $\ddot{y}\,\overset{*}{I}$ and $\ddot{y}\,\overset{*}{I}I$ must be checked to guarantee the reliability of the experiments; they may not exceed certain limit values, which depend on the construction type of the truck[1].

These accelerations and their frequencies form the foundations for evaluating quiet running and travel comfort.

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[1]For the SNCF, this limit is 3.5 m/s^2.

4.2 In the case of the accelerations $\ddot{z}\,\overset{+}{1}1$ and $\ddot{y}\,\overset{+}{4}1$, the spectrum of the power density and the arithmetic average and/or square average must be determined in certain cases.

4.3 The acceleration $\ddot{y}\,\overset{+}{4}1$ serves to check the running safety of the truck. Limit safety values[1] can be specified for a certain length of the measurement line.

4.4 $\ddot{y}$ 11 and $\ddot{y}$ 41 need be recorded only when a transverse elasticity exists between the wheel set and the truck frame.

4.5 Angle of roll motion from static measurements, to check the slant coefficient.

4.6 In the case of accelerations $\ddot{z}$ 11, the power density spectrum and the arithmetic and quadratic mean value can be evaluated. This permits an evaluation of track quality in the vertical direction.

5. Report of the Experiment

5.1 The experimental results are to be presented in the form of a report.

5.2 Besides a summary text, this report should contain data, which specify the experimental results necessary to interpret the measurements. A selection of characteristic measurement recordings should be included.

Furthermore, the measurement and evaluation procedure utilized is to be specified.

[1] The SNCF considers that the critical speed has been reached when the transverse acceleration $\ddot{y}\,\overset{+}{4}1$ has more than 6 successive oscillations with an amplitude exceeding 8 m/s^2 and a frequency of 4-8 Hz.

Appendix 2

Conditions for Setting up the Maximum Clearance Envelope

(Edition: February 1978)

1. Anticipated Truck Construction Types and Group Classification

Center Support	Lateral Support	
	with	without
	centering	
MD 36	MD 52*	Y 0270 S (Fiat)
MD 52*	LD 70 (73)	Y 32

*The MD 52 truck has two construction variants

Variant 1: with swivel ring (center support)

Variant 2: with centering pin (lateral support)

2. Maximum Clearance Envelope

The presentation of the maximum clearance envelope (space required by the truck on a tangent track including vertical excursions from the load and from wear and horizontal excursions from transverse play in the primary and secondary suspension stages) will be provided as the basic data for a new memorandum for the running gear of passenger cars.

Scale of drawing: 1:10

Format of drawing: DIN A 1, subsequent reduction: DIN A4

In exceptional cases: DIN A O, subsequent reduction: DIN A3

2.1 Dimensional Conditions - Height Dimensions -

All height dimensions are to be referred to the distance between the lower edge of the main transverse support and SO of 1000 mm, corresponding to a standard buffer height of 1060 mm (empty car). A new condition without wear is to be used as a basis for this dimension, relative to a specific rated wheel diameter.

In addition, height measurements are to be specified with an index for a car which has all-around maximum wear, and whose secondary suspension play has been used up to the stop.

2.2 Dimensional Conditions - Weight Dimensions -

Width dimensions are to be referred to the center of the wheel set

Longitudinal play in the primary suspension stage (including transverse bearing play) is to be accounted to the width dimensions of the frame and of those parts that are rigidly connected therewith.

The transverse play in the secondary suspension stage (including elasticity of the stops) is to be accounted to the width dimensions of the secondary suspension or upper traverse and to all parts that are rigidly connected therewith.

Explanation:

Since the maximum clearance envelope of the trucks is to be referred to the boundary lines of the UIC Memoranda 505-2 and 3, which are constructed from the truck center, the width dimensions must be evaluated from the bottom to the top (truck, wheel set, truck frame, car body).

With the task defined in this way, the problem would be unnecessarily complicated if the longitudinal center of the car became the reference zero line.

2.3 Displays

2.3.1 The connecting elements, and also the wheel set holder, lift-off protection device, connections for anti-skid device, grounding contact, etc., are to be presented in Table I.

Displays are to be in three views. Height dimensions according to 1060 mm buffer height of the truck in new condition.

Width dimensions as rated dimensions without consideration of transverse play. Front view* in full section or half section.

In the case of trucks with supports that are off-center in the direction of travel, two sections or a displaced section course are to be provided.

Textual explanation of dimensions are to be affixed above the writing area as above.

2.3.2 The maximum clearance envelope of the truck is to be presented in Table II.

Displays in

2.3.2.1 - side view for horizontal track with all height and length dimensions

2.3.2.2 - side view in a section through the longitudinal center of the truck, for horizontal track, with supplementary dimensions

---

*Direction of view - direction of travel

2.3.2.3 - top view; a half-view is possible for symmetric trucks; contains all transverse and longitudinal dimensions

2.3.2.4 - view in the direction of travel in a half-section through the support plane, with supplementary dimensions

2.3.2.5 - additional partial sections and dimensions, if these are particularly required

2.3.2.6 - textual specifications above the writing area, corresponding to the specifications under Point 2.1 and Point 2.2.

The underside of the maximum clearance envelope is determined by the fully loaded car, counting all wear and the play of the suspension as far as the stop, or respectively by the boundary of the vehicle (Gabarit).

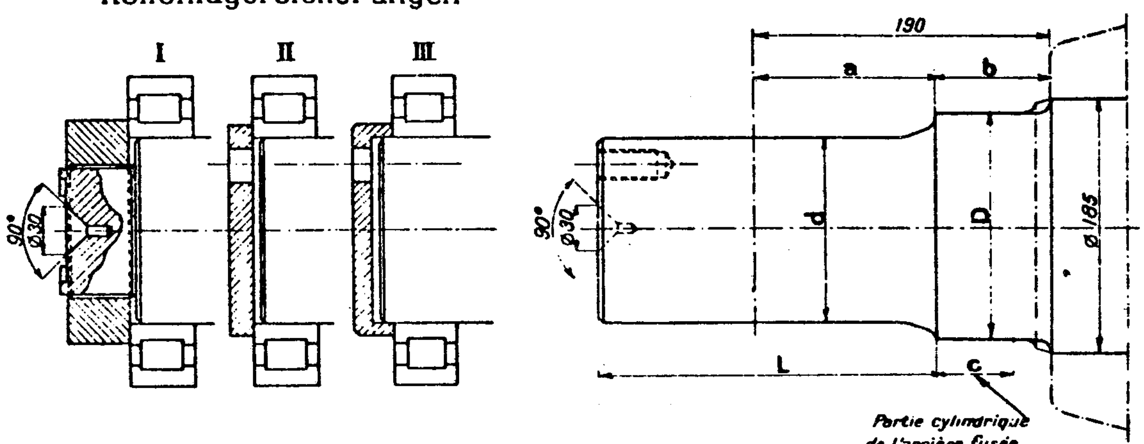
The upper side is determined by the empty vehicle (buffer height 1060 mm) without measurements for wear.

2.3.3 A perspective view of the maximum clearance envelope of the trucks is to be presented in Table III. The half-sections in the transverse and longitudinal center are to be drawn into this perspective view.

BOITES A ROULEAUX – ROLLER BEARING BOXES – ROLLENACHSLAGER
FUSÉES D'ESSIEUX – AXLE JOURNALS – ACHSSCHENKEL

PLANCHE 5
PLATE 5
TAFEL 5

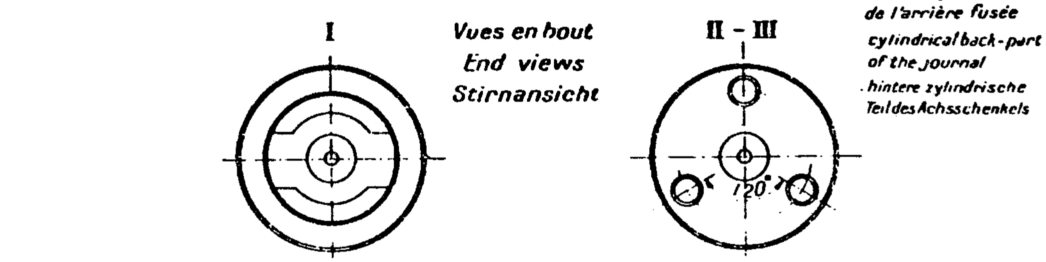
Dispositifs de blocage des roulements
Locking devices of rollerbearings
Rollenlagersicherungen



La portée de calage de l'essieu peut être précédée d'une entrée conique figurée en traits mixtes aux dessins ci-contre.

The outer end of the wheel seat may be tapered off as shown by the chain dotted line.

Der Nabensitz der Achsen kann einen in nebenstehender Zeichnung strich-punktiert angegebenen konischen Übergang haben.



	I - II	III	
d	120 p 6	120 p 6	130 p 6
L	179 + 0,5	191 + 1	217 + 1
a	101	101	117
b	89	89	73
c	50 min	50 min.	45 min
D	146 u 9	146 u 9	160 t 7

APPENDIX 2.1

Calculation of Permissible Railhead Load on American Rails RE 136, according to the Method of the German Federal Railway

Data

Wheel diameter for 80 t vehicle = 33" = 83.82 cm Ø = 41.61 cm Radius

Wheel diameter for 100 t vehicle = 36" = 91.44 cm Ø = 45.72 cm Radius

Wheel diameter for 120 t vehicle = 38" = 96.52 cm Ø = 48.26 cm Radius

Tensile stress of USA rails

σ = 100,000 psi = 703 N/mm^2

Calculation for 120 t vehicle:

Assumption: Safety factor ν = 1, since only freight traffic runs, long term stress with infinitely many load alternations:

maximum permissible wheel force $Q = 6.00 \cdot 10^{-7} \cdot r \cdot (\sigma/\nu)^2$ (kN)

$$Q = 6.00 \cdot 10^{-7} \cdot 482.6 \cdot (703/1)^2$$

$$Q = 142 \text{ (kN)}$$

r = wheel radius (mm)

ν = safety factor

Q = wheel force (kN)

Actual wheel force Q:

$$\frac{\text{Self-weight + load}}{\text{4 axles}} = \frac{\sim 200 + 1200}{4 \times 2} = 175 \text{ (kN)}$$

Result:

Permissible Q $\ll$ actual Q

i.e., the rail is considerably overloaded and will rupture prematurely

Calculation for 100 t vehicle:

Safety factor $\gamma = 1$

Permissible $Q = 6.00 \cdot 10^{-7} \cdot 457.2 \cdot 703^{2}$ (kN)

Permissible $Q = 134$ (kN)

Actual $Q = \frac{200 + 1000}{8} = 150$ (kN)

Result:

Permissible $Q < $ actual Q

i.e., the rail likewise cannot withstand this load for a long time, but the difference from the permissible load was not as great as in the calculation for the 120 t vehicle.

Measures to avoid rail ruptures:

1. Increase wheel diameter
2. Increase the strength of the rail steel.

APPENDIX 2.2

German Federal Railway

Test Booklet for Switches

for

Switches and Crossings with Rails

UIC 60

for the Area of the Signaling Post....................

(Switchman Post.....................................)

at the Station......................................

(Block post...)

899 49 Test Booklet for Switches (Form UIC 60) A4h16 5c70

in 7d250 Munich VII77 M 2001

1. Directives for Testing Switches and Crossings

1.1 Tests

Apart from the general test according to the business directive for service agency supervisors - DV 162 - § 25, every switch and crossing must be tested once a year, and twice a year in the case of a local speed limit in excess of 160 km/h. Additional investigations will be ordered separately in case of need.

The tests, which are to be performed by the signaling service, are specified in the directive for the maintenance of signaling systems - DV 892.

1.2 Test Officials

Switches and crossings in continuous primary tracks are to be tested by the supervisor of the railroad district or his representative. The supervisor of the railroad district may delegate the tests of all remaining switches to a suitable official among the track foremen (construction). In exceptional cases, the German Federal Railway may allow deviations from the preceding rule.

1.3 Scope of the Tests

For the test, the switches must be cleaned to the extent necessary. The test must extend to all components (compare Section 2), to the track surface, the alignment, and the maintenance condition of switches and crossings. The result of the test must be recorded on the switch file sheet.

Damaged parts must be repaired or renewed. Removal of defects or respectively the proper condition of switches/crossings must be certified in the switch file sheet.

1.4 Notification by the Agency Supervisor

The test booklet for switches must be presented to the supervisor of the Operations Office or his representative on the occasion of an audit, a test of a signaling system, or at another opportunity. By signing his name and specifying the date, the supervisor will, in Section 3, confirm that he has verified the entries of the district supervisor or his representative,

- at least once a year for lines with a local speed limit $\geqq$ 160 km/h,
- at least every two years for the remaining primary lines
- at least every three years for secondary lines.

At the same time, he will convince himself of the correctness of the measured results by a random local follow-up measurement at at least one switch per station gridiron.

He will supervise execution of defect removal by the district inspector.

2. Test List

2.1 Test of Screws and Rail Fastenings

2.1.1 Test all screw connections for rigid seating. Loose screws in the following screw connections must be tightened with a torque of about 980 Nm:

- Running rail - guard rail shackle,
- Frog - Wing rail,
- Fastening of the rail supports.

2.1.2 Check prescribed tensioning of the elastic clips

2.1.3 Test positioning and condition of clips in the slide plates

2.1.4 Test functioning of screw and nut locking devices

2.2 Dimensional Tests

2.2.1 Test tie position according to laying plan

2.2.2 Test gage according to switch file sheet

2.2.3 Check back-to-back space according to switch file sheet

2.2.4 Check switch openings according to switch file sheet

pendix I The position of the measuring points in the area of simple frogs is shown in Appendix I.

2.3 Test for Wear

2.3.1 Test all the rails of a switch, especially for cracks

2.3.2 Check condition of ties

2.3.3 Check condition of base plates

2.3.4 Check seating of pads

2.3.5 Check lateral wear of frogs and stock rails; in the case of advanced wear, see directions for using the stock rail and frog test gage

2.3.6 Check wear of the remaining rails

2.3.7 Check condition of frogs - wear and burr formation

2.3.8 Check condition of welding joints and insulating joints

2.3.9 Additional wear test for switches with rigid frogs

Appendix II (compare Appendix II)

2.3.9.1 Wear and burr formation at the frog block and wing rails

2.3.9.2 Condition of the transition to the junction lines at the linkage.

2.3.9.3 Rigid seating and condition of block supports, rail supports, and retainers.

2.3.9.4 Rigid seating of locking plate and its screw connections

2.3.9.5 Condition of weld joints in the junction lines (full rail/standard rail)

2.3.10 Additional work tests for switches with elastic frogs (compare Appendix III):

2.3.10.1 Wear and burr formation at the frog block and wing rails

2.3.10.2 Condition of the running surfaces of the bias joint

2.3.10.3 Solid seating and condition of block supports, rail supports, and retainers

2.3.10.4 Solid seating of locking plates and their screw connections

2.3.10.5 Condition of welded joints between the frog block and junction lines

2.3.10.6 Condition of the bias joint (contact of the basic point, screw connnection with bolts, castellated nut and pin).

2.4 Function Tests

2.4.1 Check contact of tongues with stock rails

2.4.2 Check contact of tongues to rail supports

2.4.3 Additional function tests for switches with rigid frogs

2.4.3.1 Solid contact of the frog with the wing rails, check by resetting the frog several times

2.4.4 Additional function tests for switches with elastic frogs

2.4.4.1 Solid contact of the frog to the wing rails

2.4.4.2 Mobility of the expansion joint

899 49/1 Switch File Sheet A4L.7b-250 light grey

1. German Federal Railway; 2. Station; 3. Tower area; 4. Switch number; 5. Tower connection; 6. Mechanical electrical; 7. Simple switch; 8. I-A curve switch; 9. With rigid frog; 10. Measurement points; 11. Produced according to switch sketch number; 12. Delivery plant; 13. Installed on; 14. Removed on; 15. Date of test and name of tester; 16. Design dimensions (mm); 17. Operational limit dimensions (mm); 18. Main track; 19. Branch track; 20. Gages; 21. Actual dimensions (finding); 22. Guide values; 23. Guide rail switch opening; 24. Frog switch opening; 25. Defects found (according to Section 2 of the Test Booklet for Switches), replacement parts required; 26. Defects removed name/date; 27. Proper condition of the switch is certified: Name, Title, Date; 28. 1) Cross out if not applicable; 2) The S_z dimensions must be measured when the switching device is in its branching position; 3) The Sh and Sh_z dimensions are to be measured 200 mm behind the frog; 4) Impermissible deviations are to be underlined in red; 5) Gage measurements S_1 - S_g at a distance of 10 tie spacings (X) each - main tracks and branch tracks; 6) If the L and L_z dimensions have a plus deviation, the associated H-dimensions may not have any minus deviations; 7) If the L and L_z dimensions have a minus deviation, the R-dimensions may not have any minus deviation.

254

Deutsche Bundesbahn 1

Bahnhof 2 .... Stellwerksbez 3 ....

Weiche-Nr. 4 .... Stellwerksanschluß 5 ... mech., elektr.[1] 6

7 E... 60-300-1:9 / -300-1:14 / -500-1:12 / -500-1:14 / 60-760-1:14 / -760-1:18,5 / -1200-1:18,5 [1]

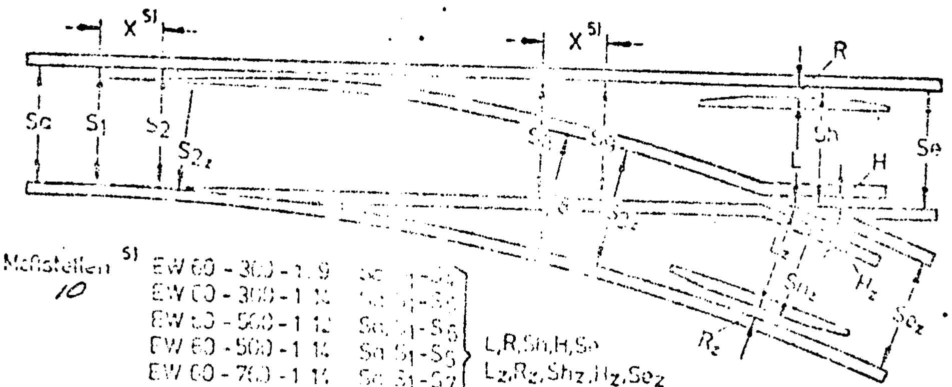
I-A-Bogenweiche 8 $\frac{R}{R_z}$ = ———— mit starrem Herzstück 9

11 Angefertigt nach Weichenskizze Nr. ....

12 Lieferwerk ....

13 Eingebaut am ....

14 Ausgebaut am ....



Meßstellen 5) 10

EW 60 - 300 - 1:9	Sa, S1 - S5	
EW 60 - 300 - 1:14	Sa, S1 - S6	
EW 60 - 500 - 1:12	Sa, S1 - S6	L, R, Sh, H, Se
EW 60 - 500 - 1:14	Sa, S1 - S6	$L_z, R_z, Sh_z, H_z, Se_z$
EW 60 - 760 - 1:14	Sa, S1 - S7	
EW 60 - 760 - 1:18,5	Sa, S1 - S7	
EW 60 - 1200 - 1:18,5	Sa, S1 - S9	

1	2	3												4	5	6	7	8	9
15 Tag der Prüfung und Name des Prüfenden		Spurweiten 20												Leitweiten 22	Radlenkerrillen 23	Herzstückrillen 24	Vorgefundene Mängel (nach Abschnitt 2 des Weichenprüfungsbuches) erforderliche Ersatzteile 25	Mängel beseitigt 26 Name Datum	Der ordnungsgemäße Zustand der Weiche wird bescheinigt. 27 Name Amtsbezeichnung Datum
	Sollmaße (mm) 16	1433												1394	39	44			
	Betriebsgrenzmaße (mm) 17	1433^{+10}_{-3} ($S_{max} - S_{min} \leqq 10$ mm)												$1394^{+2\ 6)}_{-2\ 7)}$	39^{+4}_{-2}	44^{+4}_{-2}			
	Stammgleis 18	Sa	S_1	S_2	S_3	S_4	S_5	S_6	S_7	S_8	S_9	$Sh^{3)}$	Se	L	R	H			
	Zweiggleis 2) 19	-	-	S_{2z}	S_{3z}	S_{4z}	S_{5z}	S_{6z}	S_{7z}	S_{8z}	S_{9z}	$Sh_z^{3)}$	Se_z	L_z	R_z	H_z			
Istmaße (Befund) 4) 21																			
	Stammgleis	[illegible]																	
	Zweiggleis																		
	Stammgleis																		
	Zweiggleis	[illegible]		[illegible]															
	Stammgleis																		
	Zweiggleis																		
	Stammgleis																		
	Zweiggleis																		
	Stammgleis																		
	Zweiggleis																		

1) Nichtzutreffendes streichen.
2) Die S_z-Maße sind bei abzweigender Lage der Zungenvorrichtung zu messen.
3) Die Maße Sh und Sh_z sind 200 mm hinter der Herzstückspitze zu messen.
4) Unzulässige Abweichungen sind rot zu unterstreichen. 28
5) Spurweitenmessungen S_1 - S_9 im Abstand von je 10 Schwellenfeldern (X) - Stamm- u. Zweiggleis.
6) Bei Plusabweichungen der L- u. L_z-Maße dürfen die dazugehörenden H-Maße keine Minusabweichungen haben.
7) Bei Minusabweichungen der L- u. L_z-Maße dürfen die R-Maße

899 49 / 1 Weichenkarteiblatt A 4L. 7b-250 hellgrau

Anlage 1 (Vorbem.)

Deutsche Bundesbahn 1

Bahnhof 2 Stellwerksbez. 3

Weiche-Nr. 4 Stellwerksanschluß 5 mech., elektr. 1) 6

7 Einfache Weiche 60-300-1:9/-500-1:12/-500-1:12 (45,361 m) 1)
60-760-1:14/-1200-1:18,5/-1200-1:18,5 (65,615 m)

8 1-A-Bogenweiche $\frac{R}{R_z}$ = ———

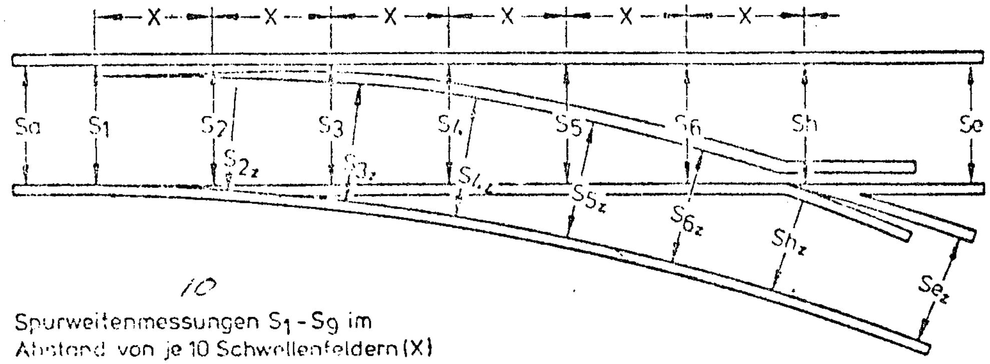
mit beweglicher Herzstückspitze 9 $\frac{\text{federnd}}{\text{gelenkig}}$ 1)

11 Angefertigt nach Weichenskizze Nr ....

12 Lieferwerk ....

13 Eingebaut am ....

14 Ausgebaut am



10 Spurweitenmessungen S_1 - S_9 im Abstand von je 10 Schwellenfeldern (X) im Stamm- und Zweiggleis

Anlage 2 (Vorbem.)

1	2	3												4	5	6
15 Tag der Prüfung und Name des Prüfenden		Spurweiten 20												Vorgefundene Mängel (nach Abschnitt 2 des Weichenprüfungsbuches) erforderliche Ersatzteile 22	23 Mängel beseitigt Name Datum	Der ordnungsgemäße Zustand der Weiche wird bescheinigt: Name Amtsbezeichng. Datum 24
	Sollmaße 16	1433														
	Betr. abs-grenzmaße	17 1433^{+10}_{-3} ($S_{max} - S_{min} \leqq 10$ mm)														
	Stammgleis 18	Sa	S_1	S_2	S_3	S_4	S_5	S_6	S_7	S_8	S_9	Sh 3)	Se			
	Zweiggleis 2) 19	-	-	S_{2z}	S_{3z}	S_{4z}	S_{5z}	S_{6z}	S_{7z}	S_{8z}	S_{9z}	Sh_z 3)	Se_z			
		Istmaße (Befund) 4) 21														
	Stammgleis															
	Zweiggleis															
	Stammgleis															
	Zweiggleis															
	Stammgleis															
	Zweiggleis															
	Stammgleis															
	Zweiggleis															
	Stammgleis															
	Zweiggleis															

1) Nichtzutreffendes streichen.
2) Die Sz-Maße sind bei abzweigender Lage der Zungenvorrichtung bzw. der Herzstückspitze zu messen. 25
3) Die Maße Sh und Sh_z sind 200 mm hinter der Herzstückspitze zu messen.
4) Unzulässige Abweichungen sind rot zu unterstreichen.

890 49/2 Weichenkarteiblatt A4L 7b-250 hellgrau

899 49/2 Switch File Sheet A4L 7b-250 light grey

1. German Federal Railway; 2. Station; 3. Tower area; 4. Switch number; 5. Tower connection; 6. Mechanical electrical; 7. Simple switch; 8. I-A curve switch; 9. With elastic/rigid frog; 10. Gage measurements $S_1 - S_g$ at a distance of 10 tie spacings (X) each in main tracks and branch tracks; 11. Produced according to switch sketch number; 12. Delivery plant; 13. Installed on; 14. Removed on; 15. Date of test and name of tester; 16. Design dimensions (mm); 17. Operational limit dimensions (mm); 18. Main track; 19. Branch track; 20. Gages; 21.Actual dimensions (finding); 22. Defects found (according to Section 2 of the Test Booklet for Switches), replacement parts required; 23. Defects removed name/date; 24. Proper condition of the switch is certified: Name, Title, Date; 25. 1) Delete if not applicable; 2) the S_z dimensions are to be measured with the switching device or the frog in its branching position; 3) the Sh and Sh_z dimensions are to be measured 200 mm behind the frog; 4) Impermissible deviations are to be underlined in red

J

Deutsche Bundesbahn 1

Bahnhof 2 .... Stellwerksbez 3 ....

Weiche - Nr. 4 .... Stellwerksanschluß 5 ... mech., elektr.[1] 6

7 Einfache Weiche 60 - 2500 - 1 : 26,5

8 I - A - Bogenweiche $\frac{R}{R_Z}$ = ———

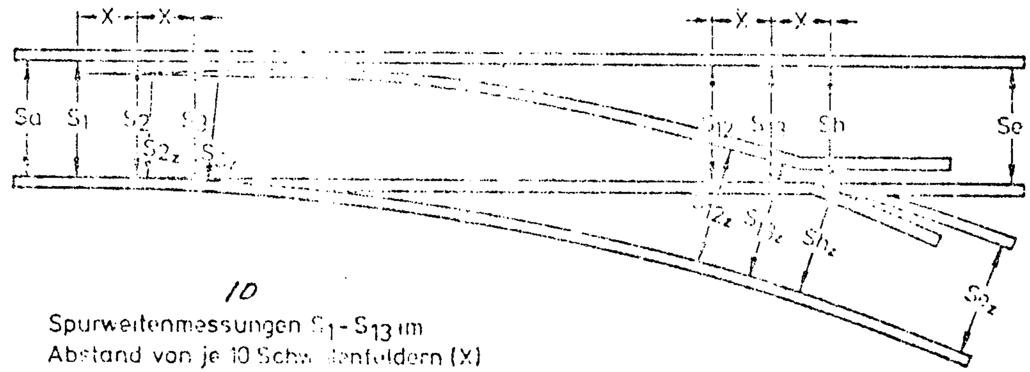
mit federndbeweglicher Herzstückspitze 9

11 Angefertigt nach Weichenskizze Nr. ....

12 Lieferwerk ....

13 Eingebaut am ....

14 Ausgebaut am ....



10 Spurweitenmessungen $S_1 - S_{13}$ im Abstand von je 10 Schwellenfeldern (X) im Stamm- und Zweiggleis

257

1	2	3																4	5	6
15 Tag der Prüfung und Name des Prüfenden		Spurweiten 20																Vorgefundene Mängel (nach Abschnitt 2 des Weichenprüfungsbuches) erforderliche Ersatzteile 22	23 Mängel beseitigt Name / Datum	Der ordnungsgemäße Zustand der Weiche wird bescheinigt. 24 Name Amtsbezeichng. Datum
	Sollmaße (mm)	16 1435																		
	Betriebsgrenzmaße (mm)	17 1435^{+8}_{-5} ($S_{max} - S_{min} \leqq 10$ mm)																		
	Stammgleis 18	Sa	S_1	S_2	S_3	S_4	S_5	S_6	S_7	S_8	S_9	S_{10}	S_{11}	S_{12}	S_{13}	Sh	Se			
	Zweiggleis[2] 19	-	-	S_{2z}	S_{3z}	S_{4z}	S_{5z}	S_{6z}	S_{7z}	S_{8z}	S_{9z}	S_{10z}	S_{11z}	S_{12z}	S_{13z}	Sh_z[3]	Se_z			
Istmaße (Befund)[4] 21																				
	Stammgleis																			
	Zweiggleis																			
	Stammgleis																			
	Zweiggleis																			
	Stammgleis																			
	Zweiggleis																			
	Stammgleis																			
	Zweiggleis																			
	Stammgleis																			
	Zweiggleis																			

1) Nichtzutreffendes streichen.

2) Die S_z-Maße sind bei abzweigender Lage der Zungen an [illegible] bzw. der Herzstückspitze zu messen

3) Die Maße Sh und Sh_z sind 200 mm hinter der Herzstückspitze zu messen

4) Unzulässige Abweichungen sind rot zu unterstreichen

25

899 49/3 Switch File Sheet A4L 7b-250 light grey

1. German Federal Railway; 2. Station; 3. Tower area;
4. Switch number; 5. Tower connection; 6. Mechanical electrical;
7. Simple switch; 8. I-A curve switch; 9. Elastic frog;
10. Gage measurements S_1 - S_{13} at a distance of 10 tie spacings (X) each in main tracks and branch tracks; 11. Produced according to switch sketch number; 12. Delivery plant; 13. Installed on;
14. Removed on; 15. Date of test and name of tester; 16. Design dimensions (mm); 17. Operational limit dimensions (mm); 18. Main track; 19. Branch track; 20. Gages; 21. Actual dimensions (finding);
22. Defects found (according to Section 2 of the Test Booklet for Switches), replacement parts required; 23. Defects removed name/date;
24. Proper condition of the switch is certified: Name, Title, Date;
25. 1) Delete if not applicable; 2) the S_z dimensions are to be measured with the switching device or the frog in its branching position; 3) the Sh and Sh_z dimensions are to be measured 200 mm behind the frog; 4) Impermissible deviations are to be underlined in red

Deutsche Bundesbahn 1

Bahnhof 2 .... Stellwerksbez 3 ....

Kreuzung - Nr. 4 .... 5 Stellwerksanschluß mech elekt 1) 6

7 Kreuzung [illegible] - 1:14 / - 1:18,5 / - $\frac{1200}{\omega}$ - 1:11,515

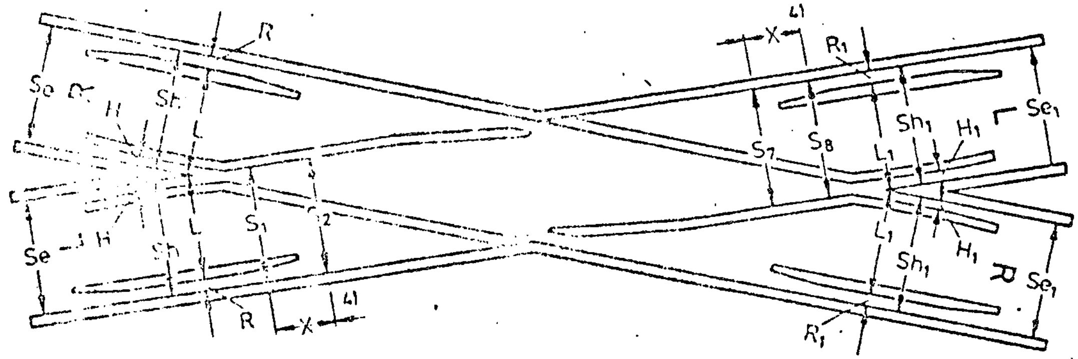
8 mit beweglich - doppeltem Herzstück

10 Angefertigt nach Weichenskizze Nr ....

11 Lieferwerk ....

12 Eingebaut am ....

13 Ausgebaut am ....



259

1	2	3												4		5		6		7	8	9
14 Tag der Prüfung und Name des Prüfenden	Sollmaß 17 Befund [illegible]	Spurweiten 19												21 Leitweiten		22 Radlenkerrillen		23 Herzstückrillen		24 Vorgefundene Mängel (nach Abschnitt 2 des Weichenprüfungsbuches) erforderliche Ersatzteile	25 Mängel beseitigt Name Datum	Der ordnungsgemäße Zustand der Weiche wird bescheinigt: 26 Name Amtsbezeichng. Datum
		15 1433												1394		39		44				
		16 1433^{+10}_{-3} ($S_{max} - S_{min} \leq 10$ mm)												1394^{+2}_{-2} 5) 6)		39^{+4}_{-2}		44^{+4}_{-2}				
	18	Se	Sh	S1	S2	S3	S4	S5	S6	S7	S8	Se1	Sh1	L	L1	R	R1	H	H1			
Istmaße (Befund) 3) 20																						
siding																						

1) Nichtzutreffendes [illegible]

2) Die Maße [illegible] Außen hinter der Herzstückspitze zu messen

3) Unzulässige Abweichungen sind rot zu unterstreichen

4) Spurweitenmessungen S1 - S8 im Abstand von je 10 Schwellenfeldern (X) bei entsprechender Lage der beweglichen Herzstückspitzen in den Kreuzungsgleisen L u R Kr 60 - 1:14 .... S1 - S6 ; Kr 60 - 1:18,5 .... S1 - S8 ; Kr 60 - $\frac{1200}{\omega}$ - 1:11,5 .... S1 - S5.

5) Bei Plusabweichungen der L- und L1-Maße dürfen die dazugehörenden H-Maße keine Minusabweichungen haben

6) Bei Minusabweichungen der L- und L1-Maße dürfen die R-Maße keine Minusabweichungen haben.

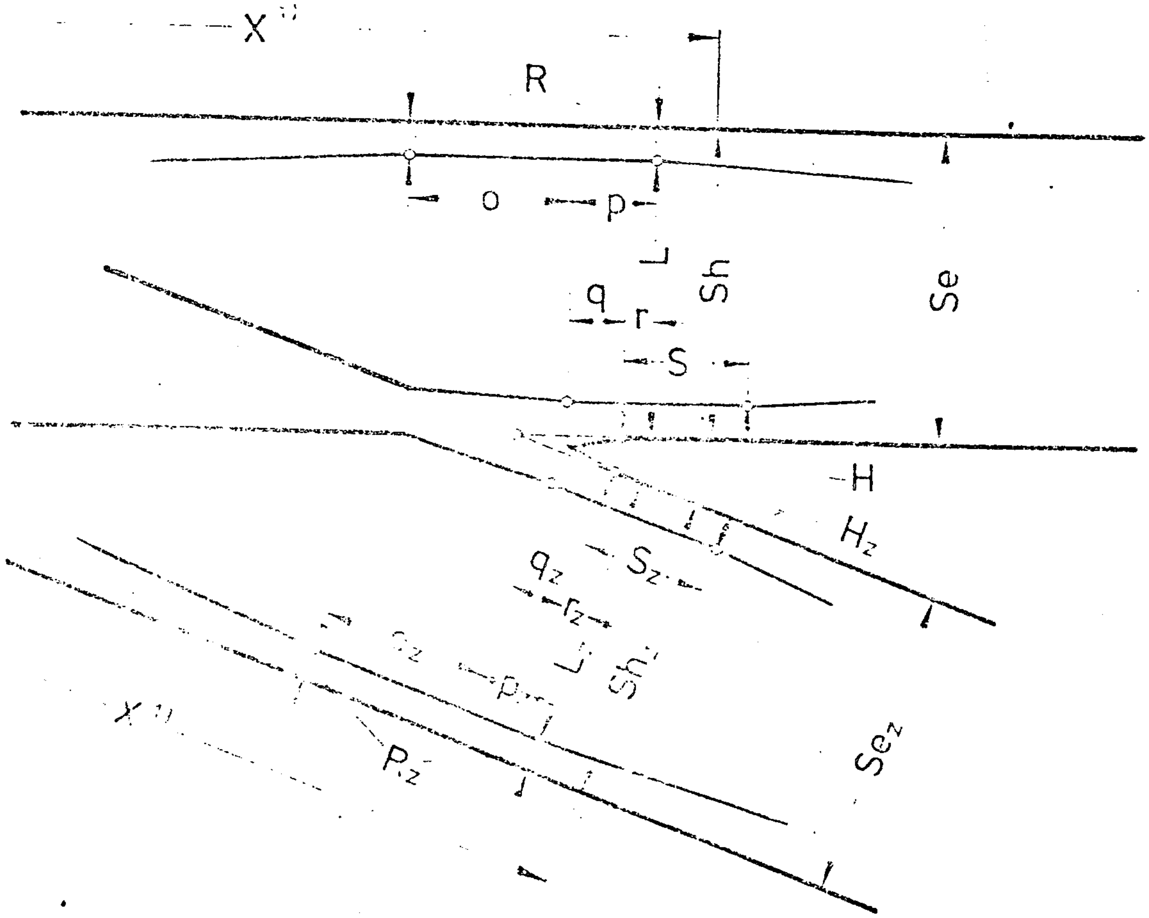
27

899 49/4 Switch File Sheet A4L 7b-250 light grey

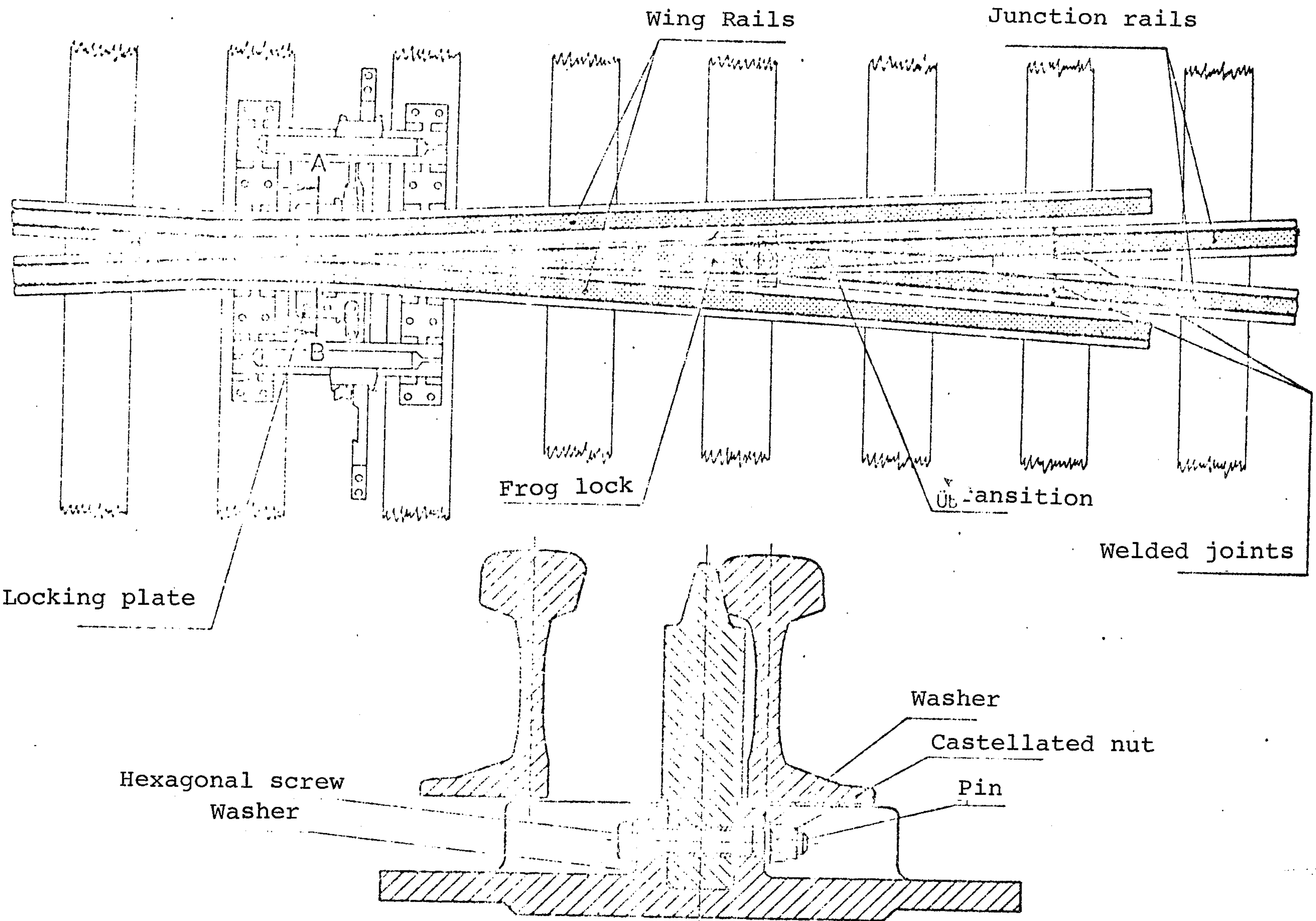
1. German Federal Railway; 2. Station; 3. Tower area; 4. Siding number; 5. Tower connection; 6. Mechanical electrical; 7. Siding; 8. With double frog; 10. Produced according to switch sketch number; 11. Delivery plant; 12. Installed on; 13. Removed on; 14. Date of test and name of tester; 15. Design dimensions (mm); 16. Operational limit dimensions (mm); 17. Siding L; 18. Siding R; 19. Gages; 20. Actual dimensions (finding); 21. Guide values; 22. Guide rail switch opening; 23. Frog switch opening; 24. Defects found (according to Section 2 of the Test Booklet for Switches), replacement parts required; 25. Defects removed name/date; 26. Proper condition of the switch is certified: Name, Title, Date; 27. 1) Cross out if not applicable; 2) The Sh and Sh_Z dimensions are to be measured 200 mm behind the frog; 3) Impermissible deviations are to be underlined in red; 4) gage measurements $S_1 - S_8$ of 10 tie spacings (X) each, with appropriate positioning of the frogs in the sidings L and R Kr 60-1:14 S_1-S_6;Kr60-1:18.5 S_1-S_8;Kr60-$\frac{1200}{oo}$ -1:11.5 S_1-S_5 5) If the L and L_1 dimensions have a plus deviation, the associated H-dimensions may not have any minus deviations; 6. If the L and L_1 dimensions have a minus deviation, the R-dimensions may not have any minus deviation.

Position of Measurement Points

in the Area of Simple Frogs



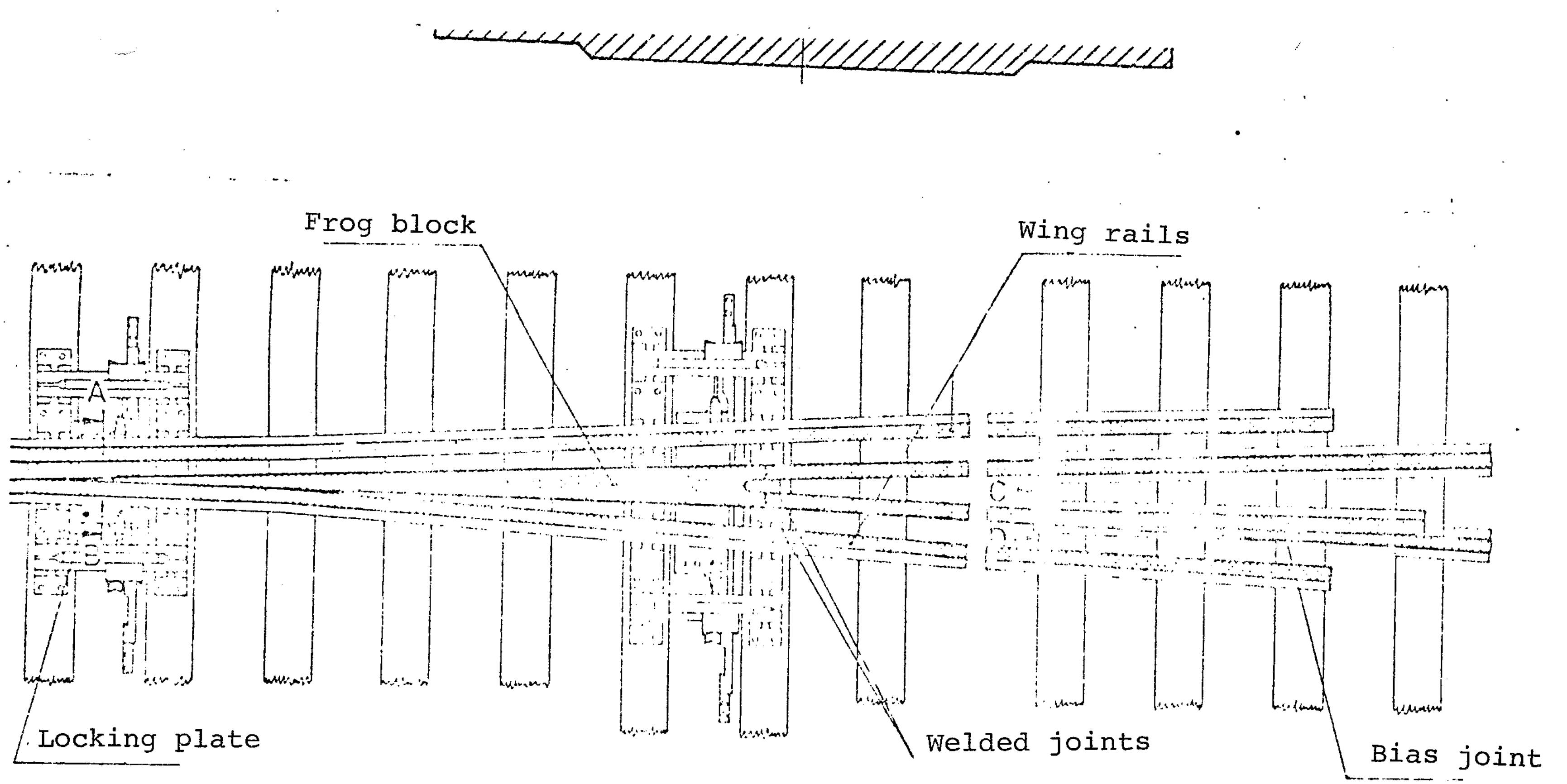
Simple Frogs	o	p	$q=q_z$	r	$s=s_z$	o_z	p_z	r_z
EH 60 - 300 - 1:9	878	522	150	372	320	860	540	390
EH 60 - 300 - 1:14	1022	778	150	628	500	1016	734	534
EH 60 - 500 - 1:12	1129	271	150	121	450	1117	283	133
EH 60 - 500 - 1:14	1022	778	150	623	500	1016	784	634
EH 60 - 760 - 1:14	1260	1040	150	890	560	1251	1049	899
EH 60 -1200 - 1:18,5	1628	672	150	522	790	1621	679	529
EH 60 - 1:14	1022	778	150	628	500	1022	778	628
EH 60 - 1:18,5	1282	1018	150	868	790	1282	1018	868
EH 60-1:11,515 / 1:9,631	1128	272	116	156	381	1128	272	122
EH 60-1:11,515 / 1:15,047	1143	798	150	648	491	1143	798	648

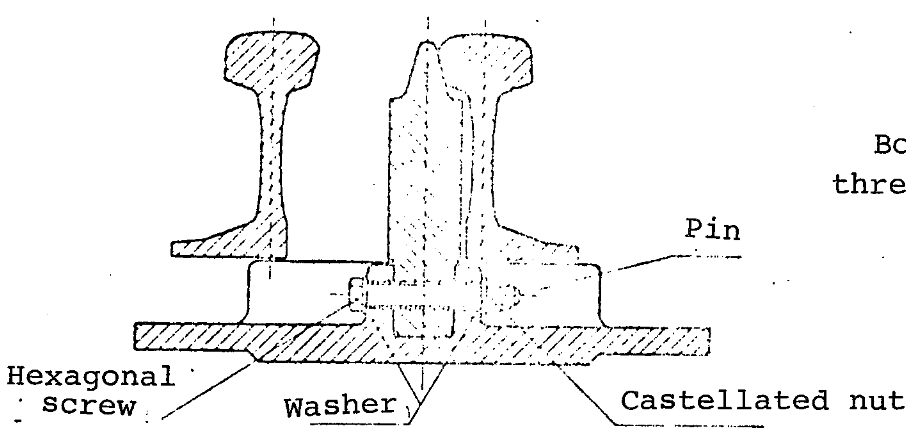


Section A - B

Screw connection of the locking plate

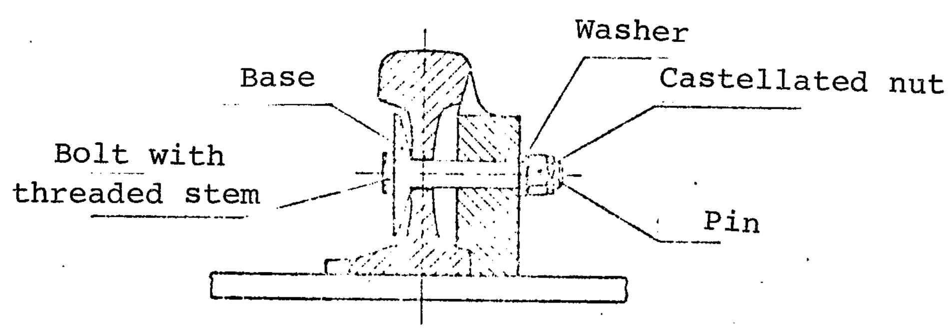
Elastic Frog





Section A - B

Screw connection of the locking plate



Section C - D

Screw connection at the bias joint

APPENDIX 3.1

SOURCE: Eisenbahn Ingenieur (Railroad Engineer) 25 (1974) No. 5 pp 169-174

TITLE: Track Stability at High Speeds*

AUTHOR: Prof Dr. Eng. J. Eisenmann

Basic Principles

The safety of the track panel can be impaired by various causes, which are briefly listed below (8):

- Internal forces,
- Internal forces and external loads, which cause a reduction of the track lateral strength in the neighborhood of the load contact area,
- External loads at the load contact point.

RAIL CRUSHING, BASED ON INTERNAL LONGITUDINAL FORCES CAUSED BY HIGH TEMPERATURE

In combination with the following triggering circumstances:

a) Excessive alignment defects, caused by
   - poor subsoil
   - constraint points in the track (switches, grade crossings),
   - defectively superelevated curves (excessive centrifugal force toward the high rail and excessive descending force** towards the low rail)
   - transition from a refurbished to a non-refurbished track, especially in the area of constraint points, such as switches, bridges, and grade crossings
   - construction defects

* Lecture at the 1974 Track Conference of the VEDI on March 21, 1974, in Frankfurt (M)

** German: Hangabtrieb

b) Local stress peaks in the rail, caused by

- failure to adjust the joint openings in a rail with joint openings,
- too low a neutral temperature with continuously welded rails
- too low a laying temperature relative to the neutral temperature, in the case of rails with a length greater than 120 m, and the omission of stress neutralization before welding,
- an unfavorable change of the stress condition with continuous rails in curved areas (example: severe temperature change during or immediately after track refurbishing, which leads to a reduction of the arc; after operational influences begin to take their effect, and as a result of the associated increase of track lateral strength, this deformation does not readjust evenly),
- stress build-up at the switch-point area of switches, caused by the joining of two rail sections into one; transition from UIC 60 to S 49 without intermediate rail S 54,
- braking forces in the area of lines on a falling grade, especially before constraint points (switches).

c) Reduced track lateral strength, caused by track refurbishing, especially when machine compaction of the ballast is omitted, also caused by poor ballasting of the ties. As can be seen from the table and from Figure 1, the track lateral strength falls severely after track refurbishing. Under the subsequent action of operational influences, it rises again. After 150,000 to 250,000 Lt, it reaches nearly its original value. A favorable result is obtained by using tie spacing and front head compactors, corresponding to an operational effect of about 60,000 Lt. Newer technologies also promise a better result with track refurbishing.

Schwellenart Schwellenabstand 3	vor 1	nach 2	nach	nach	nach	nach
			Durcharbeitung mit Gleisstopfmaschinen 7			
			mit Vorkopfverdichtern 8	mit Schwellenfachverdichtern 9	mit Schwellenfachverdichtern und Vorkopfverdichtern 10	[illegible] Betriebsbelastung [illegible] 11
4 Holzschwellengleis Form I [illegible] cm	13-95 iM [illegible]	27-6[illegible] iM 45	keine Besserung der Werte 12	42-55 iM [illegible]	6[illegible] [illegible]	53-6[illegible] iM 60
5 Betonschwellengleis B 58 63 cm	2-15 iM 13,5	65-8[illegible] iM 72	74			iM [illegible] (nach 3[illegible])
6 Betonschwellengleis B 70 60 cm	15	70-[illegible] iM [illegible]	670			

Gleishebung bei der Durcharbeitung 1-3cm, Schotterüberstand 40cm 13

Table: Track lateral strength at 2 mm displacement in kp/cm

1. Before; 2. After; 3. Type of tie and inter-tie space; 4. Track with wooden ties Form I, 63 cm; 5. Track with concrete ties B 58, 63 cm; 6. Track with concrete ties B 70, 60 cm; 7. Refurbishing with tamping machines; 8. With front head compactors; 9. With inter-tie space compactors; 10. With inter-tie space compactors and front head compactors; 11. (illegible) the action of operational influences 60,000 - 80,000 Lt, with and without compacting devices; 12. No improvement in values; 13. Track lift-off during refurbishing 1-3 cm, ballast overhang 50 cm

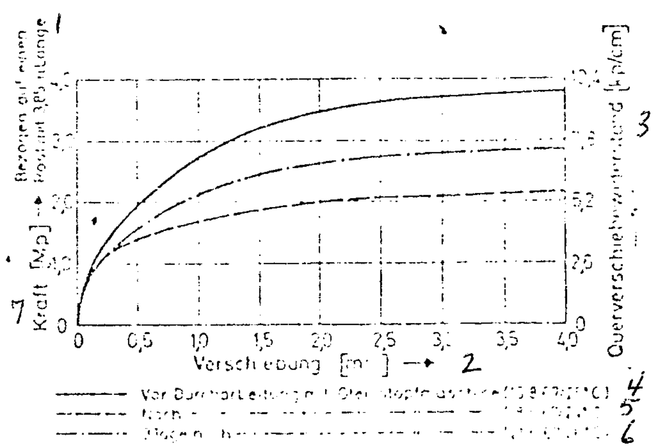


Figure 1: Track lateral resistance for a track with wooden ties - results of measurement

1. Relative to a panel with length 3.85 m; 2. Displacement (mm); 3. Track lateral strength (kp/cm); 4. Before refurbishing with tamping machine (16 8 67/25°C); 5. After refurbishing with tamping machine (18 8 67/24°C); 6. 3 days after refurbishing with tamping machine (28 8 67/26°C); 7. Force (Mp)

The type of tie also has a large effect. With the prestressed concrete tie B 70, the track lateral strength after track refurbishing lies in the same order of magnitude as in the case of a wooden tie of Form I before track refurbishing, corresponding to its undisturbed state. The B 70 tie has a track lateral strength that is 70% larger than that of the wooden tie. The reason for this is its greater weight and greater face area.

d) Reduced panel rigidity, caused by a rail fastening that has become loose or does not retain its tension. Under contract with the Federal Railway Central Office in Munich, experiments have recently been performed on a track panel with a tie spacing of 63 cm. These experiments have shown the following result: With a K-track on wooden ties, with a tie-plate Rph 6 or Rph 1, the so-called equivalent moment of inertia of the UIC 60 and S 49 is 3.5 or respectively 4.0 times the horizontal moment of inertia of the two rails, when the tie screws are half tightened and a vibration is applied. This corresponds to conditions prevailing with a loosened rail fastening. In the case of concrete ties, and with only a 110 mm long Rpb 1, one obtains 3.0 times the value as with the S 49. This shows the effect of the contact surface between the rail and the base plate. By half tightening the tie screws, the drop of tensile force of the tie screws, resulting from operational influences, is taken into account. This must be based on the gradual crushing of the poplar wood pad and progressive spring characteristic of the double spring washer Fe 6, which is used to hold the tension. When tightening clips are used instead of the clamping plate and the spring washer, a somewhat smaller value is obtained, despite full tightening. This is caused by the fact that, in comparison to a clamping plate, the connection between the rail base and the base plate is no longer so rigid. The contact pressure per rail support point should here be greater than 1800 to 2000 kp, if at all possible. In the case of concrete ties with W-fastenings and a plastic pad, and under otherwise similar experimental conditions, and using the UIC 60, one obtains 2.1 times the value of the horizontal moment of inertia of the two rails. Of course, it must here be taken into account that this rail fastening holds its tension extremely well. Consequently, the equivalent moment of inertia declines only little even after extended operating effects. In the case of K-track, the tie screws must be retightened at regular intervals, in order to guarantee the prescribed value. At the same time, damaged poplar wood pads and spring washers must be renewed.

CLASSICAL TRACK BUCKLING DURING THE SUMMER AS A CONSEQUENCE OF INTERNAL LONGITUDINAL FORCES UNDER THE ACTION OF OPERATING INFLUENCES

Classical track buckling is triggered in the area of the so-called lift-off wave. The peak value of the lift-off wave is located at a distance of $3 \cdot L$ from the axle, where L corresponds to the basic value of the long-tie track[1]. The track lateral strength is reduced in this area. This reduction is caused by the small lift-off of the ties. The lift-off of the ties can be determined by the computational method of Zimmermann (1, 2). The result of a measurement is shown in Figure 2. The lift-off is especially conspicuous with an axle distance corresponding to 6·L. The reason for this is that the bending lines are here superposed in an especially unfavorable manner. A further reduction of the track lateral strength is caused by vibration, which is also present, especially at high speeds.

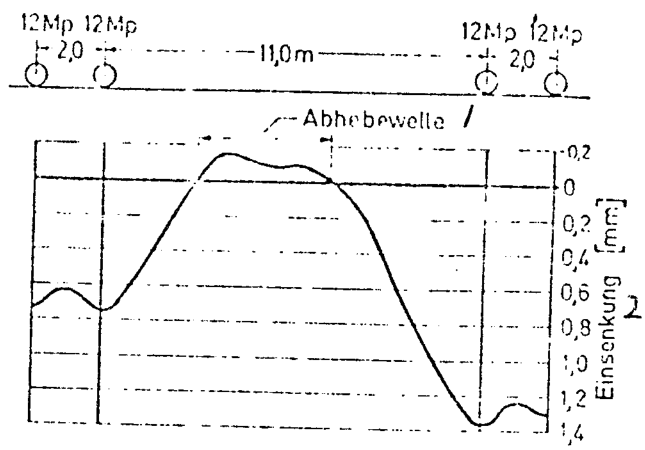


Figure 2: Vertical bending line of the rail with a track having wooden ties - results of measurement
1. Lift-off wave; 2. Subsidence (mm)

---

[1]Remark

$$L = \sqrt[4]{\frac{8 \cdot E \cdot J \cdot a}{C \cdot F}} \; [cm]$$

E = Elastic modulus of the rail in kp/cm^2
I = Moment of inertia of the rail in cm^4
a = Tie spacing in cm
C = Ballast figure in kp/cm^3
F = Tie contact area in cm^2
The base value L lies in the range between 70 and 150 cm, depending on the rail profile, tie spaces, tie form, and ballast figure.

Measurements were performed on operational track, under contract with the Federal Railway Central Office in Munich and in collaboration with Professor Birmann. These measurements show, that in the area of the lift-off wave, the track lateral strength is reduced by 20 to 40% at high rail temperatures. The effect of vibrations can be estimated at 10 to 20%.

According to experience, when a train is rolling over the track, track buckling occurs not underneath the first axles but underneath the last ones. This creates a suspicion that the multiple action of axles, combined with repeated lift-off of the track panel, causes a build-up effect both before and behind the axle. This build-up effect then leads to the classical track buckling. Track buckling is favored under the circumstances which we have explained.

The appearance of classical track buckling is characterized by a wave length of up to 15 m with amplitude up to 100 cm (Figure 3).

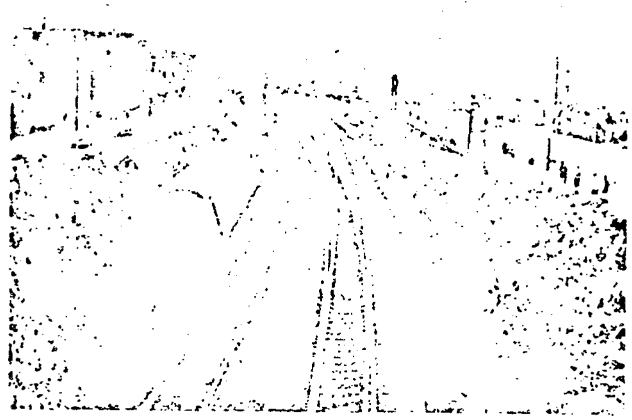


Figure 3: Classic track buckling, starting from a switch

TRACK DISPLACEMENT CAUSED BY LARGE LATERAL FORCES

The action of large lateral forces can also cause an impairment of the safety of the track panel. Such forces trigger track displacement. Track displacements are also favored by the compressive forces present in the rail as well as by alignment defects.

The appearance of large lateral forces is favored by the presence of periodic track quality deficiencies. Depending on the wave length of the track quality deficiency as well as on the running speed of the train, this can cause increased horizontal vehicle vibrations. At first, these vehicle vibrations start from small track quality deficiencies. They then lead to a gradual increase of the periodic deficiency, combined with increased vehicle vibration in subsequent trains. As the track quality worsens, there is danger of more severe track displacement, combined with derailment. Dynamic track displacement can be favored by the above-mentioned local circumstances. The appearance of such displacements of the track panel is characterized by wave lengths longer than 20 m.

The lateral displacement of the track panel has up to now been studied only under the effect of static load. According to French experiments (3), the sum of the guide forces should satisfy the following conditions, in order to avoid lateral displacement of the track panel:

$$H \leq \alpha \cdot \left(1 + \frac{2Q}{3}\right)$$, here:
[illegible]

Here, 2Q = axle load,

χ = coefficient, depending on the track construction and track maintenance; for French conditions (minimum value for shoveled track with wooden ties and 46 kg rails), the value χ = 0.85 is recommended. According to experiments performed by the Institute for the Construction of Provincial Traffic Routes of Munich Technical University, this value can also be used for German conditions, under circumstances immediately following properly performed track maintenance.

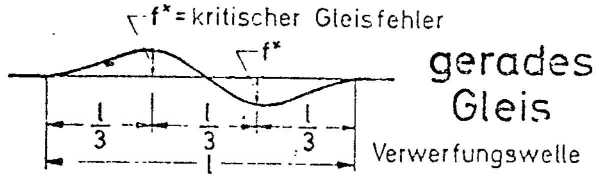
The effect of dynamic forces, on the other hand, is less well known, especially their axle time. Furthermore, the interaction between track quality and vehicle construction type on the one hand and running speed on the other hand have up to now been researched only little. However, the relationships are quite well known qualitatively. The subsequent explanations will not discuss this lateral displacement of the track panel. It should here be noted that the resulting track defects can lead to classic track buckling at high rail temperatures.

Theoretical Foundations of Track Stability

Spontaneous, horizontal track buckling is based on internal longitudinal forces resulting from higher temperatures. Such track buckling can be evaluated by the computational method developed four decades ago by Professor Meier (4, 5). Despite the assumptions on which this method is based, the computational result provides sufficient information for practical purposes. The good agreement with large scale experiments, performed by Professor Raab in Karlsruhe, should be especially emphasized (5, 6). It should further be noted that vertical track buckling does not occur with the high weight of ballasted panels.

A summary presentation of the computational rule is shown in Figure 4. A distinction is here made between a tangent track and a curved track. Different buckling waves are associated with each type. It should here be noted that the computational rule shown for curved tracks will yield excessively unfavorable results for curves with a radius of less than 500 to 700 m. The reason for this is that "curve breathing" would here become effective.

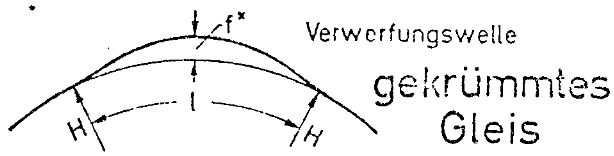
The critical temperature rise Δt is especially significant for the following considerations. This quantity provides information concerning the temperature rise of rails, beginning with which, for a prescribed track defect f, one must expect a transition to the labile state. In order to avoid track buckling, this value Δt must be larger than the actual temperature rise ΔT. The latter quantity is 35 to 50°C, taking into account the deviations in rail tightening that occur in practice. Additional forces are activated underneath the train - lateral forces, braking forces - as well as vibrations. As a result, on the basis of present experience and depending on vehicle speeds, the temperature increase of the rails must lie 10 to 50°C below the computationally determined critical temperature rise. With a safety margin of only 10°C, and taking into account local and operational contingencies, a slow run with strongly



$$l = 3 \cdot \pi \cdot \sqrt{\frac{2 \cdot E \cdot J}{P_0}} \; [cm] \; ; \; f^* = 8{,}7 \cdot w \cdot \frac{E \cdot J}{P_0^2} \; [cm]$$

Daraus die kritische Temperaturerhöhung

$$\Delta t = \sqrt{\frac{8{,}7 \cdot J \cdot w}{\alpha^2 F^2 E \cdot f}} \; [°C]$$



$$l = 2 \cdot \pi \cdot \sqrt{\frac{2 \cdot E \cdot J}{P_0}} \; [cm] \; ; \; f^* = (w - \frac{P_0}{H}) \cdot \frac{16 \, E \cdot J}{P_0^2} \; [cm]$$

Daraus die kritische Temperaturerhöhung

$$\Delta t = -\frac{8 \cdot J}{\alpha \cdot F \cdot H \cdot f} + \sqrt{\left(\frac{8 \cdot J}{\alpha \cdot F \cdot H \cdot f}\right)^2 + \frac{16 \cdot J \cdot w}{\alpha^2 F^2 \cdot E \cdot f}} \; [°C]$$

wobei

E = Elastizitätsmodul des Schienenstahls in kp/cm^2
α = Temperaturdehnzahl von Stahl
F = Fläche der beiden Schienen in cm^2
J = Ersatzträgheitsmoment des Gleisrostes in cm^4
H = Gleishalbmesser in cm
w = Querverschiebewiderstand in kp/cm
f = Angenommener Gleisfehler in cm (2,0 bis 2,5 cm)
P_0 = $\alpha \, \Delta t \cdot E \cdot F$ = kritische Gleisdruckkraft in kp

Figure 4: Computational rules for evaluating track stability at high rail temperature - method of Professor Meier

1. f* - critical track deficiency; 2. Tangent track; 3. Buckling wave; 4. Resulting critical temperature rise; 5. Buckling wave; 6. Curved track; 7. Resulting critical temperature rise

whereby

E = Elastic modulus of the rail steel in kp/cm^2

α = Heat expansion coefficient of steel

F = Area of both rails in cm^2

J = Equivalent moment of inertia of the track panel in cm^4

H = Curve radius in cm

w = Track lateral strength in kp/cm

f = Assumed track deficiency in cm (2.0 to 2.5 cm)

$P_0 = \alpha \cdot \Delta t \cdot E \cdot F$ = critical track compression force in kp reduced speed must be expected. With a speed of 160 to 200 km/h, the safety margin should be 40 to 50°C. The domain beyond 200 km/h still requires further research. The vibration intensity is here increased, and the resulting drop of track lateral strength is significant. This is especially true when the wheel set runs unstable, combined with increased guiding forces. On the basis of the present state of research, this can be counteracted by increasing the weight of the track panel and also by enlarging the contact and face area of the ties (7).

Besides the materials characteristics, the following are also components of the formulas cited in Figure 4:

- The initial track lateral strength w of the track panel. Depending on the track form and track condition, w lies between 4 and 15 kp per cm track length, under static conditions and with 2 mm displacement. Taking into account the lift-off wave as well as the vibrations present underneath the train, the track lateral strength is 33% less on the average. This must be the starting point when evaluating the track safety at higher speeds (dynamical value).

- The equivalent moment of inertia I. Depending on the tightening between the rail and the tie, and depending on the type of rail fastening, vibrations will cause this quantity to assume 1.0 to 4.0 times the value of the horizontal moment of inertia of the two rails.

- the rail cross section, which is 22% larger with the UIC 60 than with the S 49. Taking into account the dependence existing with tangent track, this change can be described by a factor 1.5, in comparison to the track lateral strength and equivalent moment of inertia.

- The track deficiency f. Depending on track conditions, and taking into account the error resulting from assuming a constant value for the track lateral strength and the equivalent moment of inertia, f lies in the order of magnitude of 1.5 to 2.5 cm. The smaller value can be used when the track is well maintained as well as for the more rigid rail profile UIC 60.

Evaluation of track stability consequent upon heating

TANGENT TRACK WITHOUT CONSTRAINT POINTS

Figure 5 shows the temperature rise Δt, which is critical for track buckling. With a tightened track panel, in its undisturbed state before track refurbishing, Δt provides a sufficient safety margin with respect to actual heating of the rails. This holds true both for the S 49 rails and the UIC 60 rails. It also holds for wooden ties of Form I and K-track with a relevant track lateral strength of 5.6 kp/cm on the average; also for concrete ties with W-fasteners, with a track lateral strength of 9.0 (10.0) kp/cm with the B 68 (B 70). Accordingly, even at high running speed, track safety is not endangered by track buckling.

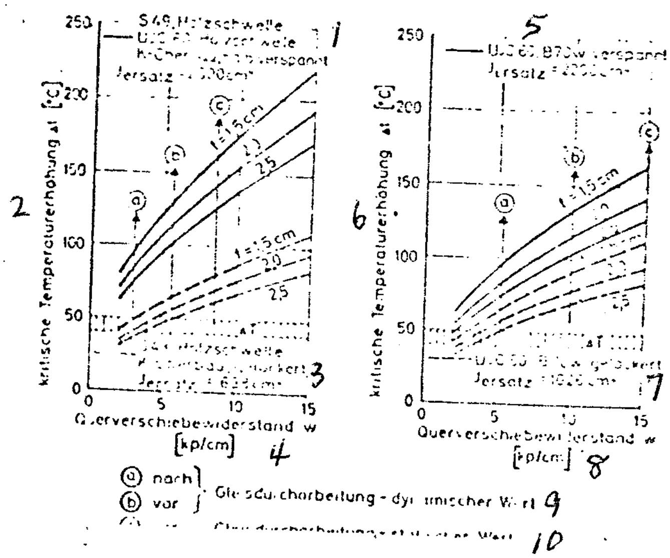


Figure 5:

1. S 49 wooden tie, UIC 60 wooden tie, K-track, half tightened, $J_{equivalent} = 2600\ cm^4$; 2. Critical temperature rise Δt (degrees C);
3. S 49 wooden track, K-track, loosened, $J_{equivalent} = 638\ cm^4$;
4. Track lateral strength W (kp/cm); 5. UIC 60, B 70 w, tightened, $J_{equivalent} = 2200\ cm^4$; 6. Critical temperature rise Δt (degrees C);
7. UIC 60, B 70 w, loosened, $J_{equivalent} = 1026\ cm^4$;
8. Track lateral strength w (kp/cm);
9. Before / After Track refurbishing - dynamical value
10. Before / After Track refurbishing - static value

After track refurbishing, the critical temperature rise, based on a track deficiency of 2.5 cm for the S 49 and 2.0 cm for the UIC 60, is somewhat more than 70°C for the S 49 and 80°C for the UIC 60. This holds true for a track panel with wooden ties and K-track, With a normative track lateral strength of 3.0 kp/cm (Figure 5, left graph). In the case of a track panel with concrete ties B 70 W, and with a normative track lateral strength of 5.3 kp/cm (right graph), one obtains $80^{\circ}C$ for the UIC 60, despite its somewhat smaller equivalent moment of inertia. At high rail temperatures, this requires stoppage of track work, in order to avoid crushing of the rails. Subsequently, at high rail temperatures, it also requires a short-term reduction of running speed, in view of the safety of railway operations. The following must here be taken into account: When the rail fastenings are not adequately tightened, there exists acute danger of track buckling in track panels with wooden ties. When in inter-tie space and front-head compactor is used, conditions are more favorable. This is especially significant with tracks having wooden ties. At the same time, this prevents a rapid worsening of track alignment.

Reference is further made to the fact that, for undisturbed track condition but a loosened rail fastening, a barely measurable safety margin exists, but only with a track panel having concrete ties. With good track alignment and a normative track deficience of 2.0 cm, the critical temperature rise is $80^{0}C$ for the B 70 W and UIC 50. By contrast, in the case of tracks with wooden ties, with a critical temperature rise of only 50 to $60^{\circ}C$, track safety is jeopardized. This indicates the significance of a well tightened rail fastening.

Furthermore, in view of the drop of track lateral strength associated with speeds in excess of 100 to 120 km/h, one should avoid the use of wooden ties of Form II, with only 14 to 15 cm height. At high speeds, this would require either the arrangement of safety caps or a widening of the ballast bed.

CURVED TRACK WITHOUT CONSTRAINT POINT

In the case of curved tracks, with curvatures having a radius greater than 700 to 800 m, a similar result is obtained as for tangent track. As the radius becomes smaller, the critical temperature difference decreases. This is the quality that is here relevant for track buckling. In this connection, the following should be noted for curved tracks with a radius smaller than 700 to 800 m:

- The permissible speed and consequently the vibration intensity decreases and
- the computational formula given in Figure 4 yields too small a value for the critical temperature rise. This is based on the actual "breathing" of the curve.

Free breathing of the curve leads to complete relief of the strain. By means of Figure 6, this can be estimated for a temperature difference of 40°C. For example, for an arc angle $\emptyset = 60^{\circ}$, and with a radius of 750 m, it can be seen that free breathing comes to barely 5 cm; for a radius of 300 m, free breathing is somewhat more than 30 cm. Based on practical experience, a curve breathing of 1 to 2 cm can be assumed. Consequently, up to a free breathing of 5 cm, the longitudinal forces relevant for track buckling are noticeably reduced. Such a free breathing of 5 cm corresponds to a radius less than 500 to 700 m, with an arc angle less than 60°. This matter still requires research by long-term measurements on operational track.

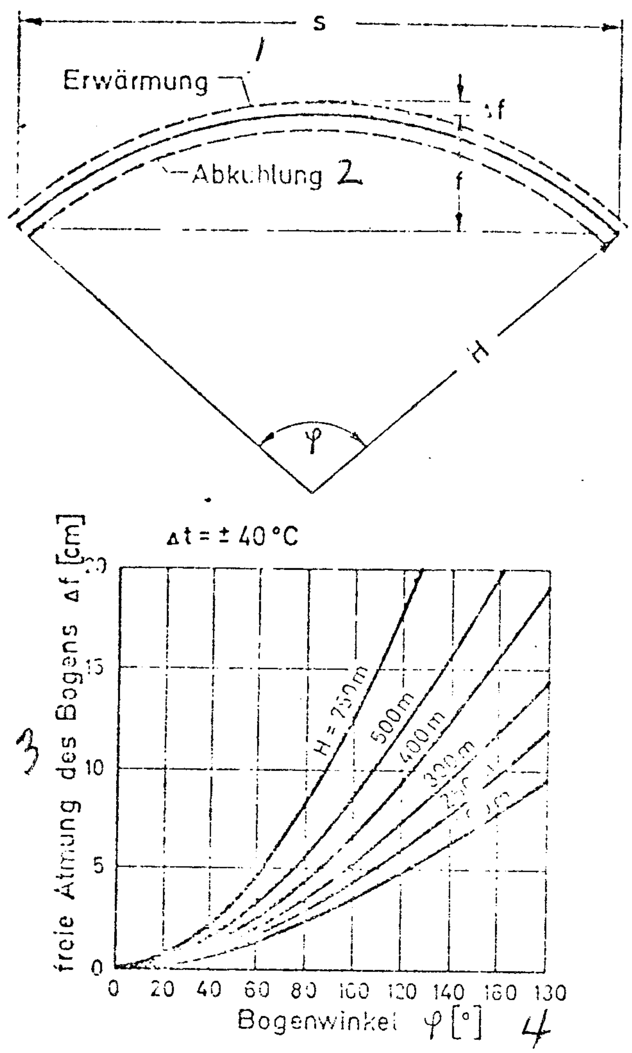


Figure 6: Free breathing of a curved track with a radius less than 750 m
1. Heating; 2. Cooling; 3. Free breathing of the curve Δf (cm);
4. Arc angle ϑ ($^{\circ}$)

UNEVEN RAIL HEATING

When the rails are heated unevenly, such as occurs when a shadow is cast by building works, embankments, or standing vehicles, the course of longitudinal forces in the rails is distrubed (Figure 7). The longitudinal force in the less heated rails is here reduced under the action of the longitudinal track resistance. This effect is combined with a longitudinal motion of the rail. This can cause the ties to assume a slightly slanted position. Corresponding to the longitudinal motion, the slant is greatest in the middle of the transition region. Accordingly, a track deformation will appear, and this deformation will also extend to the neighboring, undisturbed section. Operational influences favor the generation of track deficiency, with unfavorable effects on track stability. This still requires research by temperature and deformation measurements on operating track.

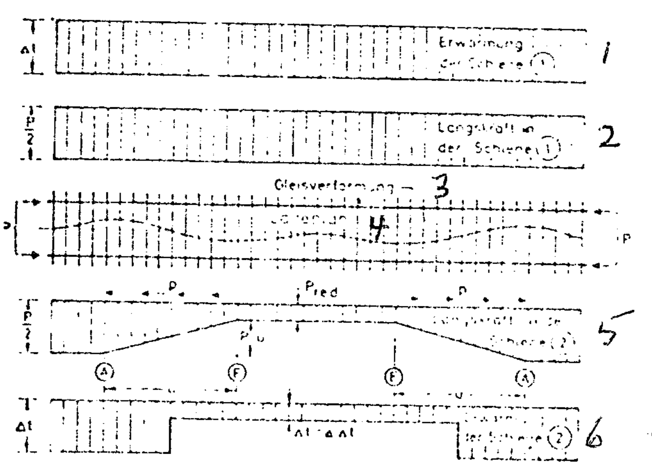


Figure 7: Course of the longitudinal force and track deformation with uneven rail heating

1. Rail heating; 2. Longitudinal force in the rail; 3. Track deformation; 4. Position diagram; 5. Longitudinal force in the rail; 6. Rail heating

FIXED POINTS

Fixed points also have a disadvantageous effect on track stability. Such fixed points exist, for example, with groups of scissor crossings, with grade crossings, or with bridges having rigidly laid tracks without a rail expansion joint. Especially with curved track sections, the "breathing" of the curve is thereby hindered. As a consequence, reduction of longitudinal forces is hindered, with the associated creation of a track deficiency.

Alignment of curved tracks at too low a temperature has a similar effect, especially in the case of track sections between fixed points, e.g. a group of scissor crossings. Track quality can here be changed relative to its condition upon welding. This can lead to an increase of longitudinal forces during heating, corresponding to greater heating of the rails. Such a build-up of forces can be avoided by separating and untensioning the rails.

SWITCHING AREAS

With strong heating, the full longitudinal force of a track amounts to 150 to 180 Mp for the UIC 60. In the case of switches, it must be taken into account that this longitudinal force must be transferred to the ballast bed by activating the longitudinal track resistance. This is connected with a longitudinal motion of the track, which assumes its peak value in the middle of the transition area. If the longitudinal track resistance of the track panel is 10 to 15 kp/cm on the average, and if the longitudinal force is 150 Mp, this yields a transition length ℓ of 100 to 150 m.

Consequently, this length is considerably longer than a switch. The dissipation of the longitudinal force accordingly extends beyond both sides of a switch.

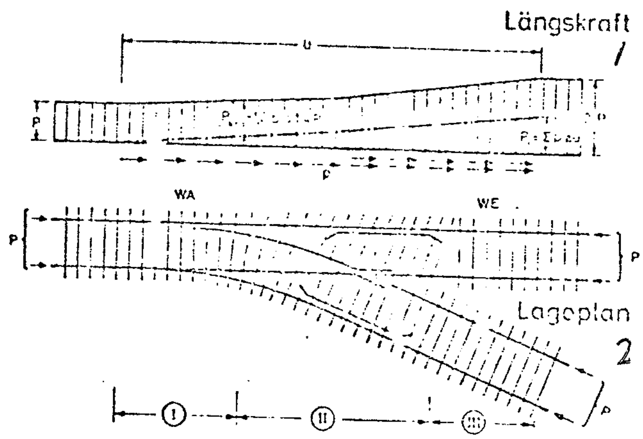


Figure 8: Course of longitudinal force in the switch area
1. Longitudinal force; 2. Position diagram

As shown in Figure 8, the transition area can be subdivided into 3 component areas, from the perspective of track stability. Section II here is the safest, because of its large track lateral strength and large panel rigidity. Section I extends from the beginning of the transitional length as far as the tongue area of the switch. In this section, track stability can be evaluated by comparing the critical temperature rise Δt (Figure 5) with the increased temperature rise of the rails. This latter temperature rise corresponds to the increased longitudinal force in this section, which is given by $P_{crit} = 1.2$ to $1.4 \cdot P$. The smaller value here holds for a 65 m long switch and the larger one for a 25 m long switch. As a result, safety is reduced, and this can lead to critical conditions especially after track refurbishing. Circumstances in Section III are also unfavorable. In this section, the inside non-continuous rails are unstressed, and this is combined with a small longitudinal motion. As is the case with uneven heating of the rails, this entails larger track quality deficiencies, which in turn cause a drop of the critical temperature rise.

In view of the above, greater precautionary measures are required during maintenance work on switches. On days with elevated rail temperatures, work should not be performed on switches. After work has been completed, the switches should be traversed slowly for a limited period of time, corresponding to the expected rail temperature. In the case of lines with a speed above 140 to 160 km/h, the track lateral strength should be increased in the endangered areas I and III. This can be achieved by widening the ballast bed and by arranging safety caps. Among the possible measures, compaction of the ballast before the tie heads, by means of asphalt or suitable plastics, should also be included. These supplementary constructive measures require only relatively little expenditure.

In the case of switches with a rail expansion joint in the rail running to the frog, care should be taken to increase the panel rigidity and to combine this with an increased track lateral strength. The purpose of these measures is to avoid track quality deficiencies, caused by the uneven longitudinal motion of the rail. Increased panel rigidity can be achieved by arranging the expansion joint on ties which extend over both rail sections. This is in fact the case in Section II.

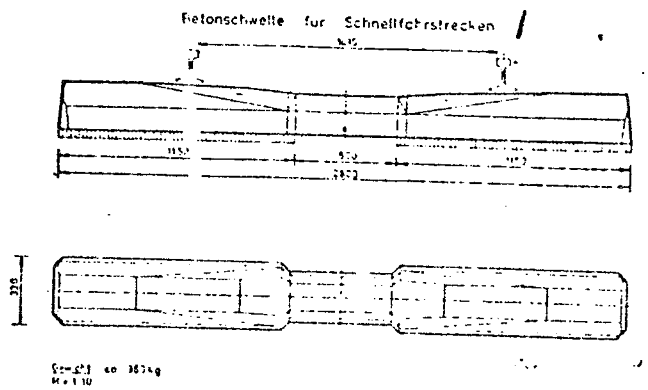


Figure 9: Proposal for prestressed concrete tie for high speed lines
1. Concrete tie for high speed lines

Summary and Prospects

The track stability of a track at high temperatures is guaranteed even in the range of high speeds, if

- a tension-retaining rail fastening is used, and care is taken to effect proper tightening during installation and during running track maintenance,
- the track panel is ballasted in accord with track directives,
- the track lateral strength is increased at speeds greater than 160 km/h in the area of switches, along a length of at least 150 m. This is done by additional constructive measures - widened ballast bed, safety caps, reinforced shoulders.
- alighment errors are quickly corrected. This requires continuous monitoring of the track.
- track repair is not performed on days when the rail temperature is high, and the speed is reduced for a limited time, following track renewal or track refurbishing; special attention should here be paid to the switch area which extends along a length of 150 m.

- Alignment work on welded curved track is not performed at too low a rail temperature, especially in the area of fixed points in the track.

These requirements are taken into account in the relevant track directives.

Track stability at speeds greater than 200 km/h still requires research. Developments will here lead to a heavier track panel with a larger support and face area of the ties. A prestressed concrete tie has been proposed with an extended length of 2.8 m and an extended width of 33 cm, and a weight of 380 kg, as compared to the present 280 kg. This tie is shown in Figure 9. As a result, an inter-tie space of 63 cm is possible, instead of the present 60 cm. Consequently, the resulting extra cost is minimal. When this tie is used, as opposed to the B 70, the track lateral strength is increased by about 40 to 50%. The quasi-static and dynamic stress of the ballast is also less (7). Consequently, use of the heavy tie track also appears possible for very high speed lines. Adquate track stability and a guaranteed precise track position and track alignment over a longer period of time are preconditions for this.

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DESCRIPTION OF THE EXPERIMENT TO DETERMINE THE EQUIVALENT MOMENT OF INERTIA OF A TRACK PANEL

A track panel, consisting of 17 ties with a tie spacing of 63 cm and S49, S54 or UIC60 rails, was constructed on an asphalt plate already present in the experimental terrain. At the fourth and fourteenth tie, that is at a distance of 6.30 m, fixed supports were placed on both sides of the track panels before the tie heads. The horizontal force was continuously applied to the ninth tie, through another fixed support, by means of an hydraulic Lucas press. It was continuously measured electrically by means of a calibrated pressure gauge. Through a carrier frequency measurement amplifier, it was conducted to an x-y recorder. The associated displacement f in the center of the track panel is measured by means of an inductive displacement sensor, is conducted through a measurement amplifier, and is recorded in an x-y recorder.

While the track panel is being displaced horizontally, it is desirable to keep friction as low as possible. For this purpose, every second tie is mounted on two plastic rollers below the rail fastening. Since the rollers move on plates of the same material, where these plates are placed on the asphalt surface or respectively under the ties, a negligibly small rolling resistance results in the amount of about 40 kp.

Depending on its rigidity, the track panel is loaded with 3 to 4.5 Mp in the (+)-direction. Load relief brings with it a permanent deformation. After the load is removed, the track panel is displaced into the opposite (-)-direction. An external vibrator is screwed onto each rail between the fifth and sixth and between the ninth and tenth tie. This creates an off-balance of 160 kp at a frequency of 50 Hertz, so as to simulate the effect of the dynamics of a running train.

Measurements were made on the ties, at 50 to 200 Hertz, and at vibrational speeds of 100 to 120 dB_v relative to $v_0 = 5 \times 10^{-6}$ cm/sec.

The behavior of the track panel under a horizontal force is characterized by a nonlinear course in the load range, a load relief range which differs from this, and a permanent deformation (elastic hysteresis).

The bending of a single span support under a single load is:

$$f = \frac{P \times \ell^3}{48 \times E \times I}$$

The cant α in the force-displacement diagram is the measure for the panel rigidity:

$$\frac{P}{f} = tg\alpha = \frac{48 \times E \times I}{\ell^3}$$

If a single span support with an elastic modulus $E = 2.1 \times 10^6$ kp/cm^2 is considered, and if the distance between the fixed supports in the experiment is 630 cm, the equivalent moment of inertia of the track panel in the horizontal direction is proportional to the cant α or to tgα (Mp/cm):

$$I_E \quad (cm^4) \quad = \quad 2.480 \times tg\alpha$$

Since the curve in the force-displacement diagram is non-linear, but the equivalent moment of inertia rather decreases with increasing displacement, the equivalent moment of inertia, shown in Figure 1, was evaluated for a displacement f = 15-20 mm.

Although the equivalent moment of inertia is larger for small displacements, the equivalent moment of inertia is taken for the above displacements, in order to be on the safe side. However, the starting point is the secant modulus and not the tangent modulus, since the track panel thereby performs the same work as an equivalent beam with rigidity $E \times I_E$.

It was determined that the equivalent moment of inertia depends not only on the magnitude of bending f, resulting from the action of the force, but also on the horizontal predeformation Δf of the track panel before the force is applied.

With a maximum force of 3,000 kp, the permanent deformation is 20-30 mm. Since the center of the permanent deformations under maximum force agrees well with the straight position, it is defined as the $\Delta f = 0$. This eliminates errors caused by the self-stresses in the system. Δf greater than 0 therefore represents a predeformation. $\Delta f = + 10$ mm, for example, means that the track panel must first be displaced by $\Delta f = 10$ mm until it reaches its straight position.

A force-displacement diagram is determined for each individual track construction. It is evaluated in accord with Figure 1. First, the predeformation $\Delta f = 0$ is determined, and then the deformation existing for each loading experiment, as well as the secant modulus for a displacement of $f = 5$, 10 and 20 mm.

The following can be stated:

1. With displacements $f > 15$ mm, the base of the rail strikes the rib and then contributes to force transmission.
2. The equivalent moment can depend on the direction of bending. Averaging the moments of inertia in both directions eliminates the effect of self-stresses in the rails.
3. When the moments of inertia of the first load were significantly higher than the following ones, they were omitted. It could be determined that the moment of inertia does not depend on the number of loads, but rather that the curve of the force-displacement diagram nearly always exhibits confluence with the hysteresis associated with maximum deformation.

As already happened in earlier experiments, the values of the B70 tie with the rail fastening W had more severe scatter than those with other fastenings. For this reason, the straight lines were for this case calculated as linear regressions. With the small slope of the straight lines, this is equivalent to taking an average.

The equivalent moments of inertia I_E are compiled in Figure 2. They include various types of track, at half the contact pressure, with and without the action of the vibrator, for a deformation $\Delta f = + 5$ and $+ 10$ mm, and a displacement of $f = 5$ and 10 mm. For the B70 tie with rail fastening W, the values for the full tightening force are plotted. As a comparison shows, the difference of the equivalent moment of inertia for $\Delta f = +5$ mm at $f = 10$ mm, as well as $\Delta f = + 10$ mm at $f = 5$ mm, is only quite small, since the equivalent moments of inertia decrease about equally in this range, with increasing deformation and increasing displacement. The arithmetic average of these two values is plotted in Figure 3. The deformation of the track panel during alignment can be described by a $\Delta f = + 5$ and $+ 10$ mm, corresponding to the track deficiency. In view of this deformation, an evaluation of the safety of the track starts from this range. The displacement f here corresponds to the return deformation of the track panel into the defective starting position, which occurs subsequent to alignment.

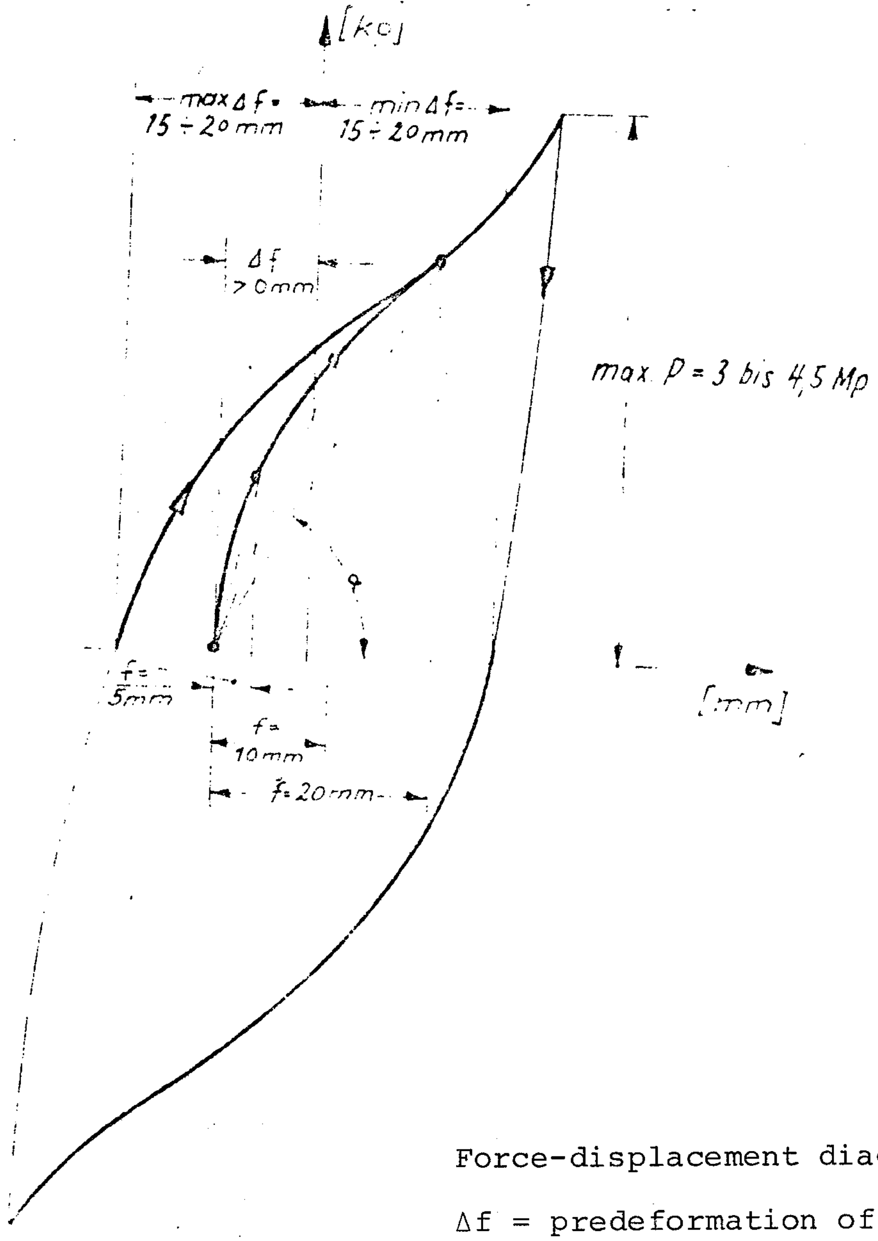
In order to be able to compare the moments of inertia, the following conditions were used for K-track: the conditions "half the tightening force" of the tie screws, because of the progressive spring characteristic, and the condition "full tightening force" for the elastic clip track, because of the linear spring characteristic and the large tightening path. Half the tightening force of the tie screws is about 1,500 kp, which corresponds to a contact force of 1,500 kp per support point or a torque of 10 kpm per screw.

The individual data of Figure 2 cannot always be directly compared. For instance, the increase of the moment of inertia of Number 1 to 2 is based solely on the increased bending strength of the rails. With the UIC 60 rail (Number 3), not only on the bending strength was increased, but also the friction surface F = width of rail base x width of rib plate was increased from 125 x 150 mm = 188 cm^2 to F = 150 x 160 mm = 240 cm^2.

Even though the bending strength of the concrete ties (Numbers 4 and 5) is higher, the moment of inertia of the track panel with the B58 tie is less than with the beech wood tie (Numbers 1 and 2). The friction surface therefore exerts an influence here, since it transmits the torque to the tie. With the B58 tie, this amounts to about F = 125 x 110 mm = 137 cm^2, instead of 188 cm^2 for the wooden tie.

Appendix 3.2

Figure 1



Force-displacement diagram

Δf = predeformation of the track panel (mm)

f = displacement of the track panel (mm)

APPENDIX 4.1

Revision of Technical Delivery Conditions UIC 860-V

Text proposed by ORE Committee D 144:

TECHNICAL DELIVERY CONDITIONS FOR RAILS

EXCERPTS

11. PRODUCTION METHODS

The methods for steel and rail production remain open to the manufacturer. However, upon wish of the buyer, he must make their type and principal characteristics known in his offer. He may not change these, without notifying the buyer before implementing the change.

During the entire production process, the manufacturer will use the best rules of technology, so that the rails will correspond to the present delivery conditions.

Ingot-cast as well as continuously cast material may be used to manufacture the rails, but the latter process may be used only upon approval of the buyer.

Precautionary measures must be taken so that hydrogen does not cause flaking in steel types sensitive to this.

Upon the buyer's request, the manufacturer will inform the buyer what steps are being taken to avoid such flaking in rails of these steel types.

14. QUALITY OF CONSTRUCTION

The rails must be free from all damaging defects, i.e. such defects as unfavorably influence the operating behavior of a rail. Among such defects belong, among others, cracks of all types, peelings, open bubbles, and material deficiencies.

Depending on the manufacturing process and steel quality, freedom from damaging internal defects is to be secured by suitable continuous non-destructive testing, e.g. by ultrasound tests. This test is performed by the manufacturer under his own responsibility. Furthermore, with agreement of the buyer, this test may permit a specific application of this rail.

Surface defects may be investigated by the buyer's representative by means of a tool. He will decide whether the defect is significant for the usefulness of the rail.

In the case of tests in the storage depot, every rail, in which a bubble is visible with the naked eye, will be rejected. Every operational process, both under cold and hot conditions, to conceal a defect is expressly prohibited.

15. MANUFACTURE

Cold-straightening must be effected step by step and without joints. When using roller straightening machines, the rail may traverse them only once in each direction, without special approval of the buyer; the identifying marks must here be protected before the action of the straightening rollers.

The rails should be subdivided to their finished lengths by milling or sawing. Burrs must be removed, without a noticeable bias resulting in the edges.

Holes must be produced by drilling. They must be perfectly cylindrical and must have smooth walls without burrs. Their edges must be slightly beveled.

16. PERMISSIBLE DEVIATIONS

16.1 Profile and Ends

		Abmessungen in mm	Toleranzen	Bemerkungen	
1. Höhe H der Schiene (1)		H < 165 165 ≤ H < 180 180 ≤ H < 190	+ 0,5 - 0,5 + 0,6 - 0,6 + 0,7 - 0,7	Profil UIC 54 Profil UIC 60 Profil UIC 71	Bei Messungen zwischen den Schienenenden darf sich das angegebene Toleranzfeld um -0,5 mm verschieben.
2. Fußbreite L (1)		138 ≤ L < 150 150 ≤ L < 160 160 ≤ L < 170	+ 1,0 - 1,0 + 1,0 - 1,1 + 1,2 - 1,3	Profil UIC 54 Profil UIC 60 Profil UIC 71	dto. (mit + 0,5 mm)
3. Nennbreite des Kopfes (2)		L < 72 72 ≤ L < 74 74 ≤ L	+ 0,5 - 0,5 + 0,5 - 0,5 + 0,7 - 0,7	Profil UIC 54 Profil UIC 60 Profil UIC 71	dto. (mit + 0,1 mm)
4. Asymmetrie des Profils (3)		L < 150 150 ≤ L < 160 160 ≤ L < 170	+ 1,2 - 1,2 + 1,5 - 1,5 + 1,7 - 1,7	Profil UIC 54 Profil UIC 60 Profil UIC 71	(4)
5. Stegdicke (5)			+ 1,0 - 0,5	Alle Profile	
6. Neigung der Laschenanlageflächen			3,6 %	Alle Profile	
7. Höhe der Laschenanlageflächen			gleiche Toleranz wie H	Alle Profile	
8. Rechtwinkligkeit der Enden	in der Schienenachse 10	H ≤ 180 180 < H ≤ 190	+ 0,6 - 0,6 + 0,7 - 0,7	Profile UIC 54 und UIC 60 Profil UIC 71	
	parallel zur Fußbasis 11	L ≤ 150 150 < L ≤ 160 160 < L ≤ 170	+ 0,5 - 0,5 + 0,6 - 0,6 + 0,7 - 0,7	Profil UIC 54 Profil UIC 60 Profil UIC 71	
9. Andere Abmessungen		wie H 13	gleiche Toleranz wie H 14		

1 height H of the rail (1)

2 width of base L (1)

3 rated width of the head (2)

4 asymmetry of the profile (3)

5 web thickness (5)

6 cant of the fish plate attachment surfaces

7 height of the fish plate attachment surfaces

8 perpendicularity of the ends

9 other dimensions

10 in the rail axis

11 parallel to the base

12 For measurements between rail ends, the specified tolerance field may shift by -0.5 mm

13 like H

14 same tolerance as H

(1) The permissible deviations of the heavy rails UIC 71 are specified provisionally. They will be specified definitively only after the first production run of these rails.

(2) Measured 14 mm below the running surface, namely in the transition region between the rounding of the running surface and the lateral surfaces of the head.

(3) Must be checked with the UIC gage.

(4) Upon request of a buyer, this additional deviation can be reduced to ± 1.2 mm for rails of steel quality 70.

(5) Measured at the level of minimum thickness.

(6) These values refer to measurements at the rail ends.

16.2 Length of rails

- up to 18 m inclusive	± 2 mm
- above 18 m up to 24 m inclusive	± 3 mm
- above 24 m up to 36 m inclusive for fish plated rails	± 4 mm

- above 24 m up to 36 m inclusive for welded rails ± 6 mm
- above 36 m according to agreement

16.3 Diameter of the Fish Plate Screw Holes

- for holes $\leq$ 30 mm ± 0.5 mm
- for holes > 30 mm ± 0.7 mm

16.4 Centering and Position of the Fish Plate Screw Holes in Height and Length

- for holes $\leq$ 30 mm ± 0.5 mm
- for holes > 30 mm ± 0.7 mm

16.5 Straightness

Upon request of the buyer, straightness of the ends will be checked with a 1.50 m long ruler (see sketch on the following page).

- In the vertical direction

  This deviation will be approved only if the end lifts off because of the defect. The largest permissible offset is then 0.70 mm.

- In the horizontal direction

  The maximum permissible offset is 0.70 mm in both directions.

  In all cases, the largest deformation should be located as far from the end as possible (1).

(1) These directives should again be checked in a group of experts, so as to be made more precise.

The rail must be straight between its two ends. It will be checked visually or by any other means that has been developed in agreement between the buyer and manufacturer (1).

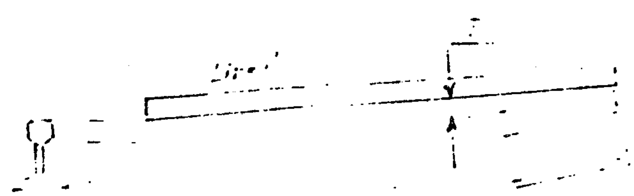
Remarks:

a) In the case of rails for welded rail, the permissible deviation in straightness can be reduced through a special agreement between the buyer and the manufacturer.

b) Rails with a deviation from straightness greater than that specified can be straightened with a press and can be resubmitted for testing.

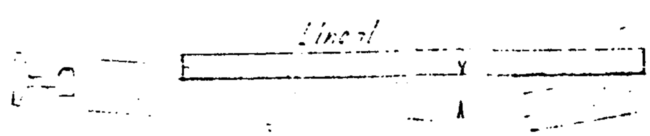
Survey of permissible deviations from straightness

In the vertical plane



$f \leqq 0{,}7$ mm

In the horizontal plane (in both directions)



$f \leqq 0{,}7$ mm

(1) These directives should again be checked in a group of experts, so as to be made more precise.

16.6 Weight

The rated weight, which is to be used as the basis for each rail profile per unit length, is to be calculated with a specific gravity of 7.85.

During production, the weight of the delivered rail is always determined from the length of these rails and the average weight per unit length. This is obtained by weighing 20 rails, half of them selected by the buyer's representative and half by the supplier plant.

The weight determined in this manner may fall below the rated weight by at most 2 percent. Below this limit, the rails will be rejected. Additional weight, which exceeds the upper limit by more than 1 percent, will not be paid for.

21. MANUFACTURING SUPERVISION

The representative of the buyer has the right to supervise manufacture in all details, by day and by night, to be present at all tests which affect melts destined for his administration, and to inspect the test results. He has the right to undertake all necessary follow-up tests, in order to convince himself that the manufacturing conditions provided for have been adhered to exactly.

His supervision will be executed in such a manner that manufacture is not thereby disturbed without compelling reason.

The manufacturer is obliged to inform the buyer at least five days before the anticipated beginning of rolling operations. In the case of deliveries to foreign administrations, this time period is extended to 15 days.

22. PRESENTATION FOR ACCEPTANCE

Before the rails are rolled from ingots, the buyer or his representative will specify those ingots from which specimens are to be taken for the acceptance tests, in agreement with the previously agreed-upon number of tests. These specimens will be arranged according to melt. They must remain available for the representative of the buyer until the acceptance tests for the rails of the appropriate melt have been completed.

In the case of rails from continuous casting, the manufacturer must inform the buyer or his representative concerning the points in the melts and castings at which the specimens have been extracted.

The acceptance work must be performed in such a fashion that it does not disturb the normal course of manufacture.

If a melt is rolled in several parts, the tests performed during the part that is rolled first may be regarded as valid for the remainder of the melt, according to agreement between the buyer and the manufacturer.

23. TYPE OF TESTS

Independent of the surface tests, the quality of the rails will be determined by chemical investigation and by various tests. These tests comprise:

- impact tests at the head sections
- tensile experiments on test rods from base sections
- macroscopic studies on specimens from head sections (for finding bubbles, as well as for the macrogrinding tests), and possibly from base sections (only for the macrogrinding tests)
- Brinel hardness test on the running surface, performed on specimens from base sections, for information.

In case of continuous casting, the manufacturer will specify the points for extracting specimens from the melt and the castings. However, these can be specified more closely by the buyer, in agreement with the manufacturer.

The original specimens as well as the specimens for the counter-investigations, may not be extracted from the "intermediate regions" in the case of continuous casting. Experiments in these regions will be performed only when the totality or portions of the neighboring melt have been rejected as not corresponding to delivery conditions or when the buyer desires additional information.

The test procedures and the conditions for performing these tests correspond to the standards applicable in the manufacturing country, in as much as they have not been specified in the present delivery conditions.

31. WARRANTY

Beginning with the manufacturing year N, which is stamped by the rollers onto the rail web, until December 31 of the year N + 5, the manufacturer guarantees the rails against every defect based on manufacture, which is not discovered during the acceptance procedure in the manufacturing plant.

If, during this warranty period, rails must be pulled from operation because of a rupture or defect, a joint inspection will be made in the presence of the manufacturer and/or an investigation in a test institution.

The buyer obliges himself to make available to the manufacturer, upon request of the latter, sections from the defective rails for the above investigation. The sections are to be taken from locations chosen by the manufacturer so that he can determine the causes of the defect. If a manufacturing error is ascertained during the inspection or investigation, the rail or rails must either be replaced without cost or must be compensated, according to the option of the manufacturer. In the latter case, the manufacturer must pay the new value at the time of removal of the rail or rails that are subject to complaint, increased by supplementary costs for tariff and transport according to the specifications of the buyer administration.

If the above-mentioned joint inspection and investigation does not lead to agreement, a decision must be reached by experts recognized by both sides. The costs will be the burden of the party who is recognized as responsible for the damage.

The defective rails remain the property of the buyer administration.

At the latest by March 31 of each year, the rails that are subject to warranty and that were removed during the preceding year must be reported to the manufacturer. The manufacturer has time to respond for 60 calendar days after his notification, or after transmission of the defective rail sections - if he has requested them.

Appendix 4.2

German Federal Railway

Prestressed Concrete Ties

Technical Delivery Conditions

TL 918 143

Edition of January 1975

1. Characteristics

1.1 By means of its procurement drawing, the German Federal Railway specifies the shape, dimensions, and weight of prestressed concrete ties, and the arrangement of parts for rail fastening The result of the relevant tests are to be entered in a form sheet according to Appendix 3.

1.2 The prestressed concrete ties are to be labelled with the brand mark, the year of manufacture, the type designation, and the width of the rail base. The type of identification will be specified by the German Federal Railway.

1.3 The concrete must have the following minimum strengths for each specimen:

compression strength	$\beta\ W28 = 60\ N/mm^2$ ($600\ kp/cm^2$)
compression strength at the time the prestressing force is applied to the concrete	$\beta\ W_V = 45\ N/mm^2$ ($450\ kp/cm^2$)
tensile bending strength	$\beta\ B7 = 6.5\ N/mm^2$ ($65\ kp/cm^2$)

1.3.1 Cements are permitted only in strength classes Cement 450 and Cement 550 according to DIN 1164.

1.3.2 Concrete additives must satisfy the requirements of DIN 1045 and must be broken predominantly in coarser granulations.

1.4 The arrangement and quality factors of the built-in stressing devices and the anchorings of their ends are to be taken from the reinforcement drawings approved for the German Federal Railway. The tightening steels utilized must be approved by the German Federal Railway. Supervision of their strength characteristics as well as their acceptance is specified by the use decisions set up by the Central Office of the Federal Railway in Munich.

The suppliers of the concrete ties must make certain that the characteristics of the tightening devices and their associated end anchorings meet specifications.

1.5 Insulating and painting materials must be approved by the German Federal Railway.

1.6 The surfaces of the prestressed concrete ties should be as free from pores as possible. Nests with a porous texture are not permitted. Monoflaking can be repaired according to proper engineering practice.

1.7 The bottom surfaces must be rough and flat.

1.8 To the extent that the German Federal Railway requires shaping of the bottom surfaces, such shaped profiles must be produced by vibrating.

1.9 Burrs, which would interfere with transport, are not allowed at the lower edges of the tie.

1.10 Tightening holes at the faces of the prestressed concrete ties must be sealed with cement mortar of minimum strength $Np = 30\ N/mm^2$ ($300\ kp/cm^2$). Other sealants, approved by the German Federal Railway, are permissible. The seal must guarantee safe and permanent protection against the penetration of moisture.

1.11 The day and month, when the prestressed concrete tie was stressed, must clearly be engraved in the sealing of the tightening holes.

1.12 The faces must be completely painted or sprayed with solidly adhering bituminous emulsion, by means of the instant stripping method, when plugging mortar is used.

1.13 The drill holes of the plugs must be free from materials which would impede the turning of the tie screws.

2. Quality tests

2.1 Quality tests are to be performed in the presence of a representative of the German Federal Railway, in accord with Section 2.2.

2.1.1 When the German Federal Railway contracts for at least 50,000 prestressed concrete ties per year, it has the right to employ an official as a continuous "supervisory official of the German Federal Railway in the prestressed concrete plant", and to employ auxiliary forces of the German Federal Railway in the concrete tie plant. The official or his representative will check that production and delivery of the prestressed concrete ties meet the specifications and he will supervise inventories of ties, auxiliary materials, and devices, which are the property of the Federal Railway.

2.1.1.1 The official and his auxiliary forces have the right to enter all rooms and storage spaces used for the production of prestressed concrete ties and to inspect all relevant production equipment.

2.1.1.2 The contractor must provide, without compensation, suitable, closable office space to house the official and his auxiliary forces, including furniture, illumination, heat, and cleaning.

2.2 Type and scope of quality tests

2.2.1 Compression strength and tensile bending strength of the concrete will be tested according to DIN 1048, sheet 1 and DIN 1045.

For fabricating the standard blocks and standard beams, fresh concrete is to be extracted from the spreader outlet and is to be compacted on a vibrating table. The hardening process for the test bodies must correspond to that of the ties. In deviation from DIN 1048, the test bodies are to be stored until the seventh day under water at + 20°C, on a batten grate. The standard blocks are subsequently to be treated, until the 28th day, according to DIN 1048, sheet 1, section 4.1.6. Four standard blocks and three standard beams are to be produced every working day and alternately every shift, in the case of multi-shift operations.

In order to determine the concrete compression strength at the time that the pretensioning force is applied, two blocks must be tested, whose age is the least among the associated group of ties being prestressed. The results of this test are to be entered into the printed form according to Appendix 1.

2.2.2 The compression strength and tensile bending strength of the neat cement are to be determined on seven day old test bodies with dimensions 4 x 4 x 16 cm. With the prestressed concrete ties, the neat cement is used with a subsequent compound to press out the channels for the pretensioning tendons.

Three test specimens are to be fabricated every day and are to be stored according to DIN 1164, sheet 7, section 2. The tests are to be performed in accord with section 3:

Tensile bending strength	β B7	=	6.5 N/cm^2 (65 kp/cm^2)
Compression strength	β W7	=	40 N/cm^2 (400 kp/cm^2)

The test results are to be entered into the printed form according to Appendix 1.

2.2.3 The finished prestressed concrete ties will be tested according to Section 2.2.3.1 through 2.2.3.3, in order to demonstrate that the specified cracking moment has been reached in the center of the tie.

2.2.3.1 For acceptance by the German Federal Railway, the prestressed concrete tie plant will make available to the supervisory official a dynamometer for a random sample test of the prestressing of the prestressed form or of the prestressed tie.

2.2.3.2 After the prestressing force has been applied, the prestressing force present in the prestressing tendons must be tested on at least three prestressed concrete ties per working day. In the case of multishift operations, the test will be performed alternately every shift. The test units must be approved by the German Federal Railway.

In the case of prestressed concrete ties with instant binding, the prestressing force must be tested on the prestressing tendons that are anchored against the forms. The results of the tests are to be entered into the printed form according to Appendix 2.

2.2.3.3 Every week, a prestressed concrete tie must be withdrawn alternately from individual production shifts. It must be stored in water for seven days. Subsequently, it must be supported as a beam on two supports with 1500 mm distance, and must be subjected to a bending test, by means of a single load at the center of the underside of the tie.

For the bending test, the bearings must be designed as roller bearings (rollers with 30 mm diameter). One of these bearings must be tiltable perpendicular to the longitudinal axis of the tie. The bearing plates under the prestressed concrete tie are to be kept 100 mm wide. The single load is to be applied over an edge with 15 mm rounding, which can be tilted perpendicular to the longitudinal axis of the tie. Between the edge and the reinforced concrete tie, is to be arranged a 30 mm wide and 15 mm thick steel plate with a 5 mm thick rubber base.

The minimum value for the bending test, at which the test ties must remain free of cracks, is 48 kN (4.8 Mp) for the B70 concrete tie.

The actual fracture load will be determined twice a month on one prestressed concrete tie. The surface of the center portion of the tie must here be moistened, so that the first penetrating crack can be recognized with the naked eye. A crack is penetrating if it extends over the tie surface and is at least 15 mm long on both sides.

In as much as the test ties satisfy the conditions of the bending test, they will be accepted as full-value prestressed concrete ties.

In the case of instant loading, the seven-day test will be replaced by an instant test on one tie, at the time that one day's output is loaded. The test tie* must be stored in water for three hours before the bending test can be performed. The limit load is 42 kN (4.2 Mp) for the B70 concrete tie.

The test results are to be entered into the present form according to Appendix 1.

3. Specimens and Samples

The German Federal Railway can demand that sample ties - up to 12 units annually - be delivered without compensation, and that simple, not particularly expensive tests or experiments be performed without compensation.

* German: "Prüfstelle" - obviously a misprint.

4. Delivery

4.1 After the quality test has been passed, the prestressed concrete ties will be considered as accepted, and will be considered as delivered except for loading. Acceptance must be confirmed on the bill.

4.2 Up to the time that they are loaded on railway cars, prestressed concrete ties must be stored by the contractor at a storage site that has been made available to the German Federal Railway. Storage must be appropriate, orderly, and separate from the ties of third parties. These storage conditions are applicable unless the ties are immediately loaded for silo stations.

4.3 The prestressed concrete ties must be stored in such a manner that separate loading, according to the age of the ties, the rail fastening, the rail profile, and the construction type is not hindered.

4.4 The foundations for the tie stacks must be dimensioned in such a manner that no damaging settling occurs. The height of the stacks may not be more than 30 prestressed concrete ties.

4.5 Every tie stack must be identified by a plate (50 x 30 cm) with the inscription, "Property of the German Federal Railway", in a durable color (white writing against a black background).

4.6 The prestressed concrete tie plant must be able to store up to one third of the annual contract quantity.

4.7 The contractor must load and ship the prestressed concrete ties on schedule, according to the directives of the supervisory official. The bills of delivery must be written by the contractor in accord with his directives. They are to be signed by the representative of the German Federal Railway.

4.8 It must be possible to load at least 3,000 ties within 24 hours.

4.9 The contractor is liable for the proper loading of prestressed concrete ties according to the specifications of the German Federal Railway. The necessary loading frames are to be kept by the contractor. They remain his property. They are to be returned as service goods by the receiving stations of the German Federal Railway. They are to be returned in open cars, so that they can be unloaded with looped ropes.

The supplementary contractual conditions for auxiliary contracts are applicable for auxiliary materials - Form 164 101.

4.10.1 Materials for rail fastenings, furnished by the German Federal Railway, will be delivered in pallets or similar packets. The cars are to be unloaded by the supplier of the concrete ties and are to be stored properly and protected against weather. The storage room and the storage areas should be able to accept the auxiliary materials required for 50,000 ties. A time of 8 to 12 weeks should be considered for turn-around of the pallets or bindings.

4.10.2 The supervisory official of the German Federal Railway or his representative will check whether the auxiliary materials are usable, if possible even before unloading. Defective deliveries or partial quantities must be noted and a complaint must be entered.

5. Warranty and Replacement

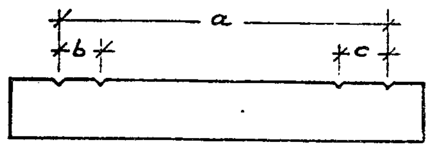
5.1 The warranty period ends upon the expiration of the fifth calendar year following the manufacturing year.

5.2 For prestressed concrete ties, which exhibit damage within the warranty period, but which are nevertheless still suitable for tracks and consequently are not removed, a reduction is to be agreed upon and compensated, in as much as these damages are based on manufacturing defect.

5.3 Prestressed concrete ties, which exhibit damage during the warranty period and which must be removed, are to be replaced, within a suitable time, by ties meeting specifications.

5.4 Prestressed concrete ties, against which exception has been raised, are stored at the cost and risk of the contractor.

Week Number: from year 19 to	CONCRETE TIE SUPPLIER PLANT



W 54	W 60	tilt / cant
a =	a =	tilt
b =	b =	
c =	c =	cant

day of test		Monday	Tuesday	Wednesday	Thursday	Friday	Saturda
1	a						
	b/c						
	tilt/cant						
	Form						
2	a						
	b/c						
	Verk/Neig						
	Form						
3	a						
	b/c						
	Verk/Neig						
	Form						
4	a						
	b/c						
	Verk/Neig						
	Form						
5	a						
	b/c						
	Verk/Neig						
	Form						
6	a						
	b/c						
	Verk/Neig						
	Form						
7	a						
	b/c						
	Verk/Neig						
	Form						
8	a						
	b/c						
	Verk/Neig						
	Form						
9	a						
	b/c						
	Verk/Neig						
	Form						
10	a						
	b/c						
	Verk/Neig						
	Form						
plant tester	PS						

TEST RESULTS ACCORDING TO TL 918 143

GERMAN FEDERAL RAILWAY TEST RESULTS ACCORDING TO TL 918 143 Appendix 1 (2.2.1, 2.2.2, 2.2.3.3)

Week Number: from
year 19 to

CONCRETE TIE SUPPLIER PLANT

				Mon	Tues	Wed	Thurs	Fri	Sat	Suppl
Day of test										
blocks	from the tie vibrating table/ laboratory vibrating table	pressure β_{WV}	kp/cm²							
			kp/cm²							
			kp/cm²							
		age in hours								
		mfg.date								
		pressure β_{W28}	kp/cm²							
			kp/cm²							
			kp/cm²							
beams		mfg.date								
		β_{37} tensile bending	kp/cm²							
			kp/cm²							
			kp/cm²							
neat cement		tensile bending β_{B7}	kp/cm²							
			kp/cm²							
			kp/cm²							
		pressure β_{W7}	kp/cm²							
			kp/cm²							
			kp/cm²							
		flowability								
ties		instant loading								
	after 7 days	mfg.date								
			4,8 t							
		mfg.date								
		fracture load in t								
		supplementary tests	t							
			t							
			t							
tester		plant								
		DB								

remarks and other tests

1) If test studies are produced daily on the tie vibrating table, test studies must once a week also be produced on the laboratory vibrating table - and vice versa.

TEST RESULTS ACCORDING TO TL 918 143

Week Number: from	CONCRETE TIE SUPPLIER PLANT
year 19 to	

Day of Test		Monday		Tuesday		Wednesday		Thursday		Friday		Saturday	
	1 2	1	2	1	2	1	2	1	2	1	2	1	2
	4 3	4	3	4	3	4	3	4	3	4	3	4	3
prestressing device 1	t												
	t												
	Σ /max Diff												
	t												
	t												
	Σ /max Diff												
	t												
	t												
	Σ /max Diff												
prestressing device 2	t												
	t												
	Σ /max Diff												
	t												
	t												
	Σ /max Diff												
	t												
	t												
	Σ /max Diff												
prestressing device 3	t												
	t												
	Σ /max Diff												
	t												
	t												
	Σ /max Diff												
	t												
	t												
	Σ /max Diff												
dynamometer test	dynamometers												
plant tester													

Appendix 4.3

GERMAN FEDERAL RAILWAY

TL 918 61
Edition of
February 1966

BALLAST OF GRANULATIONS 1 and 2

Technical Delivery Conditions

1. Characteristics

1.1 Raw material requirement

1.1.1 The rocks and slags used to produce ballast must be stable as regards frost, volume, and weather. They must have a compression strength of at least 1800 kp/cm^2 and a minimum impact strength of S = 50 (method of the German Federal Railway according to DIN 52 109).

Inferior, decomposed, and weathered stones may not be processed.

Slags must correspond to the regulations of DIN 4301.

1.2 Purity of the ballast

1.2.1 The ballast must be free of earth, clay, loam, marl, plant residues and other inferior or damaging components.

1.3 Granulation of the ballast

1.3.1 Definition of terms

The dimensions prescribed for the granulation of the ballast do not refer to the size of the stone pieces themselves, but to the diameter of round hole test sieves, by means of which the granulation is specified in terms of an upper and lower limit.

For example, ballast of granulation 1 is a sifted material, which is sifted on two round hole test sieves with 25 mm diameter and 65 mm diameter, and which then falls through the coarser test sieve, but is retained on the smaller test sieve.

1.3.2 Granulation 1

should be composed of all granulation categories between 25 mm diameter and 65 mm diameter in such a fashion that the following are always attained in the individual component granulations:

in the component granulation 25/30 mm diameter between 0 and 10 weight percent

in the component granulation 30/40 mm diameter between 15 and 25 weight percent

in the component granulation 40/50 mm diameter between 25 and 25 weight percent

in the component granulation 50/60 mm diameter between 30 and 40 weight percent

in the component granulation 60/65 mm diameter between 5 and 15 weight percent

A deviation of ± 5 weight percent in the component granulations from 30 to 60 mm diameter remains unexceptionable.

A component of excessively small pieces, amounting to 5 weight percent in the contiguous granulation 20 to 25 mm diameter, and a component of excessively large pieces, amounting to 5 percent in the contiguous granulation 65 to 70 mm diameter also remains unexceptionable.

The pieces which fall through a round hole of 10 mm diameter, and pieces which do not fall through a round hole of 70 mm diameter, as well as pieces exceeding a length of 85 mm, should not be admixed to the delivery.

1.3.3 Granulation 2

should be composed of all granulation categories between 15 mm diameter and 30 mm diameters. 33 1/3 weight percent in the component granulation categories 15/20 mm diameter, 20/25 mm diameter, and 25/30 mm diameter may here not be exceeded.

A fraction of excessively small pieces below 15 mm diameter, up to 5 weight percent, and a fraction of excessively large pieces above 30 mm diameter, up to 10 weight percent, is unexceptionable.

Pieces which fall through a round hole of 5 mm diameter, and pieces which do not fall through a round hole of 40 mm diameter, as well as pieces longer than 15 mm, should not be admixed with the delivery.

1.4 Grain form of the ballast

1.4.1 The ballast should consist as much as possible of irregularly shaped, sharp-edged pieces. In all component granulation categories, flat-shaped as well as wedge-shaped and compacted stones should be contained.

The ratio of flat/compact material of granulation 1 must be at least 7 percent and may not exceed 20 percent.

1.4.1.3 With granulation 2, the content of flat pieces may be at most 33 1/3 percent.

2. Quality Test

2.1 General Considerations

2.1.1 The ballast will be tested by the contracting agency. The test results of the "Stone Test Office of the German Federal Railway at the Kassel Federal Railway Directorate" is binding for the contractor.

If differences of opinion concerning the test results arise with the contractor, a new specimen can be extracted at his request, and a new test can be performed.

2.1.2 All specimens must be of at least 50 kg, and must be obtained by dividing a pile into quarters.
components contained in the delivery must also be present proportionately in the specimen quantity.

2.2 Tests for Use Value

2.2.1 Tests will be performed whether the material and constructive properties satisfy the requirements under Section 1.

Test procedures:

Specimen extraction	DIN 52 101
Gross and net specific gravity	DIN 52 102
Frost resistance	DIN 52 103 Section C or respectively DIN 52 113

Compression strength	DIN 52 105
Weather resistance	DIN 52 106
Impact strength	Method of the German Federal Railway according to DIN 52 109
Delivery room weight	DIN 52 110
Furnace slags	DIN 4301

The granular composition and the purity of the granulate mixture will be determined.

Granulation form will be determined according to the method of the German Federal Railway.

2.2.1.1 For granulation 1, by the number of stones in a volume of 40 l, consisting of 5 l component granularity 25/30, and of 10 l each of component granularity 30/40, 40/50, 50/60 and of 5 l component granularity 60/65 mm diameter.

Accordingly, if less than 251 and more than 283 stone pieces are present in a volume of 40:4 = 10 l, the condition of Section 1.4.1.2 is not fulfilled.

2.2.1.2 For granularity 2, by the percentage weight of flat pieces, averaged from 1000 g each of the component granulations 15/20, 20/25, 25/30 mm diameter, which are thinner than 1/5th the hole diameter of the respective upper round hole seive.

The conditions of Section 1.4.1.3 are fulfilled, if the content of flat pieces does not exceed 33 1/3 weight percent.

2.2.2 The contractor is obliged to eliminate from processing into ballast all raw materials which the contracting agency has declared as unsuitable on the basis of its own investigations. If the contractor does not meet this obligation, the contracting agency may withdraw from the contract.

3. Specimens and Samples

3.1 Before beginning a first delivery, the contractor must, upon request, submit for testing at least 50 kg ballast, which has been produced by his enterprise. He must send this ballast to the Stone Testing Agency at the German Federal Railway Kassel, Destination Station Hbf.

3.2 The delivery may only be implemented when the ballast has been recognized as meeting specifications.

3.3 The contracting agency has the right to monitor the mining and fabrication of ballast in the various enterprises, and to withdraw specimens for investigation.

4. Delivery

4.1 The ballast is to be delivered FOB railway car shipping station of the supplier plant.

4.2 Payment will be made according to weight.

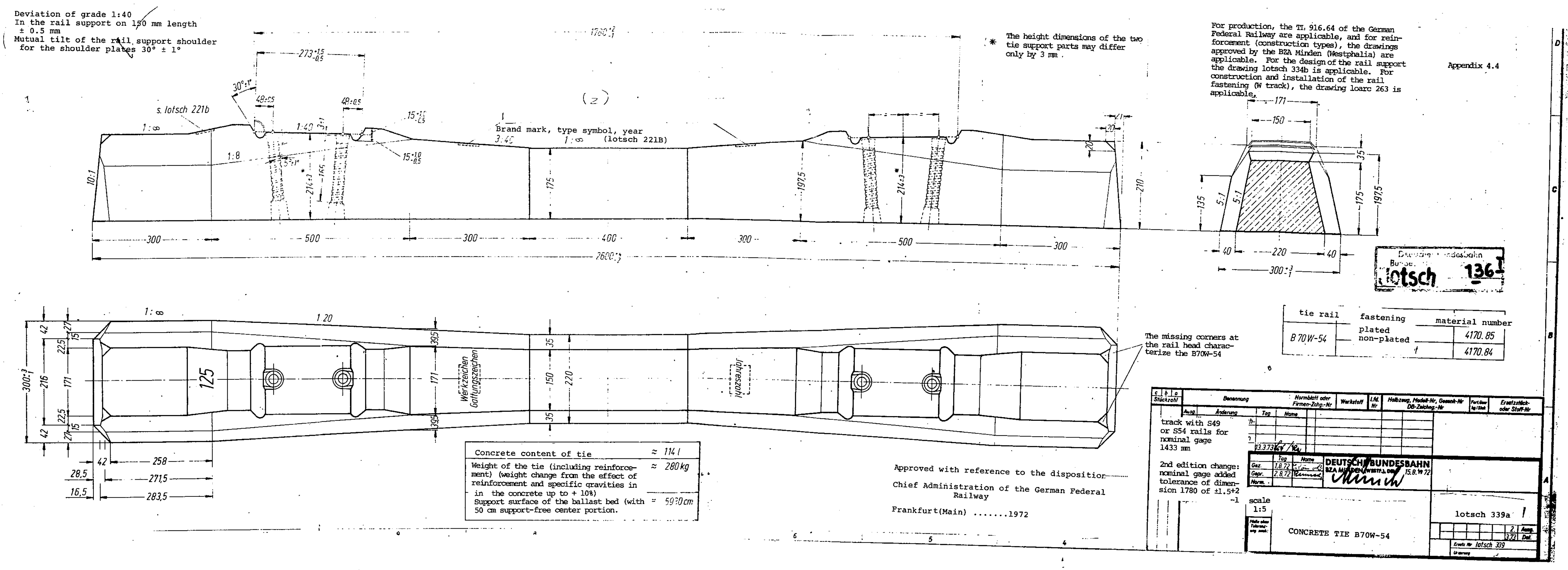
4.3 The weight of a freight car load is to be determined by the contractor at the shipping terminal or at the plant in tons with two decimal places, as an official railroad weight. The official weighing certificates are to be glued to the bill of lading.

5. Warranty and Replacement

5.1 If ballast is rejected during the final acceptance as not meeting specifications, the contractor, upon request, must replace it within three weeks by material meeting specifications. The rejected ballast will be stored at the cost and risk of the contractor.

5.2 If the contractor does not provide a replacement within the specified time limit, he must replace the amount expended by the German Federal Railway, but at least the purchase price.

5.3 For ballast which is recognized as not meeting specifications, only after it has been unloaded or after it has been installed, the reduced value is determined by employing the Stone Testing Agency at Kassel, and this reduction must be compensated by the contractor.



For production, the TL 916.64 of the German Federal Railway are applicable, and for reinforcement (construction types), the drawings approved by the BZA Minden (Westphalia) are applicable. For the design of the rail support the drawing lotsch 334b is applicable. For construction and installation of the rail fastening (W track), the drawing loarc 263 is applicable.

tie	rail	fastening	material number
B70W-54		plated	4170.85
		non-plated	4170.84

Concrete content of tie	≈	114 l
Weight of the tie (including reinforcement) (weight change from the effect of reinforcement and specific gravities in the concrete up to + 10%)	≈	280 kg
Support surface of the ballast bed (with 50 cm support-free center portion.	≈	5930 cm

Approved with reference to the disposition.........

Chief Administration of the German Federal Railway

Frankfurt(Main) 1972

track with S49 or S54 rails for nominal gage 1433 mm

2nd edition change: nominal gage added tolerance of dimension 1780 of ±1.5+2 -1

scale 1:5

DEUTSCHE BUNDESBAHN
BZA MINDEN (WESTF.)

CONCRETE TIE B70W-54

lotsch 339a 1

Ersatz für lotsch 339