Sudan University of Since & Technology

College of Engineering

School of Mechanical Engineering

SRC Motive Power, Rolling stock & Track

(Design, Specification, Construction & Safety)

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الآيـة

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قال تعالي:

﴿وَقُلِ اعْمَلُوا فَسَيَرَى اللَّهُ عَمَلَكُمْ وَرَسُولُهُ وَالْمُؤْمِنُونَ وَسَتُرَدُّونَ إِلَى
عَالِمِ الْغَيْبِ وَالشَّهَادَةِ فَيُنَبِّئُكُمْ بِمَا كُنْتُمْ تَعْمَلُونَ ﴾

سوره التوبة الاية : 105

صدق الله العظيم

الإهـــــداء

إلى ملاكي في الحياة .. إلى معنى الحب وإلى معنى الحنان والتفاني .. إلى بسمة الحياة

وسر الوجود

إلى من كان دعائها سر نجاحي وحنانها بلسم جراحي إلى أغلى الحبايب

(أمي الحبيبة)

إلى من جرع الكأس فارغاً ليسقيني قطرة حب إلى من كلّت أنامله

ليقدم لنا لحظة سعادة

إلى من حصد الأشواك عن دربي ليمهد لي طريق العلم إلى القلب الكبير

(والدي العزيز)

إلى الروح التي سكنت روحنا

الآن تفتح الأشرعة وترفع المرساة لتنطلق السفينة في عرض بحر واسع مظلم هو بحر الحياة وفي هذه الظلمة لا يضيء إلا قنديل الذكريات ذكريات الأخوة البعيدة إلى الذين أحببتهم وأحبوني

(اصدقائي الاعزاء)

أسمى آيات الشكر والامتنان والتقدير والمحبة إلى الذين

حملوا أقدس رسالة في الحياة**...**

إلى الذين مهدوا لنا طريق العلم والمعرفة**...**

(اساتذتنا الاجلاء)

نهدي هذا الجهد المتواضع آملين ان يساهم في دفع مسيره التطور والنماء.

الباحثون ...

الشكر والتقدير

من أي أبواب الثناء سندخل وبأي أبيات القصيد نعبر، وفي كل لمسة من جودكم وأكفكم للمكرمات أسطر... إن قلت شكراً فشكري لن يوفيكم، حقاً سعيتم فكان السعي مشكوراً، إن جف حبري عن التعبير يكتبكم قلب بة صفاء الحب تعبيرا.... كل كلمات الشكر والتقدير عاجزة عن وصف ما يجول بداخلنا من تقدير ومحبة واحترام إلي الأب ومربي الأجيال...

الدكتور: الفاتح بابكر محمد نبق

المهندس الاستشاري

الذي لم يتواني علينا بوقته وملاحظاته القيمة وإضاءة الطريق لنا لنصل الي هذا الجهد المتواضع الذي هو قطره من فيض بحر فلك منا كل التقدير والاحترام ....

الحمد لله الذي بحمده تدوم النعم، بمناسبة التخرج نتقدم بالشكر والعرفان و الاحترام بعد شكر الله سبحانه وتعالى إلى كل القائمين على هذا الصرح الكبير جامعة السودان للعلوم والتكنولوجيا و إلي هذه الأرواح النقية والعقول النيّرة التي لم تتوانى في تقديم كل ما هو مفيد وتبسيط وتذليل كل عسير في طريق الطلاب والطالبات ونخص بالذكر على سبيل الإيجاز لا على سبيل الحصر أخواننا الأفاضل و أساتذتنا الأجلاء بمدرسة الهندسة الميكانيكية والذين لم ولن نوفيهم حقهم وتقديرينا لهم قليل من كثير،نسأل الله الكبير القدير الكريم لهم التوفيق والسداد ...

الباحثون ...

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التجريدة

الأهداف العامة لهذه الدراسة هي حزمه متكامله من الدراسات و التحليلات لمكونان عمل سكك حديد السودان من اجل الاستخلاص والتوصيه بما تُّوصّل اليه من معلومات عن انحرافات المعنيين عن مواصفات المعدات المعتمده كالقاطرات وما تم من تطوير لتصميمات عربات نقل البضائع.

كذلك العلاقه الديناميكيه بين العجلات والقضبان وما ينتج عنها من عدم توازن وحوادث سقوط .

الهدف المحدد لهذا البحث هو اجراء عمليات حسابيه و تقديم دراسات تصميميه يتم بموجبها رفع السرعه في بعض خطوط السكة الرئيسيه الي 120 كلم/س بدلا من 60 كلم/س وكذلك رفع الوزن المحور (axles load) الي عشرين (20) طن بدلا عن 16.5 طن.

لقد تم هذا العمل البحثي (Research work) و قدم مدعوم بكل ما يحتاج اليه البحث لاثبات دقة ما جاء به.

هذا البحث يثبت ان تطوير خطوط السكة حديد في مجالي السرعه ورفع معدل النقل ممكننا لتخدم هذه الخطوط المتطوره منقولات السودان الي مائة عام و اكثر ان شاء الله.

تم عرض خلاصات وتوصيات اهمها ان تجري تجارب عمليه في السكة الحديد بقاطرات ماكينات الجر فيها (traction diesel engine) ذات سرعات لف متوسطة (medium speed) واخري ماكينات الجر فيها ذات سرعات عالية (high speed) للتاكد من صرف الوقود لكل نوع ليتاكد ان الماكينه ذات السرعة المتوسطة هي الامثل كما حددت مواصفاتSRC وكذلك سكك حديدية اخري كالسكة حديد البريطانيه.(British Rails)

نتائج التجارب الموصى عليها أعلاه سوف تكون مفيده جدا لسكك حديد السودان لتلتزم بتحديد مواصفاتها للماكينة ذات السرعات المتوسطة.

Abstract

The general objectives of this project is a comprehensive package of study & analysis of the main components of SRC business in order to draw conclusions & recommendations on deviations from specifications in motive power, development in rolling stock bogie design & the interaction between wheel/rail to analyze the stability & safety of trains on the track, this has been covered in chapters two & three.

The specific objective of this project shall be a preliminary technical feasibility study for some of the existing main line routes of SRC to be up-graded to a higher speed of 120km/hr. & a higher axle load of 20 tons which is a redesigning work involving train track dynamics studies. This has been done as a research work which showed that the NARROW GAUGE of SRC track with up-grading can serve Sudan freight transportation for more than hundred years.

This has been presented in the Research Work Chapter (chapter four) with calculations sported by references

Chapter five of the conclusions & recommendations recommends practical testing for the fuel consumptions to be conducted in SRC to show that diesel engines in traction application have to be of medium speed.

The results of these tests shall be very useful to SRC.

CHAPTER ONE

INTRODUCTION

1.1 Introduction

This project is intended to be linked with a place of intensive engineering applications in order to mate theory with practice as means of engineering maturity.

Sudan railways corporation (SRC) is one of the biggest engineering corporations in Sudan with a number of engineering fields of application which include mechanical, electrical & electronic engineering applied in its motive power (locomotives) rolling stock (passenger & freight wagons) and other plant & machinery equipment used in workshops for the maintenance and operation of freight and passenger trains .

Also the railway (civil engineering) has a lot to do with the study of the motive power and the rolling stock because of the interaction between wheel/rail which involves design, construction, maintenance & safety for both the track and units running on it.

The components of the permanent way; rails, fastenings, base plates, fishplates, sleepers, etc. are all of mechanical engineering design & nature.

The train track dynamics studies which helped in redesigning for enhancement of both motive & rolling stock and the track is almost done exclusively by mechanical engineering dynamicests.

Some of the train-track computer programmers are used for

a) The establishment of the dynamic characteristics of track and equipment.
b) The development & validation of mathematical models to permit the rapid analysis of the effects on dynamic stability of modifications in design, maintenance, and use of equipment and track structure.

c) The development of interim guidelines for train handling, makeup, track structures, and engineers training to reduce excessive train action.
d) Development of improved track and equipment specifications and operating practices to increase dynamic stability.

Based on the above, this project shall be executed in SRC under the supervision of one of SRC ex-engineers.

As it may be drawn from the title of the project:

Motive power, Rolling Stock & Track (Design, Specifications, construction & Safety), the four users requirements could be studied when studying the design, construction, etc.

No computer programme text or results of its use shall be part of this project. But where & how these programmes have been used shall be displayed in the project report.

1.2 General objectives of the project

The objective of this project is a comprehensive package of study & analysis of the main components of SRC business in order to draw conclusions & recommendations on deviations from specifications in motive power, development in rolling stock bogie design & the interaction between wheel/rail to analyze the stability & safety of trains on the track.

A railway train, which consists of a locomotive (motive power) &Carrages /wagons (rolling stock) runs on a railway track.

The designer/manufacturer is responsible of designing & manufacturing & the user (railway) is responsible of maintenance & operation of trains.

The results of the designer's/manufacturer's & users' efforts are evaluated by performance & safety of running trains.

To achieve good performance & safety of running of trains, design, maintenance & operation must be fulfilled as specified &required. This should include the four main components of running of trains i.e. motive power, rolling stock, the railway track & train operation.

1.3The specific objective of the project

The specific objective of this project shall be a preliminary technical feasibility study for some of the existing main line routes of SRC to be up-graded to a higher speed of 120km/hr. & a higher axle load of 20 tons which is a redesigning work involving train track dynamics studies.

Conclusions & recommendations shall be presented as contribution to help SRC by having a view point from outside.

1.4 Study area

The report of the project is planned to be presented into five chapters including this chapter (Introduction).

Chapter two shall be on motive power specifications and damage done by acceptance of equipment not to specifications.

This shall include:

Deviation from the motive power specifications (engine &bogie).

Chapter three shall be on motive power & rolling stock bogie design & performance.

This shall include

a) Types of bogies.
b) Bogie design.

c) Freight wagons & rail cars bogies in SRC
d) Different Analysis of four types of bogies.

Chapter four shall be on research work (up- grading of SRC track for higher speeds, loads, stability& safety).

This shall include

a) The track.
b) Up-grading of some SRC main line routes for higher speeds.
c) Up-grading the same lines to 20 tones axle load.
d) Stability & safety.

Chapter five, which is the last chapter of the report, shall be the conclusion & recommendation reached to which we hope they be of use especially for SRC authorities.

1.5 Methodology

The methodology to be followed is:

- Specifying the field of research.
- Collection of relevant literature & data.
- Lectures on the subjects of the project by the supervisor & discussions on the topics.
- Study, analysis & discussions with personnel in the railway field.
- Summing up, conclusions & recommendations.
- Report writing.
- Criticism of report.
- Final report writing & revision.

As it may be drawn from the title of the project: "Train Track Dynamics Programmes as a tool".

Studies on motive power, rolling stock, and track; design, maintenance & safety which made use of different programmes shall be included in the project report.

Subjects to be studied could be

1) Specifications of equipment and their enhancement over the years.
2) Design of different types of bogies (trucks) & their development to enhance their performance.
3) Safe running of trains (stability & derailments) which is directly linked with the interaction between the wheel & rail and comes under the heading "train track dynamics".

Visits

- Workshops of the central region of SRC.
- The concrete sleeper's factory in Khartoum North.
- Sections of SRC track.

CHAPTER TWO

DEVIATIONS FROM THE SPECIFICATIONS OF THE DIESEL ELECTRIC LOCOMOTIVES

2.1 Introduction

SRC have specification for almost every piece of equipment which has any significant effect on the running of trains.

The specifications of locomotives are PRIORITY NUMBER ONE to be strictly followed. Any deviation from any item of the specification shall lead to the rejection of a bidder's offer. If a better specification has been offered, the technical committee for the study of offers may recommend the acceptance of this deviation but they have to raise strong support to that, then it is left to the board of heads of departments, headed by the general manager of SRC to decide on the acceptance or rejection.

Of late, after 1990, the system of purchases has not been strictly followed. Locomotives have been purchased outside this rule.

Such locomotives have some major deviations from the specification. These deviations have costly results.

To name two of these deviations:

(a) Deviation from the specified medium speed diesel engine to the high speed diesel engine.
(b) Deviation from the traction motor axle hung nose suspension system.

Items of specifications for these two deviations shall be quoted, here under.

A graduation project, executed by a group of our colleagues in 2013 has quoted the experience and opinion of designers, users, consultants in the fields of railways on the use of high speed diesel engines (1500 rpm) in rail freight transportations, praising the use of the medium speed diesel engine as a proven traction engine. (See appendix Y).

We shall start from where our colleagues stopped and recommend practical test runs which can be executed in SRC to prove to what extent this deviation cost SRC in one item only which is the fuel consumption.

Also the traction motors suspension system specified & used in all SRC locomotives, purchased by following the right way of purchase, is the axle hung nose suspended traction motors with plain bearing versus motors supported on a suspension unit using rolling bearings.

This system is of better design but it depends on who manufured it? Has it been proven?

The chapter shall present relevant studies and information to clarify the seriousness of these two deviations to help SRC investigate further to stick to their specifications.

Subjects to be displayed & studied here under shall be on:

a) Items of specifications & deviations from them.
b) Four locomotives bogies in service in SRC.
c) The design & manufacture of the modern wheelset.
d) Rolling bearing in traction motors & suspension units.

2.2 Quotations from SRC specifications for the diesel electric locomotives

a) The diesel engine items quotation

“Engine built by liscence shall not be considered

The diesel engine shall be of proven design suitable for rail traction and shall be already in main line locomotives operating successfully in countries having desert and tropical climate with dusty winds, sand storms and high temperatures. Details of such locomotive operating elsewhere should be stated along with their duration in service.

The engine shall comply with B.S. 2953: 1958 except as here in specified and shall be tested in accordance with this specification.

All technical information specified in B.S. 2953: 1958 is required for submission.

There shall be no exclusion on the ground of identical engine having been supplied on previous contracts.

The engine shall develop 1473/1492 KW (1975/2000 HP), continuous traction rating at site, at a speed not exceeding 1100 r.p.m. It shall be capable of being up grated to 1560 KW without distress, without excessive increase in exhaust temperature and without any increase in maintenance. The engine shall comply with all provisions of this specification at this rating. Complete derating calculations shall be given."

b) The bogie, wheelsets& traction motors suspension items quotations

Bogies

"The bogies shall be of cast steel construction, of centre pivot with a bolster which will adequately support the locomotive superstructure and contain the various wheels and axles.

The bogies must provide:-

1. Good tracking to reduce flange and rail wear.
2. Satisfactory weight transfer.
3. Control lateral motion for good riding qualities.
4. Durability and reliability.
5. Simplicity for accessibility and maintenance.

Details of how all these functions were achieved must be submitted.

Fabricated and composite bogies will be considered as alternatives.

Wheel Centres, Tyres and Axles

"The wheel centres shall be of the solid rolled steel disc type in accordance with BS.468-1956 or equivalent UIC standards, and shall be fitted with tyres127 mm x 76.2mm (5" x 3") 109-119 tons /mm^2 (70-77 tons/in^2) tensile strength to BS 24 part 2-1956 or equivalent UIC standards securred by retaining rings No. 7 section as shown on SRC drawing No. 12923.

The tyre profile shall be in accordance with SRC Drg. No.8357 (revised) and the diameter of the wheels over tread shall be 914.4mm (36 inches).

Axles shall be in accordance with BS 24 2-1957 or equivalent UIC standards and shall be machined all over. High finish shall be given to the seats and bearings surfaces and permanent lathe centre of 60degres shall be left in.

Wheels shall be pressed on the axles hydraulically with a pressure of 0.39 – 0.58 tons per mm (10-14 tons per inch) diameter of wheel seat, in accordance with BS 3117, part 5-1959 or equivalent UIC standards.

Oil injection removal shall be provided for removal of wheel centre and gear wheel."

Traction Motors

"The traction motors shall be of the axle hung nose suspended, forced ventilated type designed in accordance with BS173; 1960 or with equivalent UIC standard. Felt wick lubrication for the suspension plain bearings is the acceptable practice. Fig. (2.1a)

The traction motor shaft bearings shall be rolling bearings (not plain bearings which are considered inferior to the rolling bearings)".

2.3Bogies of four locomotives in service in SRC

a) Design and deviation of class 58 bogie CP3 [1]

(English Electric (Roston diesels) locomotive bogie)

The class 58 bogie which is designated CP3 (used in British Railways (BR)) is a development of the CP1 bogie (used in SRC). To illustrate this development it is necessary to describe the features included in the CP1 bogie.

The CP1 was designed to give a good ride quality and have a minimal number of wear surfaces to reduce life costs. The frame largely fabricated from mild steel plates into box sections proportioned to avoid metal fatigue throughout its 35 year life. In areas where there difficult geometry, such as the transoms which incorporated the traction motor mountings, steel castings were used. The frame had very clean lines with a fully machined flat bottom plate and the secondary flexicoil springs were mounted directly onto the solebars. Instead of the conventional type of horn guide the primary suspensions consisted of rolling ring providing wheelset yaw and lateral stiffness. The rubber ring contributed a little to the vertical suspension stiffness which was mainly provided by coil springs.

Floating bearings were fitted at the centre axle position to permit negotiation of small radius curves. A bolsterless secondary suspension utilizing flexicoil springs were used to support the body. Rotation of the bogie was accommodated by the deflection of these springs. No equalization between axles was attempted but no novel traction connection via inclined links from the headstocks at each end of the bogie minimized weight transfer within each bogie.

Tread brakes were used with twin blocks each side of the wheel. The twin blocks were independently actuated by its own brake cylinder. Axle hung, nose suspended traction motors were arranged as the standard Class 56 locomotive so that the four outer motors face the loco centre and the two inner motors face outwards. This arrangement minimizes the bogie pitch inertia necessary for good ride quality and higher speed adhesion performance.

The CP3 bogie had to make considerable changes from that of the CP1 because of the superstructure layout. The traction links were replaced by conventional pivot with rubber traction pads mounted fore and aft, as used on the standard Class 56 locomotive.

The simple strength under frame resulted in the locomotive having a low bending frequency (4.0Hz) and in order to minimize excitation of the body flexing mode from bogie pitch movement some adjustment of the traction centre was required. The optimized height of the traction centre was determined by mathematical model analysis and showed that a drop from the original height of 910 mm to 595 mm was necessary. This put the traction centre practically on the axle centre line which is close to the optimum value for minimizing weight transfer. Details of the comparative weight transfer for CP1, CP2 and CP3 bogies are shown in fig (2.1b).

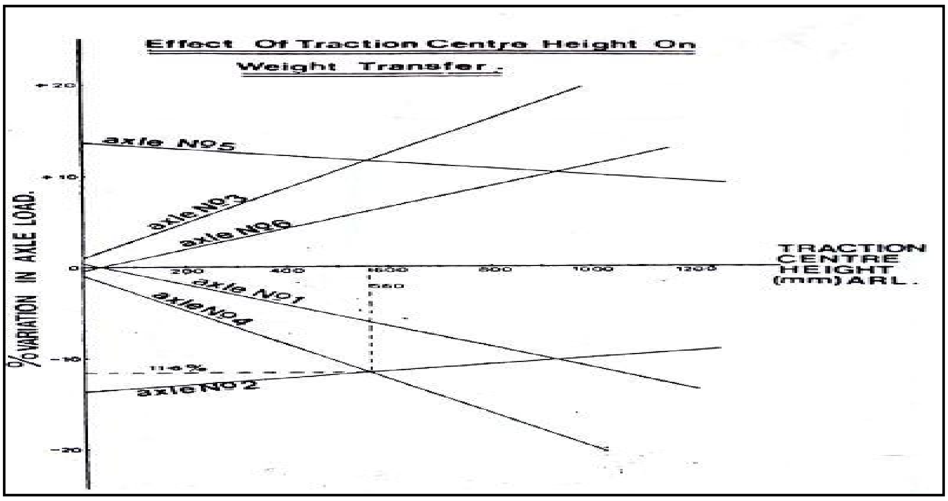


Fig (2.1) the comparative weight transfer for CP1, CP2 and CP3 bogies

WEIGHT TRANSFER.

Bogie Type	Axle No	Percentage Axle Load Variation					
		1	2	3	4	5	6
CL56, CP2 BOGIE equalised 22% adhesion		−10·6	−10·6	+20·3	−20·3 *	+10·6	+10·6
CL56, CP1 BOGIE inclined links 22% adhesion		−3·6	−12·0 *	+6·8	−6·8	+12·0	+3·6
CL58, CP3 BOGIE electrical compensation 21% adhesion		−5·5	−9·4	+11·9	−12·9 *	+9·4	+6·0
CL58, CP3 BOGIE 22% adhesion		−5·7	−10·6	+11·4	−11·4 *	+10·6	+5·7
CL58, CP3a BOGIE equalised 22% adhesion		−5·5	−11·1	+11·6	−11·6 *	+11·1	+5·5

+ve on loading.
−ve off loading.
* Indicates Limiting Axle.

Fig (2.2) weight transfer

The CP3 frame is also a mixture of castings and fabricated sections but this time the castings form the major parts of the bogie. Four major castings are incorporated, two side frame centre section portions, which also provide the flexicoil spring seating joined together by means of a transverse circular section casting. This provides motor nose supports for the outboard traction motors and the housing for the traction pads. The inboard traction motors are supported by a small rectangular section transverse casting. The rest, of the bogie frame is fabricated into box sections and has a headstock at the inboard end only.

The type of brake gear on the CP3 is similar to that on CP1. The traction motors are the same as those used on Class 56 locomotives.

During the initial operation wheel slip problems arose principally in colliery sidings in poor adhesion conditions which have necessitated two further mechanical modifications to the CP3 bogie design. The first being the fitting of additional sanding equipment to the inner axle, and the second is soft primary vertical springs to the centre axle for improved equalization between axles.

A value analysis study of the CP3 bogie was undertaken in 1983 with the objective of reducing total life costs, weight of the bogies and improved maintenance. At this time there was a firm intention to proceed with a follow on order for a further 39 locomotives. As a result of the subsequent order being limited to15 these later locomotives are to be fitted with the developed CP3 a bogie design which incorporates SAB brake equipment with the existing frame. In the interests of standardization the bogie variances were limited. A fully developed new bogie frame as identified in the study was not progressed .The main priority was to improve sanding capacity on the inner and outer axles and introduce the simpler SAB brake units.

The SAB brake unit as fitted to the CP3 a bogie has integral brake lever and block carrier mechanisms. This results in a saving of components over the Westinghouse scheme which incorporates a separate brake hanger. There is also a consequent saving on bogie frame machining.

The inner outer sandboxes are only different by their handing with increased capacity, and better access for filling the box at the outer positions in comparison with the original design. The sandbox has been designed to support both the SAB brake unit and the sand injector assembly –the main support member for the injector nozzle being an integral part of the sandbox.

The finalized design illustrates how BR engineers are using current technology to produce bogies that will give reliable performance for long periods in service (8-10 years) without the need for main works attention.

b) GLC Locomotive Truck Assembly Description GM (EMD) [2]

(General Motors/Electric Motive Diesel (Cat/EMD) USA, locomotive bogie)

Description

The trucks support the weight of the locomotive and provide a means for transmission of power to the rails. They are designed to withstand the stress resulting from road shock due to normal variations in the road bed and other conditions encountered during operation. An important function of the truck assembly is to absorb and isolate these shocks so they will not be transmitted to the locomotive under frame and the equipment mounted on the under frame.

The locomotive tractive horse power is supplied to the traction motors. The motors are geared to the driving axles which in turn apply this force to the rail through the wheel. The tractive force is transmitted through the axle journal bearing adapters to the truck frame and through truck frame pressure areas to mating pressure areas on the truck bolster. The bolster then transmits the force through its centre bearing to the carbody locomotive and supply the locomotive drawbar horse power.

The locomotive weight is applied at the bolster centre bearing. The “H” design bolster is supported at each of its four centres by truck frame mounted rubber pad. The four centres of the bolster are held between upright pedestals which are an integral part of the frame. This bolster pedestal arrangement serves to transmit force from the bolster to the frame or the frame to the bolster. Stops are provided on the bolster stops on the frame, to limit the bolster side movement.

The main frame on the truck is supported on sets of double coil springs above each bearing adapter.

Damaged Threads

All threaded holes should be checked and retapped if required. If the threaded holes cannot be reconditioned by retapping they should be plug welded, redrilled and tapped. An alternate method of reclaiming unsatisfactory threaded holes is to retap them to accommodate an oversize blot.

Missing parts

Make a thorough inspection to see that all the necessary parts are intact. Special attention should be given to wear plates, cotter keys, washers, bushings, studs, brake guides, and brake pins.

Bolster

The bolster, Fig. (2.3), is a steel casting used to transfer the locomotive weight to the truck frame. As previously explained, the truck bolster centre bearing mates with the locomotive under frame centre bearing. A neoprene rubber dust guard, Fig. (2.4), clamps over the truck center casting and rides against the under frame to prevent dust and dirt from entering the centre bearing.

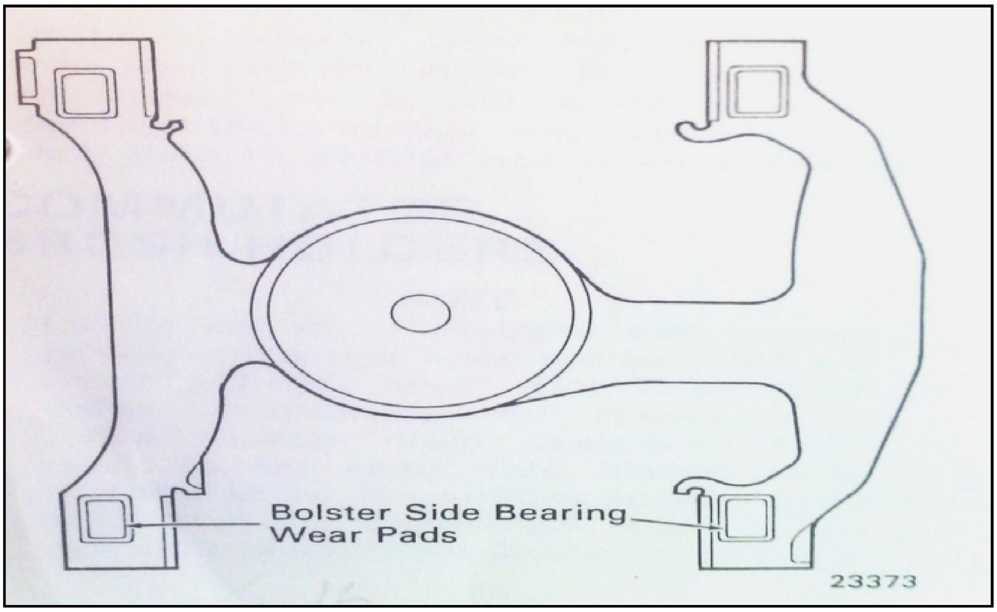


Fig (2.3) Bolster Assembly.

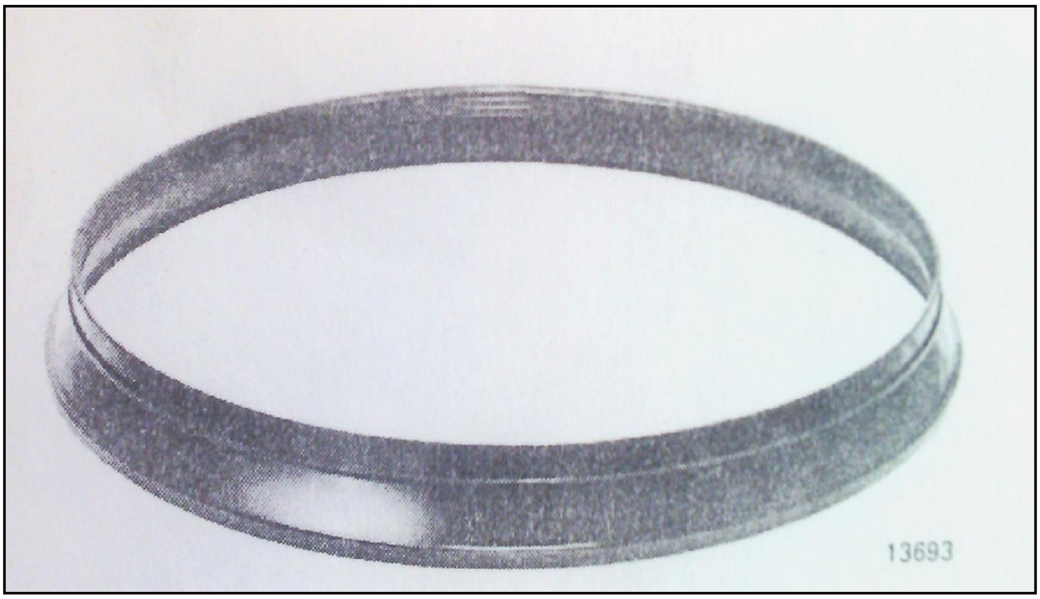


Fig (2.4) Center Bearing Dust Guard.

Side Bearing Wear Pads

The side bearing surfaces on the bolster are designed to mate with similar side bearings mounted beneath the carbody under frame as indicated in Fig. (2.5).

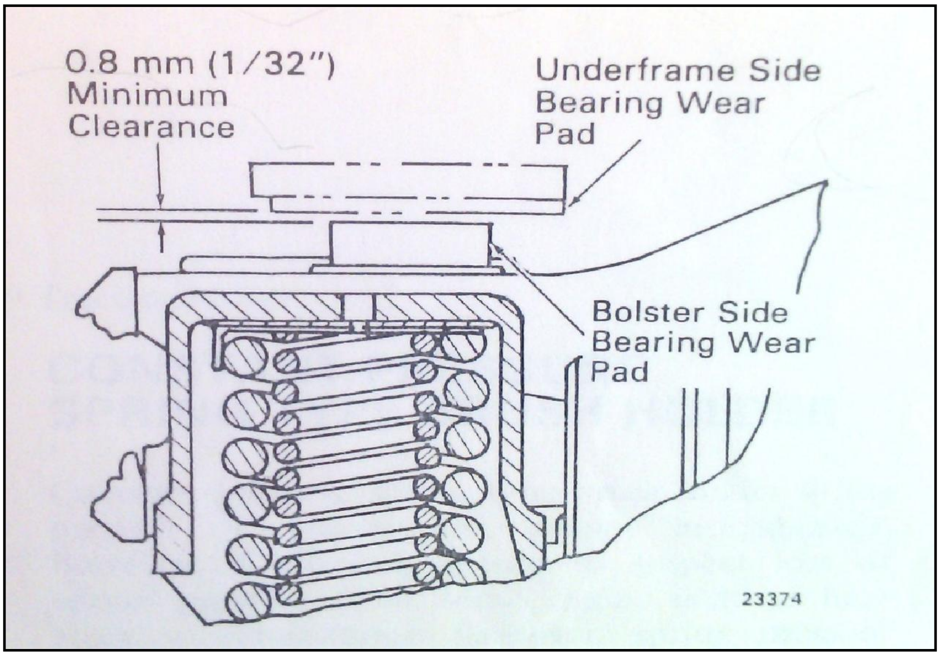


Fig. (2.5)Bearing Wear Pad clearance.

A clearance is provided between the truck bolster side bearings and the carbody side bearing during normal operation. Side bearings are designed to prevent excessive tilting or leaning of the locomotive but are not designed to carry a continuous load. On new assembly, clearance at any one pair of side bearing were pad across the locomotive centreline or diagonally opposite should be 4.0 mm (5/32″) minimum to 10.3 mm (13/32″) maximum. The other pair can be 4.0 mm (5/32″) minimum to 12.7 mm (1/2″) maximum.

Any time the side bearing clearance approaches the minimum limit, Fig. (2.6), the bolster centre bearing wear plate should be checked for wear. The side bearing wear pads should be flat and in the same plane within 0.8 mm (1/32″) as the side bearing wear pad on the opposite side of the truck. If wear pad is misaligned or is uneven, it may be repaired by building up the surface with weld and grinding to proper level.

c) General Electric Locomotives

GE “Floating Bolster” Bogies [3]

(General Electric (USA), locomotive bogie).

General Electric Universal diesel-electric locomotives have floating bolster trucks. These modern trucks eliminate the wearing parts required for lateral motion on typical swing bolster trucks, and at the same time provide riding qualities equal to or better than obtained with the swing bolster design.

The floating bolster rests on four laminated curved rubber pads which provide the lateral motion essential for good truck performance without the pins, swing hangers and gibs found on swing hanger trucks. When lateral displacement of the locomotive cab occurs the rubber pads deflect in shear and because of their curved construction, at the same time cause a superelevation of the cab like that caused by inclined swing hangers. The combination of shear deflection and cab super elevation creates a restoring force which corrects the lateral displacement and returns the cab centre. The resistance to lateral motion is graduated, so that greater the movement, the greater the restoring force. Long lived friction snubbers are used to prevent unwanted lateral oscillation.

Vertical springing is provided by soft coil springs which rest directly on the roller bearing journal boxes. Friction snubbers of the same type as lateral snubbers are used to check vertical oscillation. This spring arrangement eliminates equalizers, which can be a source of maintenance difficulty, and thereby provides easy access to brake shoes.

The floating bolster design offers three advantages of extreme importance

High adhesion factor

The floating bolster, which rests on rubber pads on the truck frame, has a large diameter centre plate. The rubber pads, while they are soft horizontally in order to provide good lateral characteristics, are stiff vertically. These two factors combine to prevent tilting of the truck frame, and the weight transfer due to tractive effort is thus minimized. General electric locomotives can therefore exert high running adhesions.

Excellent ride and tracting

The use of rubber to cushion lateral forces and long travel vertical coil springs for vertical springing, with snubbers to damp horizontal and vertical oscillation, provides excellent riding qualities for mainline railroad service.

Reduced maintenance

The floating bolster design reduces maintenance because it eliminates most of the wearing parts of older design swing link type or multi-pivot rigid bolster type trucks. In addition the rubber pad suspension of the floating bolster truck eliminates vertical steel–to-steel contact between the bolster and truck frame, thus isolating the locomotive platform from track vibration and giving longer equipment lift. The direct springing further eliminates wearing parts and provides excellent access to rigging and slack adjustments.

The excellent riding and tracking of these trucks minimizes wheel and track wear by reducing side forces due to impacts.

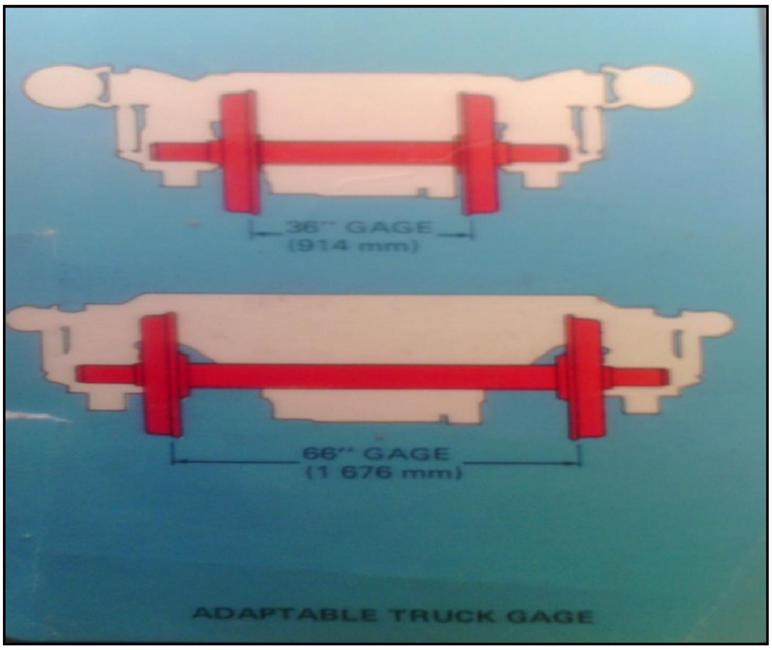


Fig. (2.6) adaptable truck gauge.

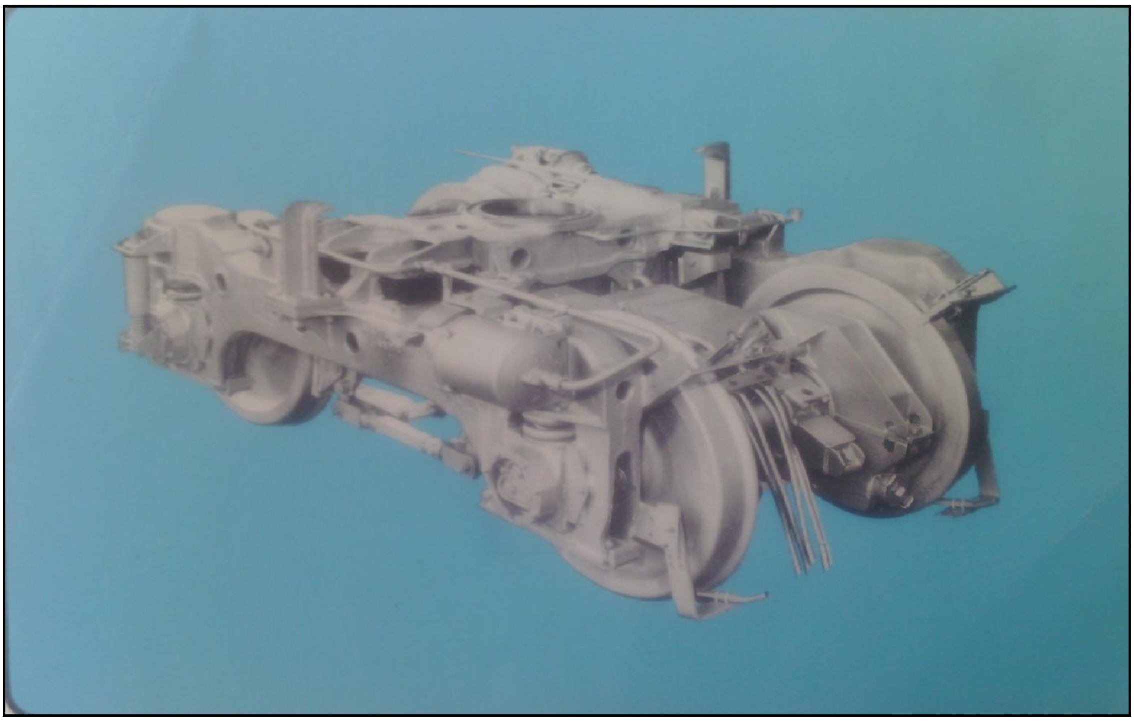
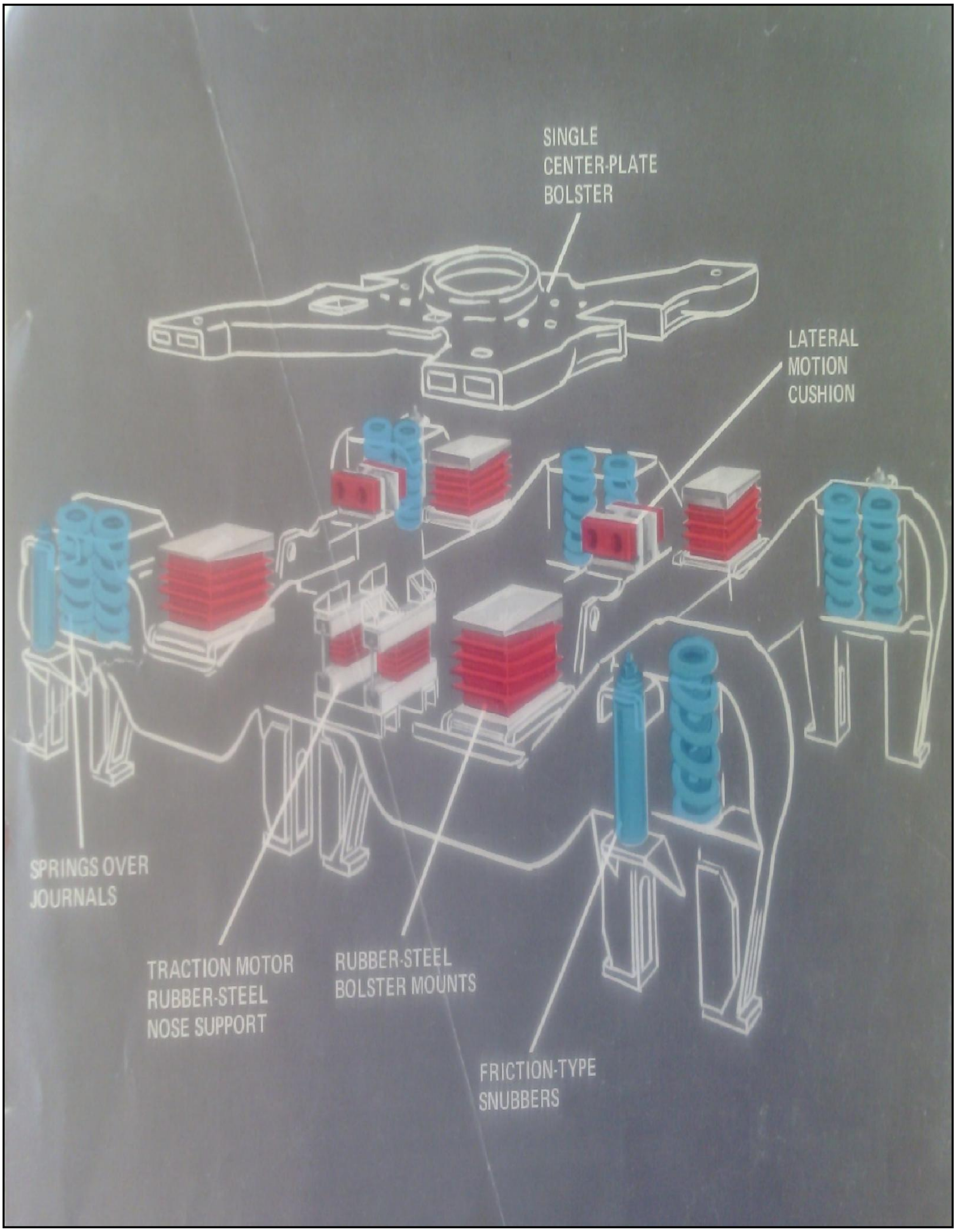
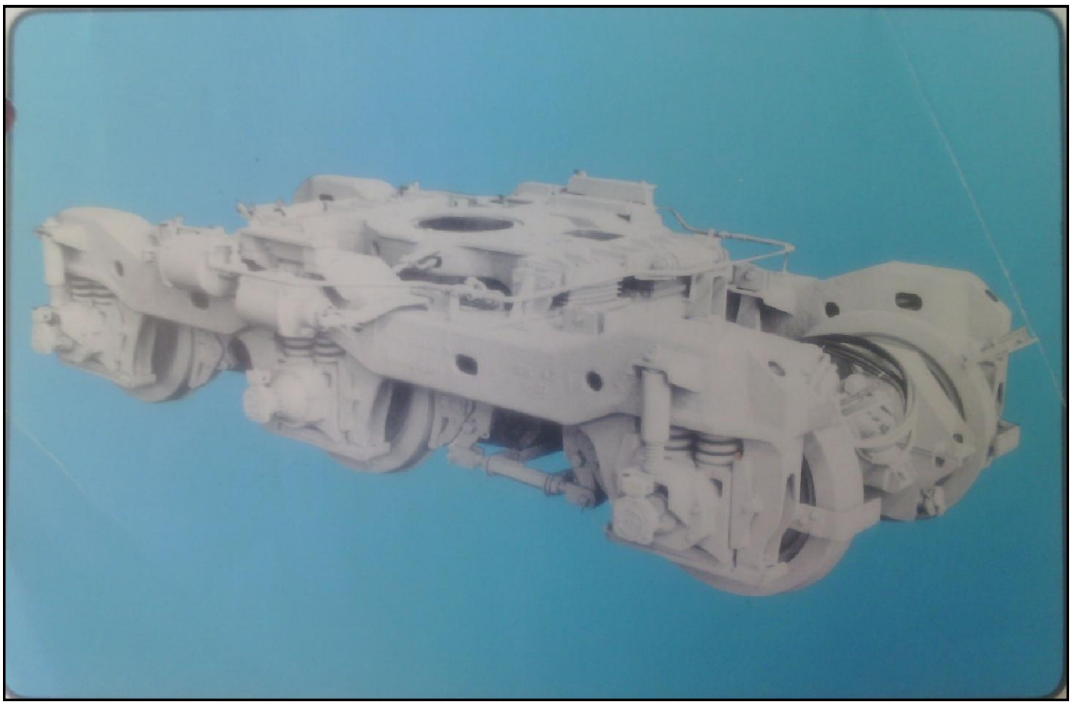


Fig.(2.7). bogie of locomotive (two axle).



Fig(2.8) springs & suspension components for the locomotive bogie



Fig(2.9) bogie of locomotive (three axle).

Fig.(2.10) rubber suspension

d) CRS of China locomotive bogie [4]

Bogie of SDD_1 Chinese diesel electric locomotive

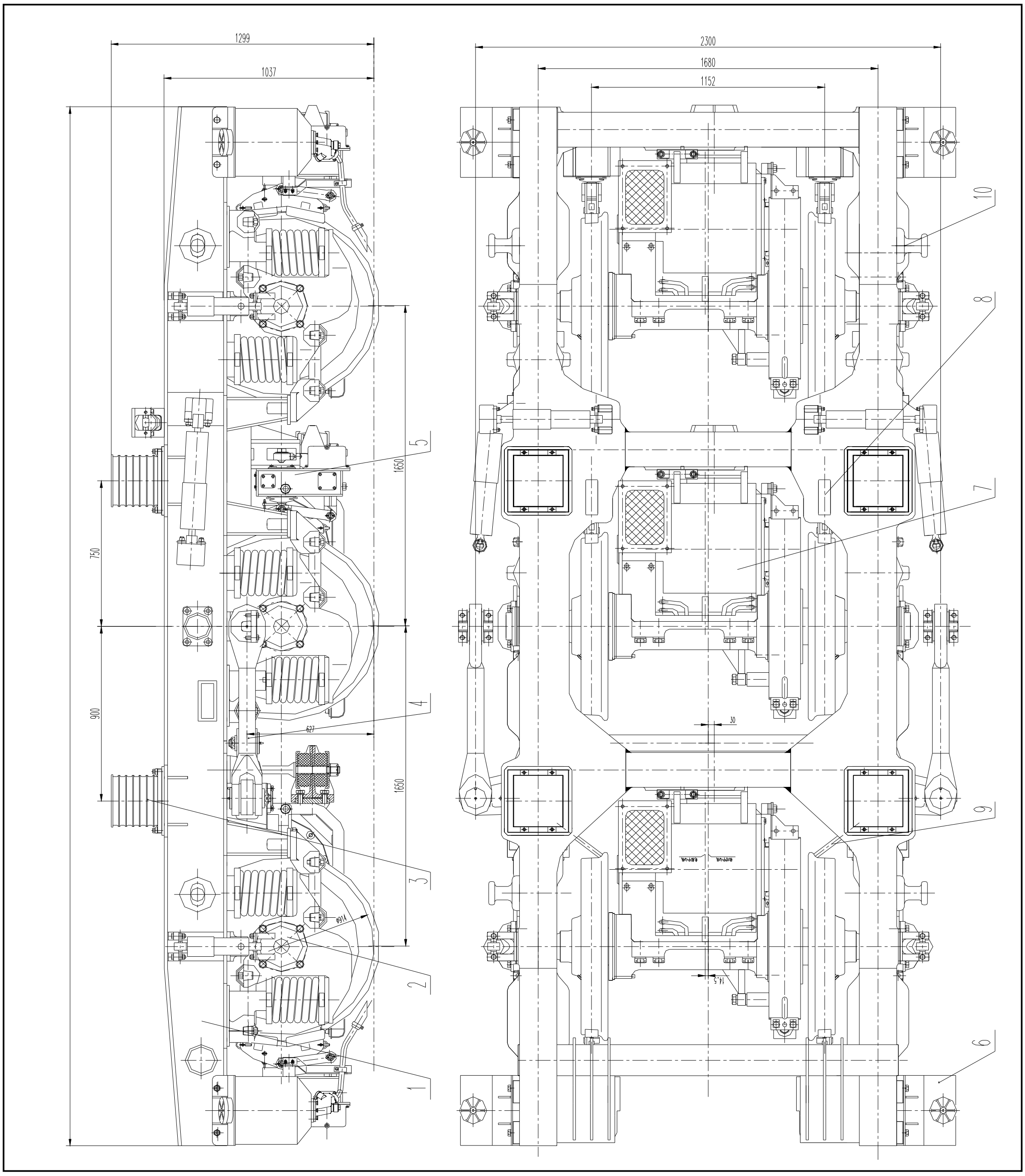
Main component and performance characters

Bogie of SDD_1diesel locomotive consists of bogie frame, journal box, wheel pair, side bearing, draft gear, brake rigging, sand box, motor suspension device, hand brake, lift devices and so on Fig.(2.11a). Main Technical Data:

Axle arrangement Co. Co

Wheel base2×1650mm

Axle load	16t
Max service speed	120km/h
Center distance of two bogies	9100mm
Bogie dead weight	16.5t
Weight under spring for each axle	3.055t
Wheel diameter 914mm (wear limited to 845mm)	
Min radius of curve	100m
Module of traction gear	7
Transmission ratio	93:18=4.6471
Total static deflection for spring suspension system	124+12=136mm
Static deflection of primary spring + rubber pad	122+2=124mm
Static deflection of secondary spring	12^{+2}mm
Primary vertical oil damper damping factor	60kN·s/m
Secondary lateral oil damper damping factor	60kN·s/m
Secondary anti-hunting damper damping factor	600kN·s/m
Free lateral play of bogie frame in respect to body	±20mm
Flexible lateral play of bogie frame in respect to body	±5mm
Flexible lateral play of journal box in respect to body to bogie frame	±(5,15,5)mm
Free lateral play of wheel pair in respect to journal box	±(0.5,0.5,0.5)mm
Traction point height above rail lever	627mm
Brake rigging	Type QB-2 & QB-2S Unit brake
Diameter of brake cylinder	177.5mm
Brake leverage	4
Braking ratio (at emergency brake)	0.593
Hand brake ratio	0.101



1.Bogie frame; 2.Wheel pair journal box; 3.Side bearing; 4.Draft gear; 5.Brake rigging; 6.Accessory assembly; 7.Motor suspension device; 8. Handbrake; 9.Wheel-rail lubricator; 10.Lifting device

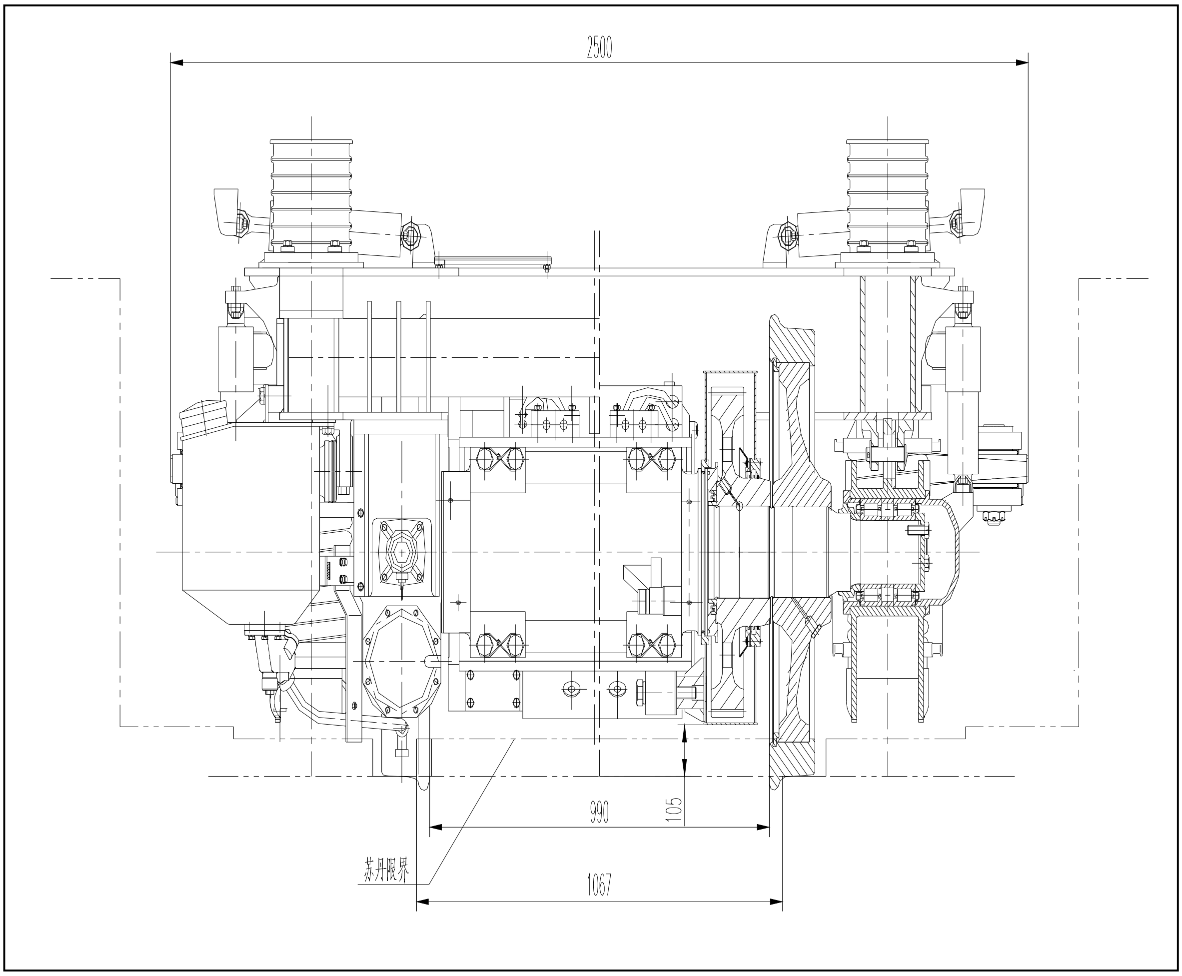


Fig (2.11a) Bogie of SDD_1 diesel locomotive

Operation and maintenance of bogie

1. Operation and maintenance of journal box

During operation temperature rise of journal box bearing is 30℃ and the max permissible temperature is 90℃. Locomotive monitor devices monitor the temperature, and send the temperature signal to computer through temperature sensor. If high temperature warning display, note the change of temperature. If axle temperature is obviously higher during operation, reduce speed operation must be done. Find out the causes after returning back to depot and can be used again only after taking necessary steps.

All tightening parts in journal box should not be loose.

2. Operation and maintenance of motor suspension device

1) All tightening parts should not be loose.
2) All components and parts of motor suspension should have no seizing, crack, serious damage, etc.
3) Motor vibration damping rubber pad should have no crack and defects. Bolts should not be loose.
4) Regularly fill oil for metal ball knuckle bearing on upper part of motor hanger (about 40,000-60,000km).
5) Check traction gear oil level during operation, it should be between upper and lower mark lines.
6) Always monitor the engagement of traction gear. Check gear surface if any abnormal noise. There should be no lack of oil and surface scoring.
7) Traction gear housing should be fast installed without breaking, damage and leakage.
8) Traction motor cooling air duct canvas pipe should have no damage. Clamping and ties should be securely.

9) Check whether the anti-dust rings on both ends of axle box are loosen. Axle suspension and suspension device should meet following requirements.

Both ends of axle suspension bearing box have oil filling hole, and fill lubrication grease per 150000 km as prescriptive oil type. During operation temperature rise of journal box bearing is 30□ and the max permissible temperature is 90. On both ends of axle hung bearing have temperature sensor to monitor the temperature. If axle temperature is over limited valve during operation, reduce speed operation must be done. Find out the causes after returning back to depot and can be used again only after taking necessary steps.

The axle clearance of axle hung bearing should be 0.10~ 0.25mm, during operation the max. clearance should be no more than 0.3mm. Otherwise make adjustment.

Check the connect bolt between axle box and motor should no loose, and the axle box should have no crank.

The hanger of motor should have no crank, and bolt have no crank.

Regularly fill grease to ball knuckle of motor hanger seat (about 40000 ~ 60000 km).

Bogie frame

Brief introduction

Main body of bogie frame Fig.(2.11b) adopts steel plates welded box structure. Within the box reinforcing plates are provided with to improve the rigidity and force loading. The bogie frame consists of two left and right symmetrical side beams, two same structure cross beams, front-end beam, rear end beam and all supports. All beams are welded with Q345C steel plates. Side beam top is welded with side bearing pad plate and lateral damper seat, on the side primary vertical damper seat, side stop seat, anti-hunting damper seat are welded. At the bottom is welded journal box upper and lower tie rod seat & draft gear bell crank arm seat. At both ends sand box supports are welded. At the bottom of cross beam, front and rear end beam are welded with motor hanger and brake seats.

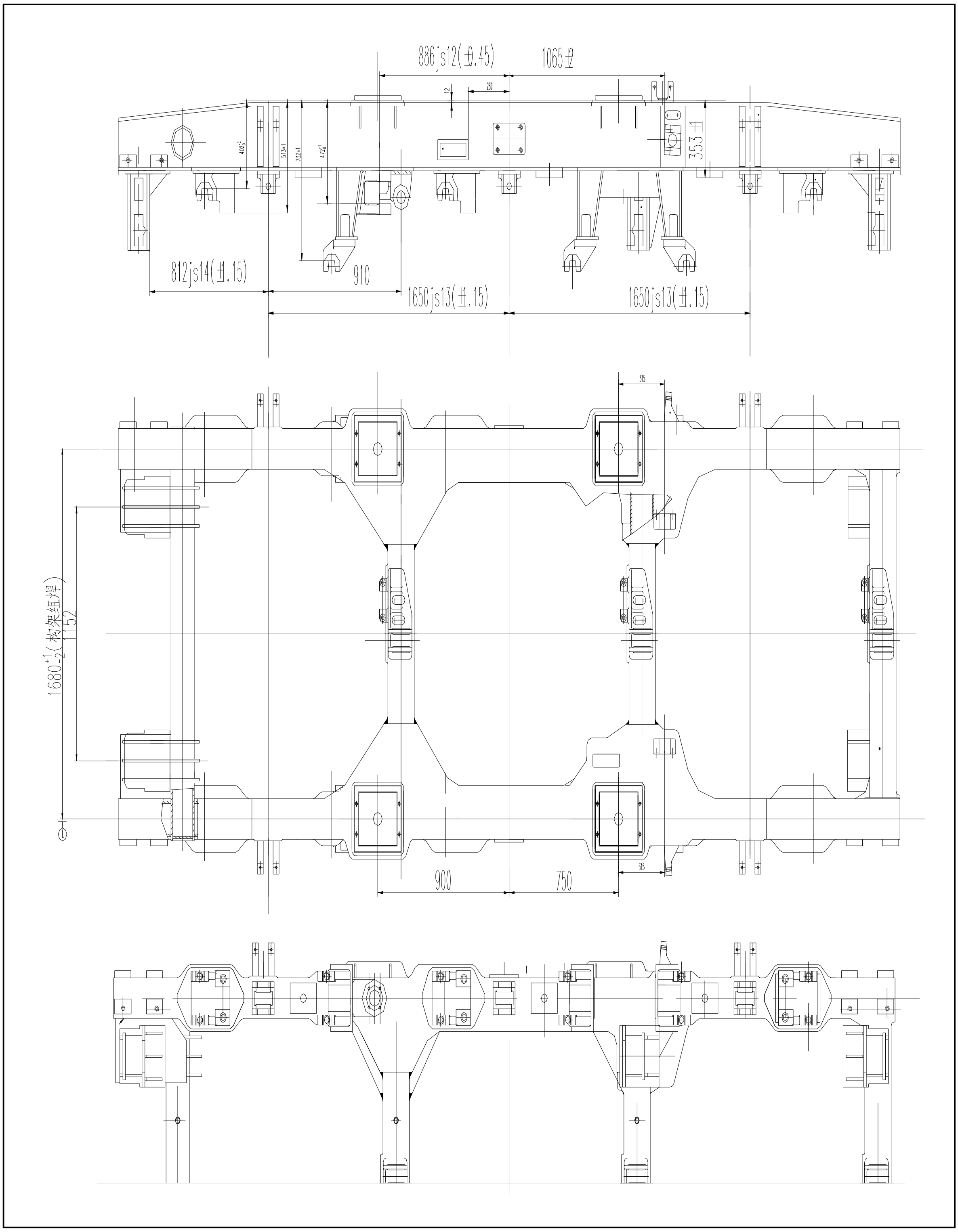


Fig.(2.11b) Structural frame.

Journal box

General

Journal box Fig.(2.12) is journal box tie rod location structure without guide frame. The journal box consists of journal box body, cylindrical roller bearing unit, primary spring, journal box tie rod, end cover, rear cover, dust ring and glands. Between end axle journal box and structural fame is provided with primary vertical damper. On the left side of No.3 position axle a DF16 (SS) speed transducer is installed. For primary spring assembly refer to Fig.(2.13) For journal box tie rod refer to Fig.(2.14)

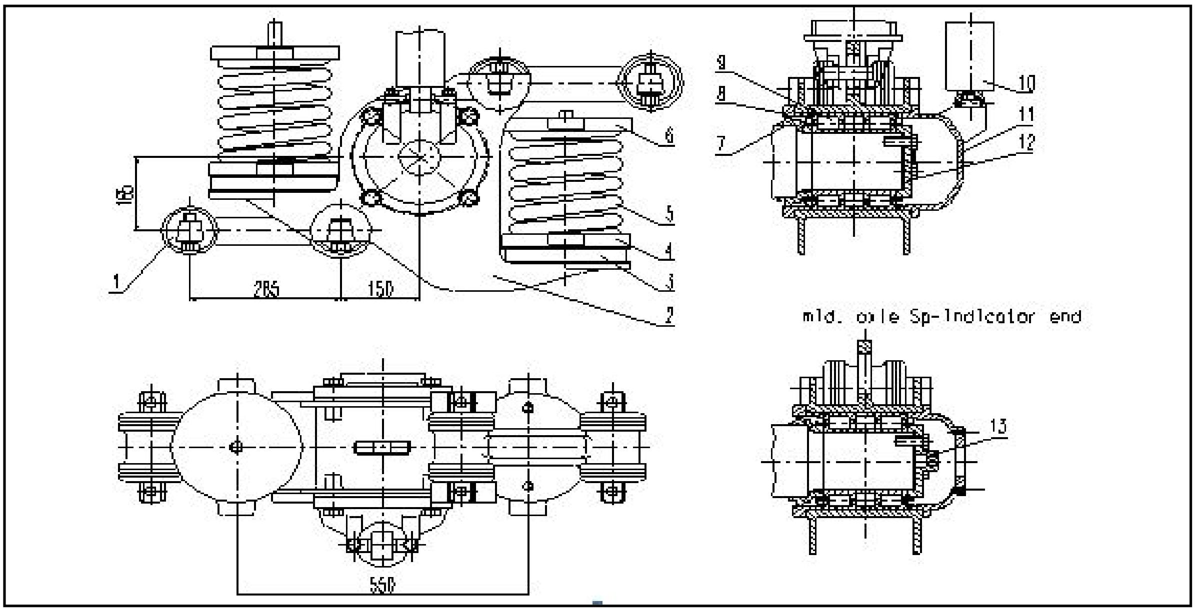


Fig.(2.12) Journal box.

1. Journal box tie rod; 2. Journal box body; 3. Damping pad; 4. Spring seat;

5. Helical spring; 6. Spring cover; 7. Dust ring; 8. Rear cover; 9. Bearing unit;

10. Vertical oil damper; 11. End cover; 12. Gland (I); 13. Gland (II).

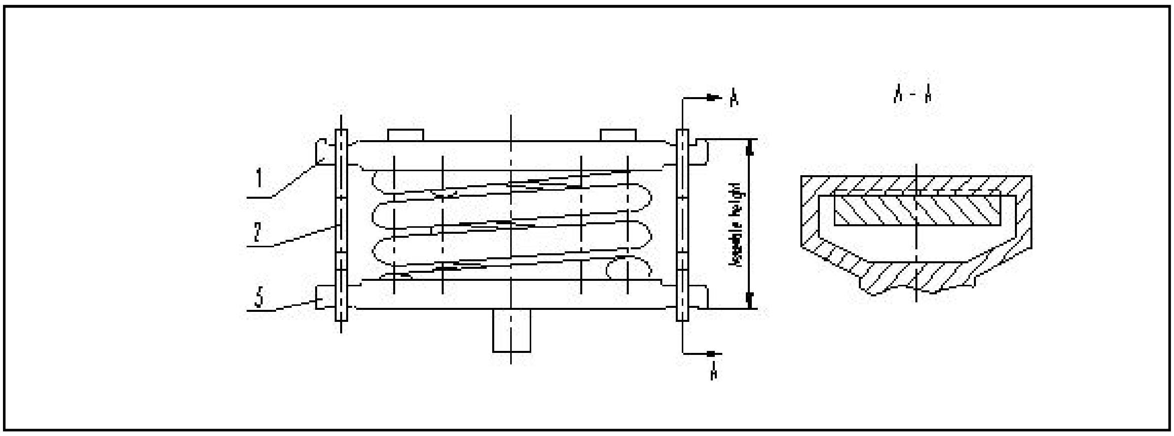


Fig.(2.13) spring Assembly

1. Spring cover. 2. Clamp ring. 3. Spring seat.

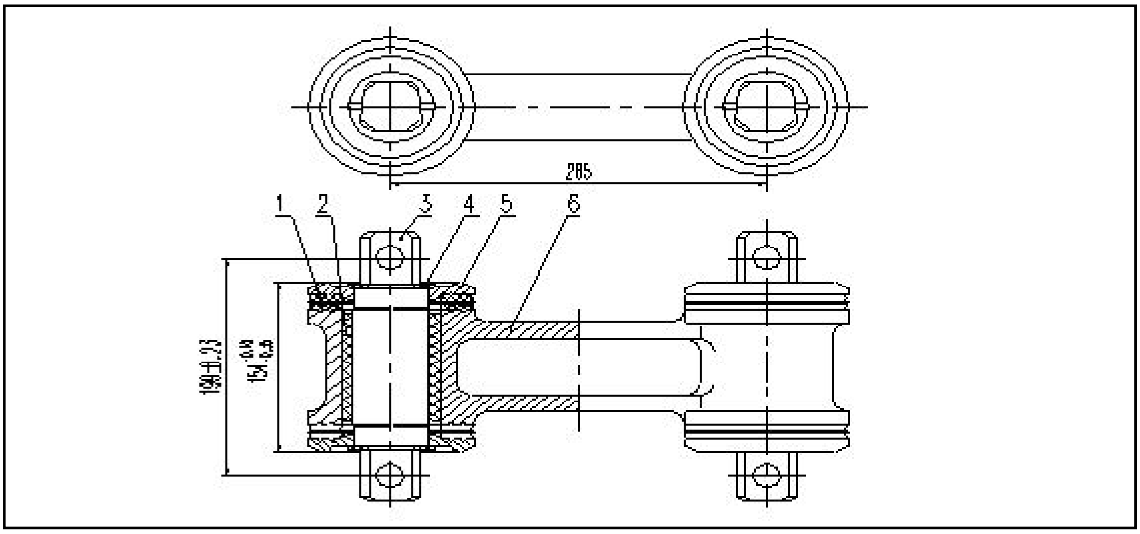
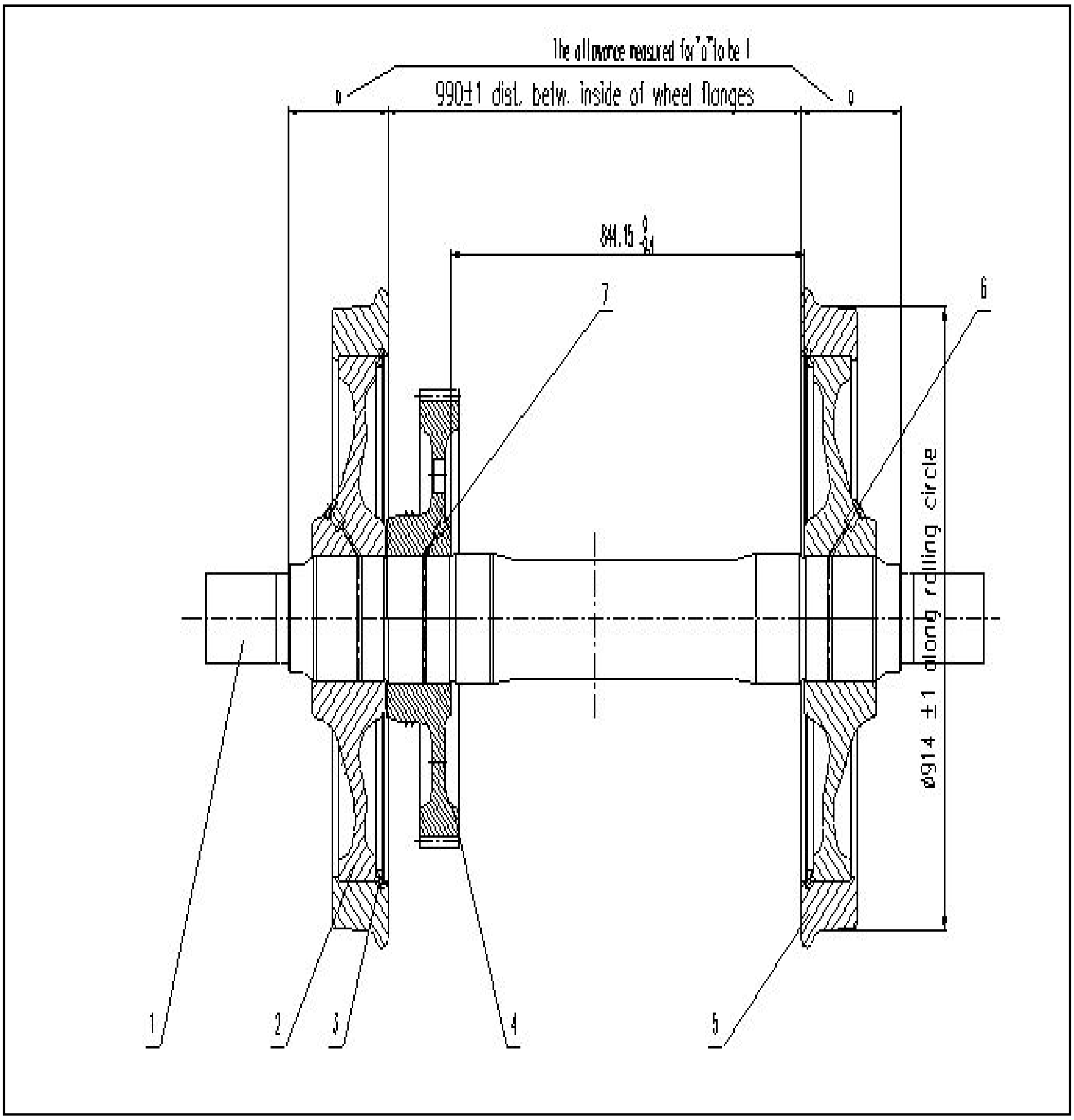


Fig.(2.14) Journal Box Tie Rod

1. End cover rubber; 2. Rubber ring; 3. Arbor; 4. Clamping ring; 5. End cover; 6. Tie rod.



Fig(2.15) Wheel Pair

1. Wheel axle; 2.Wheel core; 3 Retaining ring 4.Driven gear; 5.Tyre

Wheel pair

(1) Wheel pair consists of axle, wheel core, tyre, retaining ring, driven gear, etc. Fig.(2.15).

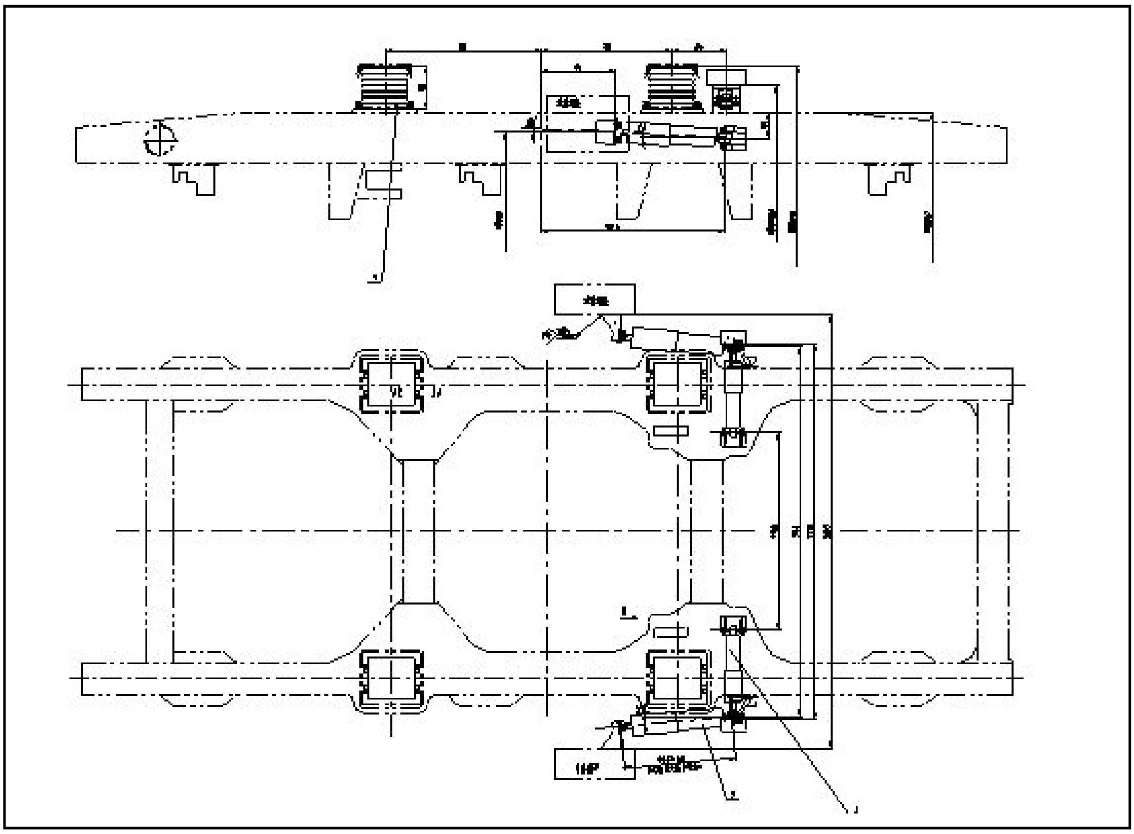
(2) Maintenance of wheel pair

- ✧ New wheel flange inside distance should be 990±1mm. Used wheel 990^{+2}_{-1}mm. Wheel flange inside distance difference in same wheel pair should not be more than 0.8mm (medium repair 1mm, the difference more than 1.2mm is forbidden).
- ✧ Run out of flange end to axle central line should not be more than 0.2mm.
- ✧ After turning of wheel, tread difference should not be more than 0.5mm, wheel flange height decrease should not be more than 1.0mm. Wheel flange thickness decrease should not be more than 0.5mm by means of template checking. A black of depth no more than 2mm and width no more than 5mm in 10~18mm area of wheel flange top. The service rejected limit of wheel is ф845mm.

Side bearing

General

Side bearing is the secondary suspension system between bogie frame and car body (See Fig.(2.16)). Type SDD_1 diesel locomotive bogie uses four point supporting rubber pad side bearing by which the transmission of static load of upper body, dynamic load and the lateral load of body and bogie can be obtained. Side bearing consists of rubber pad, secondary lateral oil damper and anti-hunting damper.



Fig(2.16) Side Bearing

1. Rubber pad; 2.Anti-hunting damper; 3.Lateral damper.

Draft gear

General

Traditional good performance parallel four link draft gear is adopted for draft gear Fig.(2.17). In draft gear mechanism one end of each traction rod is connected with traction support on bogie frame side beam through traction pin. Other end is connected with bell crank arm in bogie frame by means of pins. Left and right bell crank arms are connected via a link rod to ensure the synchronous function of left and right traction rods. Knuckle bearing is used to connect traction rod with traction seat on body and bell crank arm on bogie so that during the running of locomotive car body can move up and down, left and right in respect of bogie.

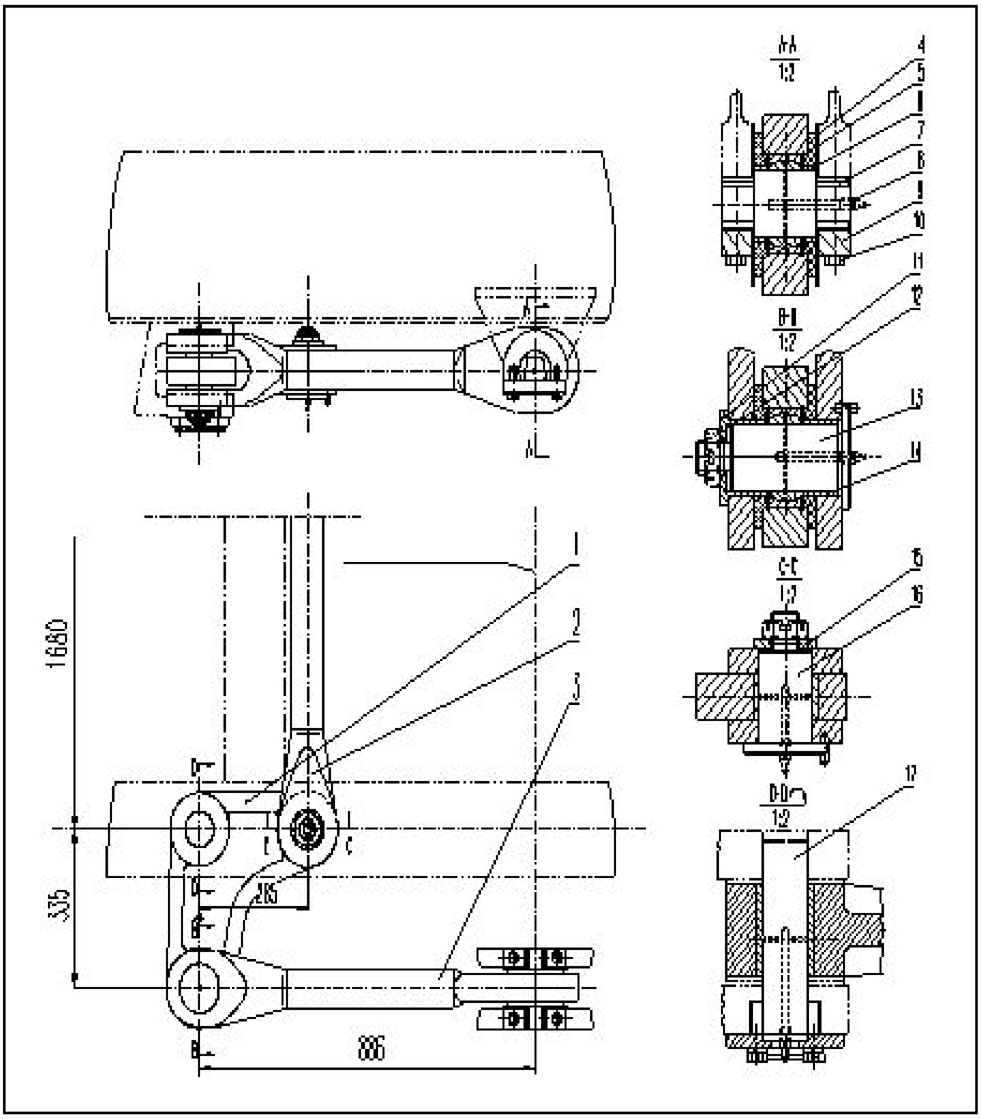


Fig.(2.17) Draft gear

1.Bell crank arm; 2.Link rod; 3.Traction rod; 4.Dust king; 5.Dust guard; 6.Spacer sleeve; 7.Traction pin; 8.Oil cup; 9.Surporting plate; 10.Bolt; 11.Gland; 12.Nut; 13.Traction rod pin;

14. Spacer sleeve; 15.Washer; 16.Link rod pin; 17.Crank bell arm pin.

Brake rigging

General

Six unit brakes Fig(2.18) with independent function are installed in each bogie. Unit brake, which can be connected with hand brake, is used on intermediate shaft of rear bogie. All wheel are braked by single brake shoe.

Each unit brake is installed with two-piece brake shoe, easy for changing and good for contacting and heat dissipating during braking. There are two kinds of brake unit: QB-2 AND QB-2S. The difference between them: QB-2 brake unite cannot connect with hand brake, while QB-2S brake unite can be connected with hand brake. Both of them are consist with brake cylinder assembly, box, adjusting devices of brake shoes clearance, bolt reset mechanism, brake head, brake shoe support, and brake shoe. Except level 14 is different, the left parts are all the same. Fig.(2.19) shows the QB-2S unit brake.

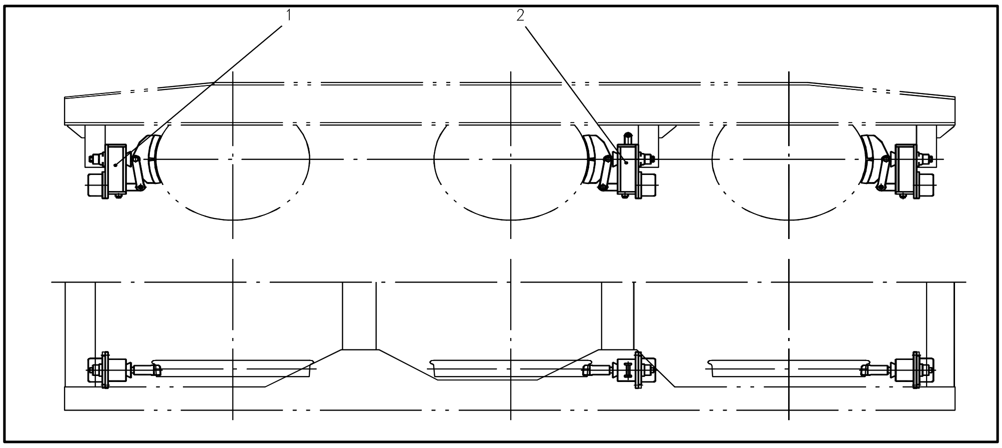


Fig.(2.18) Layout of Unit Brake

1.Unit brake not connected with hand brake; 2. Unit brake with connection of hand brake.

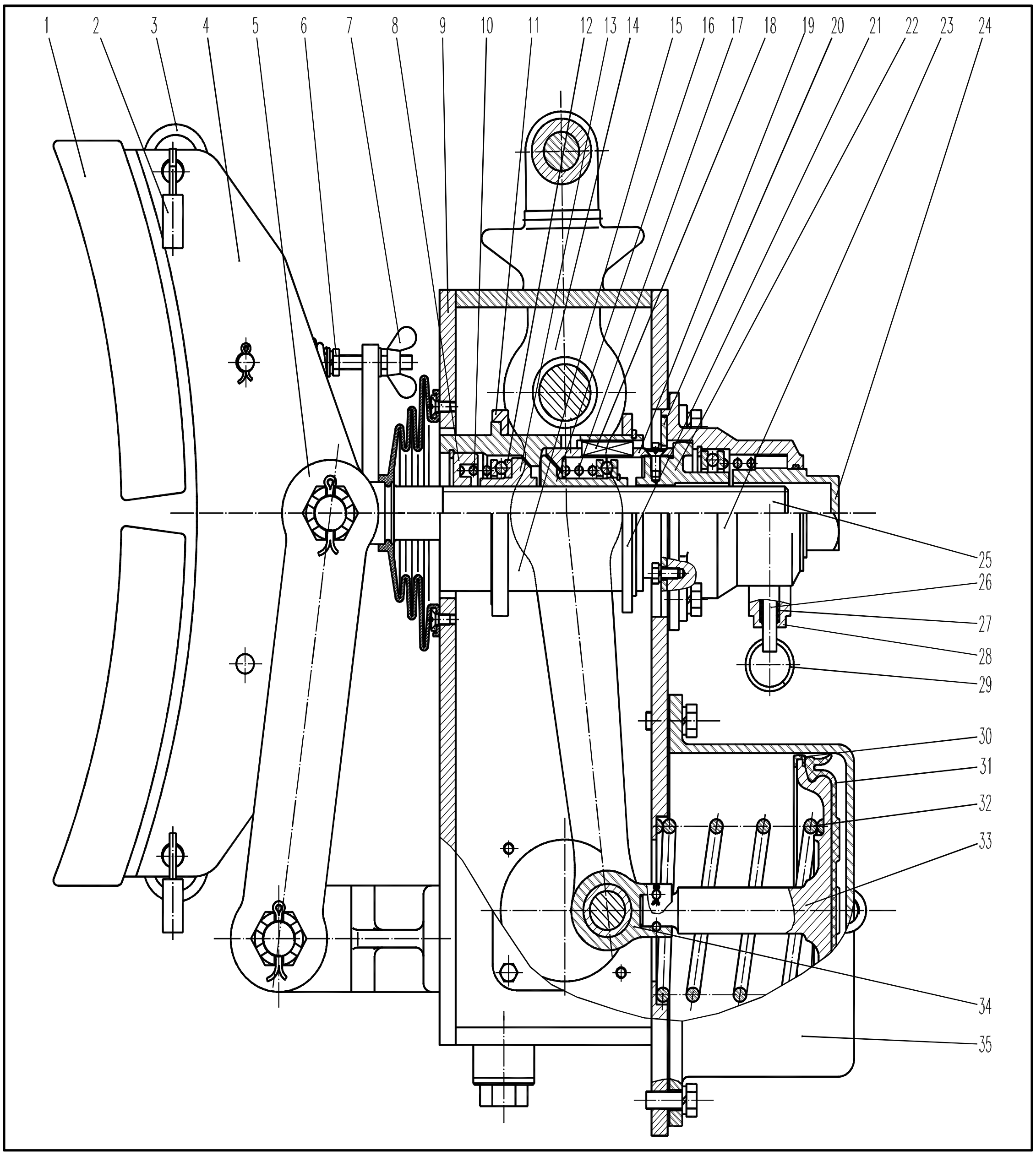


Fig.(2.19) QB-2S Unit Brake

1. Brake shoes; 2.Pin; 3.Brake shoe key; 4.Brake head; 5.Brake shoe support; 6, 7.Nut; 8.Guide sleeve; 9.Housing; 10.17.Adjusting spring; 11.Thrust retainer; 12.Bearing; 13.Adjusting Nut; 14.Lever; 15.Adjusting sleeve; 16.Guided nut; 18.Flat key; 19.Reset retainer; 20.Guide nut sleeve; 21.Spacer retainer; 22.Clamping ring; 23.End cover;

Motor suspension device

Motor suspension device Fig.(2.20) mainly consists of driving gear, traction motor, rolling box, motor hanger, air duct assembly and gear housing assembly. The device is featured by reliable operation, simple maintenance etc.

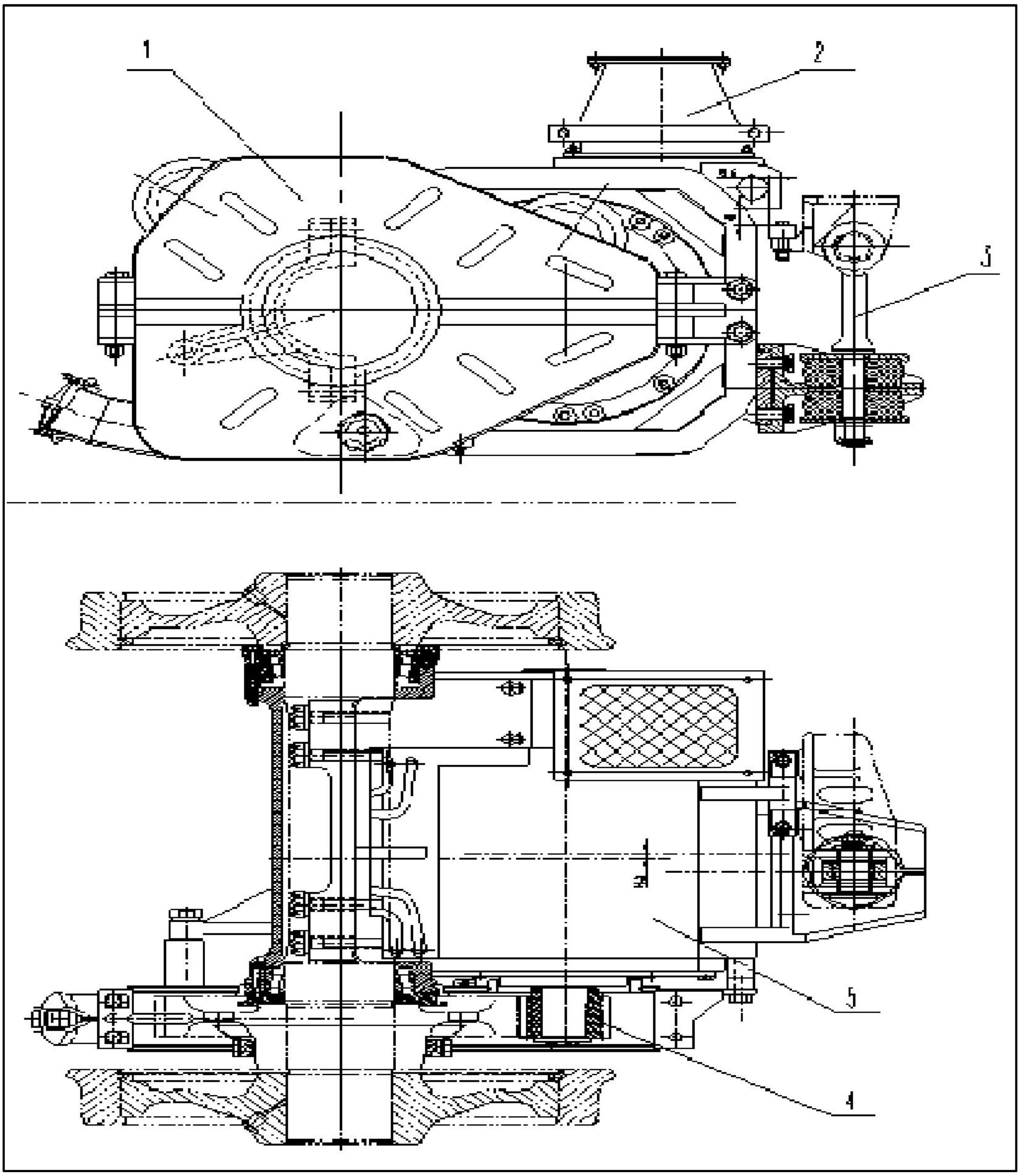
Inspection of traction gear backlash

- ✧ After assembly of traction motor with wheel pair and before assembly with gear housing No.12 fuse should be used to measure backlash of driving gear and driven gear.
- ✧ For new gear the backlash should be within 0.437～0.685mm (2.5mm at medium repair, 3.2mm is forbidden to use).

Running in test

Whenever reassembly is done for traction motor suspension running-in test should be done as per requirements so as to test the operation conditions of all bearings, traction gears and whole system.

- ✧ After assembly of motor suspension device all fitting positions must meet requirements and all connecting bolts must be tightened.
- ✧ The clearance between gear housing inside surface and motor output end surface should be basically uniform.
- ✧ Turning wheel pair by hand without seizing.
- ✧ DC Power cable connection of motor should not be loose.
- ✧ Test with power on, and let wheel pair operate at low speed to see if there is any abnormality. If there is no abnormality in 5 min, then rise output voltage, and let wheel pair operate at the speed no less than 300/m; If there is no abnormality, cut off the power soon, resolve the trouble, then go on to do running test.



Fig(2.20) Motor Suspension Device.

1. Gear housing assembly; 2.Air duct assembly; 3.Motor suspension; 4.Driving gear; 5.Traction motor.

- ✧ Keep wheel pair speed not less than 300rpm. Turn wheel pair in cw and ccw respectively for 30 minutes, when motor changes rotation direction, running-in test must be done after the full stop of motor operation. There should be no abnormal noise, collision, seizing and high local temperature rise in the running-in test. All moving parts are in good conditions. Driving and driven gears engage normally.

  After running-in test following should be checked:

- ✧ All connecting bolts should not be loose.
- ✧ Rolling axle bearing temperature rise should not be more than 30° and temperature should not be more than 90°
- ✧ Journal box bearing temperature rise should not be more than 30° and temperature should not be more than 90°
- ✧ Check gear housing. There should not be leakage (but one drop of slight seepage oil per minute is allowed).
- ✧ Take down the test records.

2.3.1The modern wheelset [5]

Synopsis

This study deals with the design and manufacture of railway wheelsets for modern operating conditions. The effect of modern machine tools upon the manufacture of wheelsets in relation to quality and reliability.

The factors to be taken into account in the design and workshop maintenance of wheels and axles are dealt with. Reference is made to the use of computer aids for the analysis of service stresses and to assist in the evaluation of new wheel design. Methods for the assembly and also aids to the dismantling of wheelsets are discussed. The benefits of correct transition radii and stress-relieving grooves upon service life are described as well as the effect of the position of the boss of the wheel in relation to the wheelset.

The refinement of the existing concept to provide a lighter wheelset as a means of offsetting some of the increasing costs of energy is reviewed.[5]

Introduction

The exacting requirements of current specifications and the improved methods of inspection and testing with which modern wheelsets and components must comply, necessitate the very best of manufacturing practices.

This study with some of the major factors which influence the designs of the modern wheelset and details how the latest machining and assembly techniques are employed to ensure successful quality and reliability in service.

It does not concern itself with the production of forgings prior to machining, nor does it seek to assess the effect of components such as gears, motors, bearings etc. which may be added during final assembly.

Factors influencing the development of the modern wheelset

The cumulative effect of induced mechanical, thermal and residual stresses have a profound effect upon the service life of a wheelset .These stresses are particularly prevalent in the design of the wheel.

Mechanical stresses at the tread

A serious effect of mechanical stress at the tread is SHELLING. This is a fatigue failure due to contact stresses and is to be found on wheels having high static or dynamic loads. The magnitude of this effect is directly related to the geometry of the wheel and rail, the vertical wheel/rail forces and speed.

A new wheel 916 mm diameter with standard contour supporting a 100 KN load on a new rail produces a calculated area of contact of about 159 mm^2. This results in a concentration of very high stress which is repeated with each revolution of the wheel and gives ideal conditions for fatigue failure.

The resultant contact stress , which can be computed mathematically, can be of such proportions as to cause the metal to yield plastically at the point of maximum shear stress, i.e. just below the surface, with small fatigue cracks below the surface which propagate towards the surface where a small piece of the tread metal may fall out fig (2.21).

Fig (2.21) shelling of wheel tread due to excessive loading.

The remedial measures that can be taken to reduce the incidence of shelling are:-

a) Reduce the vertical loads.
b) Eliminate the vertical impact forces between wheel which are features of wheel flats, wheel eccentricity and imbalance, dipped rail joints, raised welds etc.
c) Increase the wheel diameter.
d) Use wheel material of higher fatigue resistance.

Points (a), (b) and (c) above require engineering design and point (d) requires metallurgical development.

The cure for shelling lies primarily in the hands of the user, but the wheel manufacturer is assisting by constantly striving to produce optimum wheel material by changing the analysis or heat treatment but without worsening the steel's response to thermal effects.

Mechanical stress in the web

In any consideration of wheel strengths, attention must be paid to the behavior of the wheel web under high fluctuating stresses. The rim width and thickness, hub diameter and length and position of the rim relative to the hub are wheel parameters dictated by service requirements and leave little room for manoeuvre. The web shape, however, can take on many forms and, up to quite recently, has been developed through intuitive design. Nevertheless, despite this seemingly haphazard approach, wheels were produced which proved adequate for operating speeds up to 160 km/h.

With the advent of high speed trains, operating at speeds in excess of 200 km/h, it became increasingly clear that greater attention would be required to the design of wheels in order to ensure that the more rigorous duties did not increase the risk of failure. In this respect, it is indeed fortunate that the development of high speed trains coincided with a rapid growth in computer techniques.

Calculation of stresses due to various loadings are now possible by computer program and the reliability of the predicted stress has been proven by strain gauge measurements and other practical tests. The method however is complex and time –consuming, particularly in the initial stages of idealizing the wheel as a finite element network.

With the object of determining the best basic form of wheel web, detailed stress analyses have been under-taken on varying web shapes using the finite element method assuming various combinations of vertical and lateral mechanical loads based on service experience to represent the normal duties of a 20-ton axle load freight vehicle wheelset operating at speeds up to 120 km/h. thermal stresses were also calculated for both severe drag braking and normal braking to stand still using conventional tread brakes. The designs considered which are shown in Fig(2.22) are based on monoblock wheel designs used by various European rail way administrations.

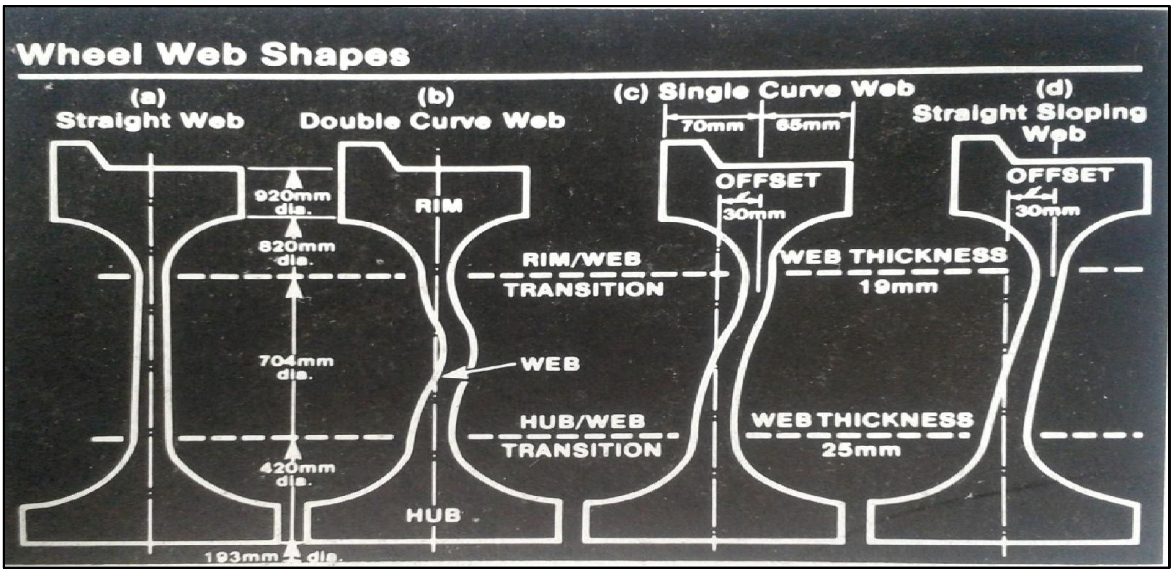


Fig (2.22) Wheel web shapes.

When subjected to a lateral load, the maximum bending stress in each wheel occurs at the hub/web transition, irrespective of web shape.

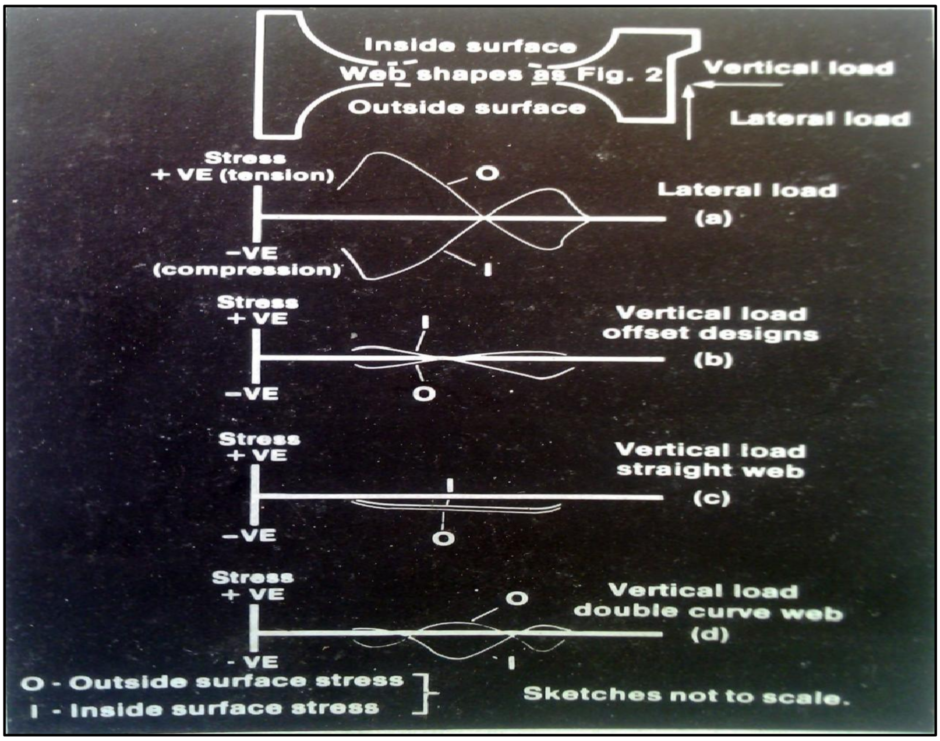


Fig (2.23) Radial stress curves resulting from various combinations of mechanical loading.

The shape of the web, however, has a significant effect on the magnitude of the maximum radial stress induced by the load Fig. (2.23).Wheels without offset, i.e. straight web and double curved web, have almost identical maximum stresses. Similarly wheels with offset, i.e. single curved web and straight sloping web, have a similar order of stresses, but they are somewhat lower than for designs without offset.

Vertical loading

The shape of the radial stress curve due to centre vertical loading depends largely on web shape. Design with a large web offset have stresses similar to the form produced by lateral loading but of much lower magnitude. A straight web gives an almost uniform compressive stress through the whole section along the loaded radius since there is little or no bending.

A double curved web has three stress peaks, two of which are at the hub/web transition (as for offset designs) and a third at the mid-web maximum stress for the design but the magnitude of 40 N/mm^2 for 110 kN load is much smaller than the maximum stress-about 150 N/mm^2 – induced by a lateral load of 60 kN.

It should be mentioned here that the standard loads used in calculations are 110 KN vertical and 60 KN lateral. These are considerably in excess of normal loads experienced in service which are about 50 KN vertical and 8 KN lateral. Occasionally, however, such gross loads occur when striking rail irregularities at speed.

The maximum radial stress determined by actual experiment are given in Fig(2.24) and clearly indicate that a typical lateral load gives rise to higher stresses in the web than a centre vehicle load. It must, however, be remembered that a lateral load occurs far less frequently than a vertical load. These experimental values agree closely with values calculated by the finite element analysis.

As the wheel rim wears in service, the general stress patterns maintain their similarity but the magnitudes are greater. At scrapping size the maximum stresses can be expected to increase by up to 25%.

Stresses from loads adjusted to the standard loads of 110 KN vertical and 60 KN lateral are shown in brackets.

Wheel Shape	Vertical Load KN	Max. Radial Stress from Vertical Load N/mm²	Lateral Load KN	Max. Radial Stress from Lateral Load N/mm²
Single Curve dia. 1000mm offset 35mm web.th.40mm	270 (110)	70 (29)	90 (60)	105 (70)
Straight Sloping dia. 840mm Other dimensions not given.	270 (110)	130 (53)	90 (60)	200 (134)
Double Curve small offset dia. 914mm web.th.24mm	89.5 (110)	35 (47)	57.7 (60)	150 (156)
Double Curve no offset dia. 920mm web.th.25mm	100 (110)	40 (44)	60 -	124 -

Fig (2.24) Maximum radial stress measured in various experimental investigations.

Although the foregoing deals with radial stress, similar consideration apply to hoop stresses induced by the mechanical loads, but in general these are of a lower order than the radial stresses. As the wheel rotates under the influence of constant mechanical loads, the maximum stress range in generally between the loaded radius (0 °)and the opposite radius (180 °). The actual stresses are maximum at 0° decreasing to zero at around 90° and changing sign to reach maximum again at 180°,but the absolute value at this position is always lower than at 0° Fig. (2.25).

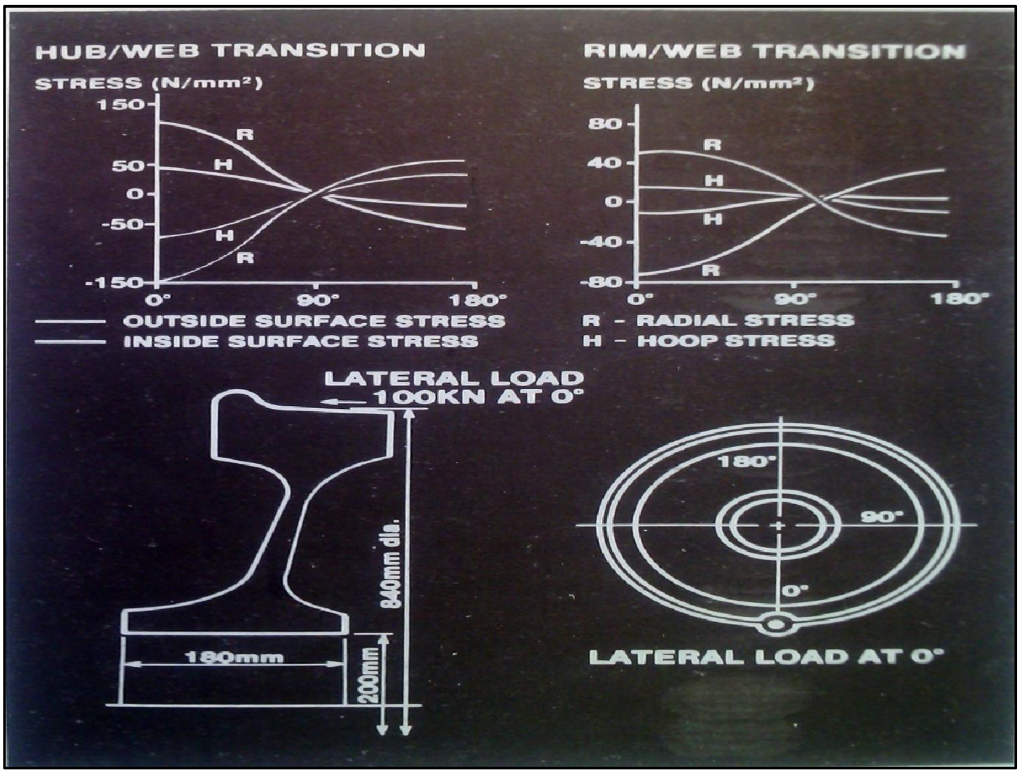


Fig (2.25) Variation of stress around the wheel due to a lateral load.

Thermal stress

Thermal load due to tread braking

In addition to its primary task of carrying load, in most cases the wheel also has to act as a brake drum and the frictional heat transferred to the tread tends to expand the rim radially. This expansion is restrained by the web which is at a lower temperature consequently stresses are produced in the web. These stresses are much higher than those induced by mechanical loading. The magnitude of the stresses and the position of the maximum stress is directly related to the geometrical shape of the web.

Drag braking, when temperatures can reach the 600-700 °C range, produces higher web stresses than stop breaking where temperatures are in the range 100-300°C depending on speed and braking time.

The effect of web shape on the radial stress patterns due to severe tread braking is shown in Fig(2.26) it will be seen that the straight web designs have almost uniform tensile stress through the web while offset designs have stress peaks at both hub/web and rim/web transitions. In effect they are as for a centre vertical load, but with the rim expanding outwards they are of opposite sign and of much higher magnitude.

Under these severe conditions it has been found that the peak stresses can exceed the material yield stress.

In addition to radial expansion, the restrained rim moves axially relative to the hub and this movement is influenced by the radial stiffness of the wheel.

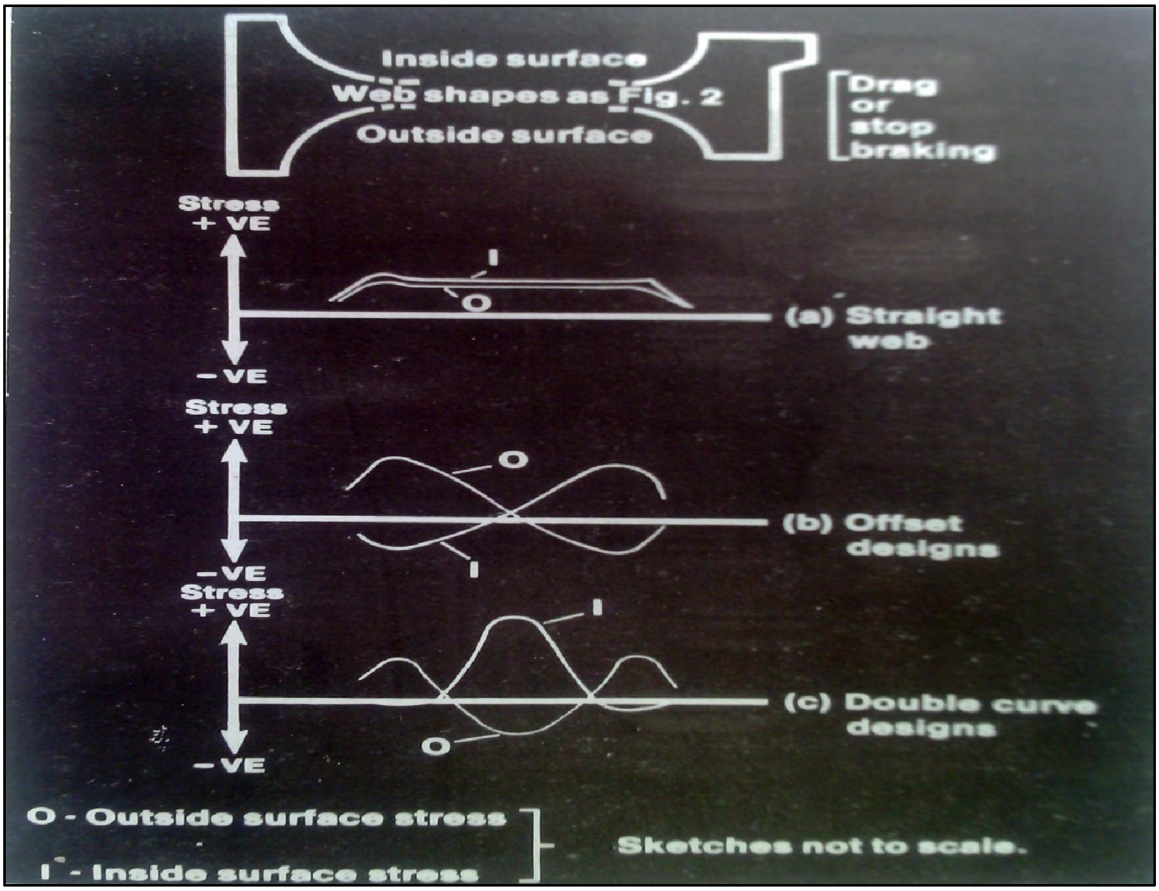


Fig (2.26) Radial stress curves resulting from the thermal effect of tread braking.

A comparison of the behaviours of the various wheel shapes due to this effect shows that designs with an offset suffer much higher lateral deflections (average 2.5 mm) than do designs without offset which deflect less than 1.0 mmFig.(2.27).

From this it is clear that a wheel design having a large offset is to be avoided where severe braking will be encountered in order to minimize axial movement of the rim.

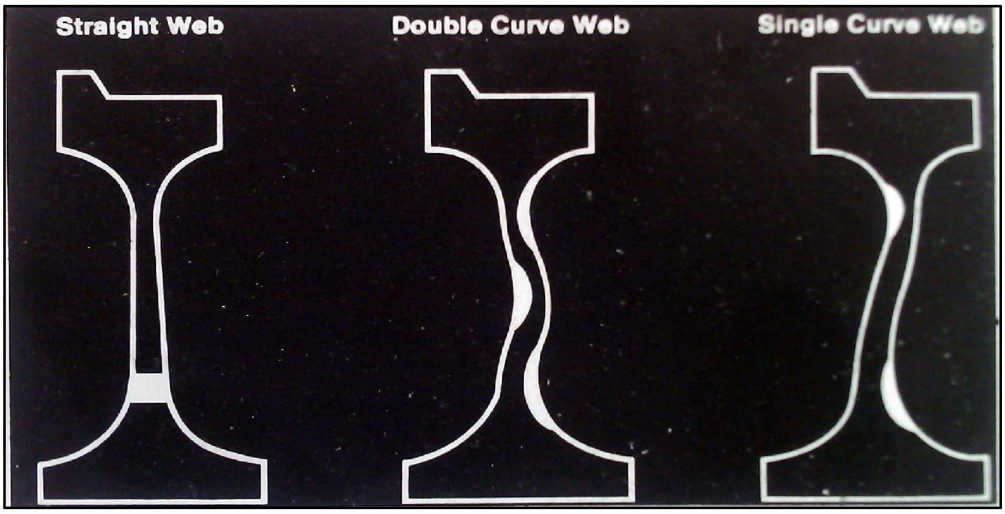


Fig (2.27) Area of web likely to yield first under thermal loading.

A permanent deformation of the rim can result after the cooling of the wheel following severe braking.

Stress in rim

Compressive hoop stresses in the rim due to prolonged braking can be sufficiently high to cause yielding at the surface and hence residual tensile stress when the wheel cools down. Repeated heating and cooling cycles under these conditions increase the tensile stress leading to thermal cracks with a possibility, eventually, of transverse fatigue cracks and spalling where layers of the surface material, perhaps 2mm thick, become detached.

The desirable factors of an optimum wheel design are contradictory and so it is not possible to achieve in one design all the features necessary to overcome all the service problems. One must consider the most important requirements for a particular service then reply on the recommendations dictated by experience and research.

Designers, however, should always bear in mind that even small changes in wheel form can have significant effects on the service life. For example:

a) Fillet radii should be as large as possible with a gentle transition from web to hub.
b) Volume of material in the rim should be to optimum proportions to resist twisting and beneficially affect the web stresses during prolonged tread braking.
c) Where there is a choice the largest wheel should always be used to reduce contact stresses and provide a better heat sink for heat transference.

Axles

It is generally accepted that it is a process of fatigue which cracks axles in service.

In the context of fatigue, the major design criterion is shape and since fatigue is a localized effect, the problem finally reduces to one of eliminating or reducing discontinuities in local shape, i.e. stress concentrations.

Stress concentrations do not affect the static strength of ductile materials but they do have a profound effect on the fatigue strength under repeated loading.

The most common stress concentration areas are: a step in the axle and an interference- fitted component on the axle.

A sharp change in section produces a much greater stress peak Fig.(2.28) than does a gradual transition from the large diameter -e.g. a raised seating down to the smaller diameter.

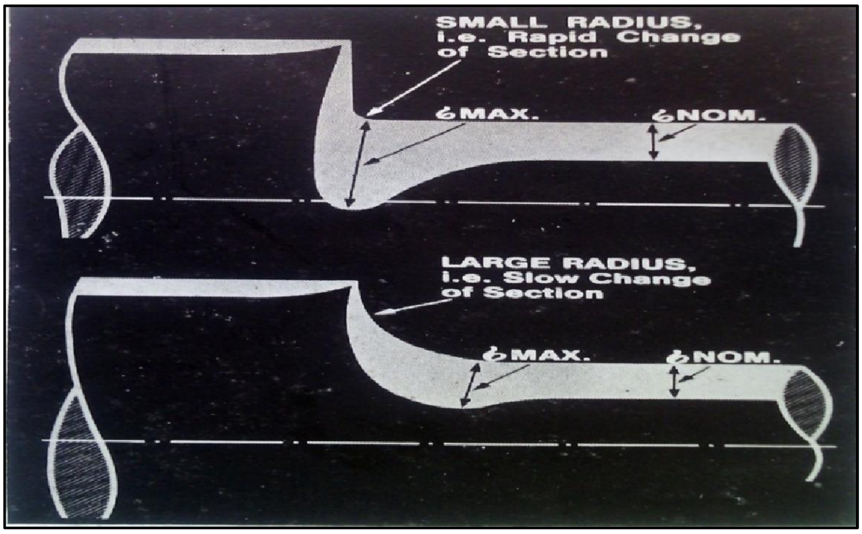


Fig (2.28) Effect of change of section on the stress concentration.

The ideal transition curves for stepped axles, which result in low stress concentration factors, are shown in Fig.(2.29) the importance of these features is that the ends of the transition curve should be tangential, both to the smaller diameter and also to the transverse face at change of section. Unfortunately these ideal transition curves cannot often be achieved in practice because they necessitate a difference of about 50% in shaft diameters. Actually for adequate fatigue strength an axle having a diameter difference of 15% to 20% is quite satisfactory. The final practical solution, therefore, is a wheelset 15% to 20% greater in diameter than the axle body and connected to the latter by means of a transition curve which may be parabolic, elliptical or consist of two conveniently sized radii, provided that the ends of the transition curve run out tangentially into the end faces of the wheelset and into the axle body respectively.

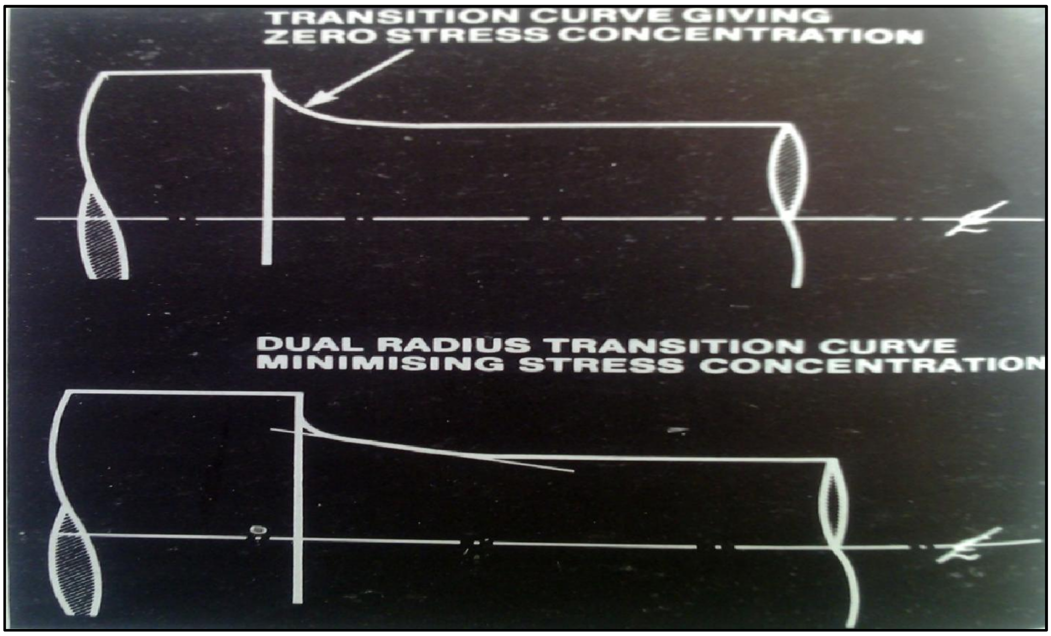


Fig (2.29) Ideal transition curves for stepped shafts under bending or torsion stress.

The geometry of the axle, however, does not always allow the use of a raised wheelset as a means of reducing stresses. A typical example is that of a driving axle with axle-hung motor. In this situation grooves can be used deliberately to counter the stress concentration effect.

(a) Between the road wheel and gear wheel seatings;
(b) Between the seating of the gear wheels and the adjacent motor suspension tube bearing and
(c) Between the seatings of the road wheel and the motor suspension tube bearing at the non-driving end of the axle.

The effect of a groove is two-fold: not only does it separate and prevent the interaction of two very close stress concentration effects of two adjacent interference-fitted components, it also reduces the peak value of each stress concentration; Fig.(2.30) shows a typical groove. From the illustration it can be seen that the groove is tending to simulate the improved stress distribution of a single raised wheelset with an adequate transition radius. Owing to the reduced cross section at the groove, the bending stresses tend to increase in this area, so to minimize this effect it is advisable to subject the groove to light cold rolling or burnishing or polishing.

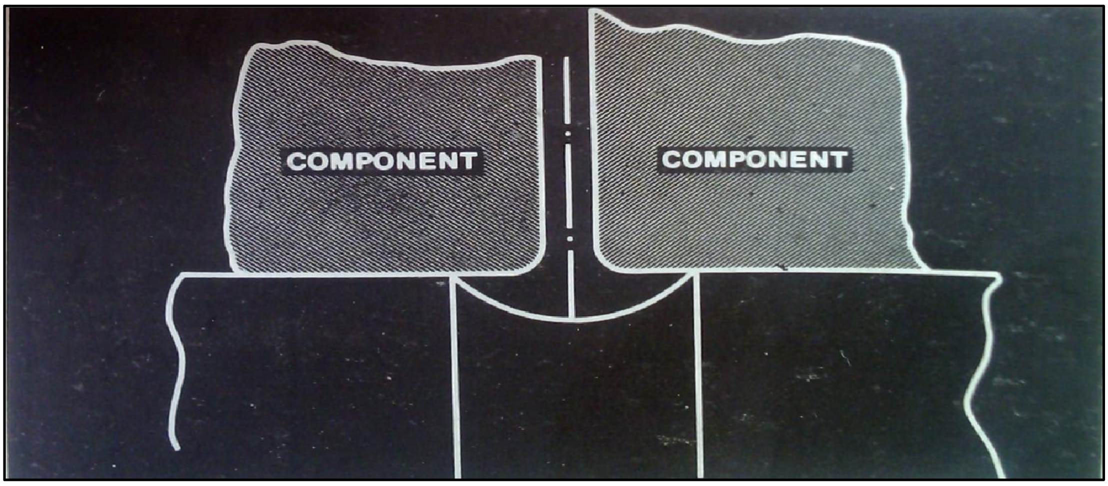


Fig (2.30) Stress relief groove under adjacent press-fits.

The use of stress relief grooves to reduce the stress concentration effects as described, is basic to the design procedure for motored axles.

Assembly of wheel and axle

Influence of interference-fitted components

As previously stated, the interference fitting of a component has a deleterious effect upon the fatigue strength of a plain axle due to the 'notch' effect. There are, however, other factors which must be considered. The fitting action produces a residual tensile stress in the longitudinal direction in the outer surface of the wheelset which can reduce the allowable dynamic stress range on the axle Fig.(2.31). In addition the position of the peak stress near the inner edge of the seating almost exactly coincides with the position of the geometrical stress concentration and the edge of the fretted zone. This provides a possible explanation of the fact that fatigue cracks in wheelsets usually are encountered just inside the inner end of the press-fit.

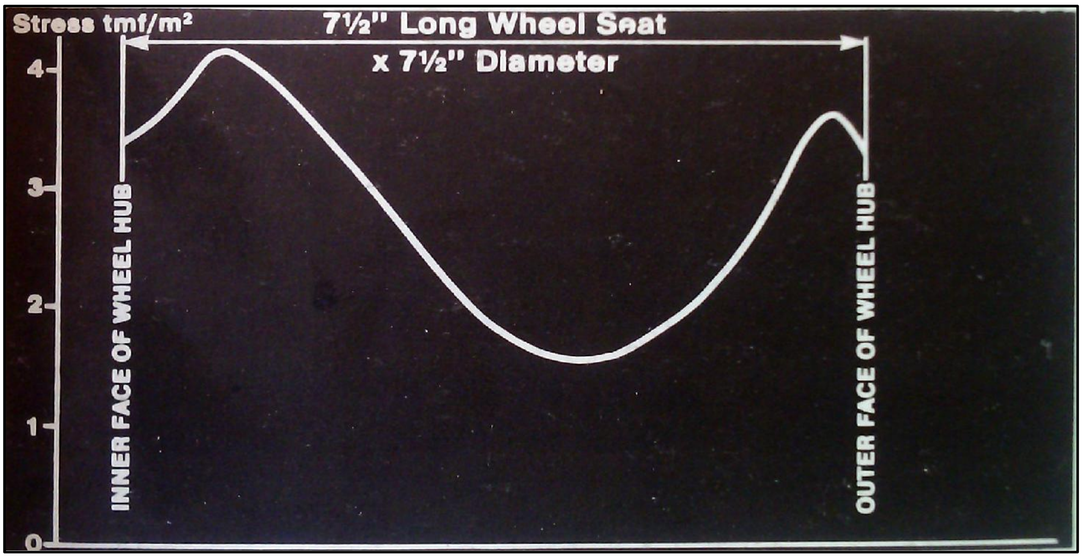


Fig (2.31) Distribution of longitudinal surface tensile stress in an axle wheelset due to the action of the press-fit alone.

Perhaps the main effect, which can cause drastic reduction in the fatigue strength of a component subjected to repeated load, is fretting corrosion.

Fretting is caused by repeated small relative movements between two surfaces loaded in contact. In its initial stages it is a purely mechanical process of making and breaking micro-welds across the axle-to hub interface during cyclic straining. As the process develops, the metal particles plucked from the surfaces at these small areas are disrupted, corrode. Surface damage is caused, producing small pits which result in points of initiation of fatigue flaws. Laboratory and service experience have established that there is a close correlation between the production of fretting and the initiation of fatigue cracks under an interference fit.

A geometrical feature which is widely accepted as a design improvement to limit the effect of fretting is to allow the wheel hub face to overhang the axle wheelset by about 6 mm. This is shown in Fig.(2.32).

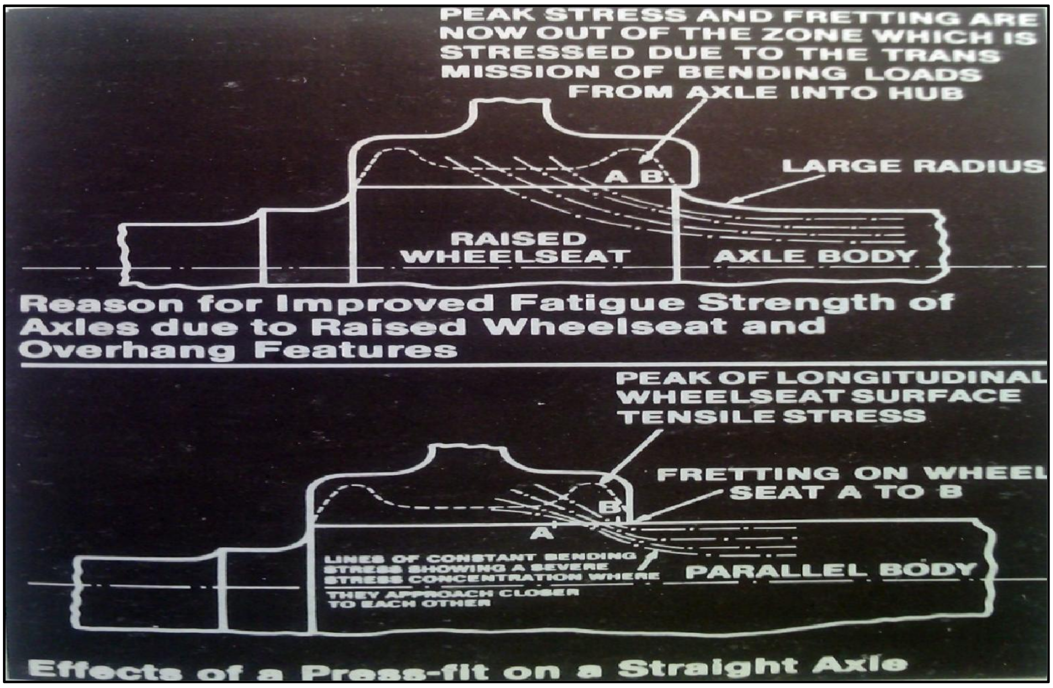


Fig.(2.32) Press-fitting effect on axle wheelset.

The wheel bore which fit flush with the axle shoulder tend to cone or plastically deform in the area immediately within the inside hub face due to alternating straining. The overhang feature overcomes this by moving the suspect area out of contact with the axle.

Surface treatment

In the vast majority of cases, fatigue failures are initiated on the surface of components, consequently surface treatments are sometimes applied to the axle wheelsets in order to produce a surface which is more resistant to fretting corrosion and which has greater hardness than the original axle steel. The hardness and strength of the surface layers are enhanced by inducing residual compressive stresses to counteract the tensile component of the applied bending stress. These effects are typical of surface treatments such as cold rolling and water quenching.

Summary of points basic to satisfactory axle design

(a) Account must be taken of any stress concentration effects.
(b) Raised wheelsets should be adopted wherever possible.
(c) Wheelset to axle body transition curves should be tangential to the transverse face of the wheelset as well as to the axle body surface.
(d) Adjacent interference-fitted components must be separated and a stress-relief groove used in the seating between the two components.
(e) The hubs of interference- fitted components should be allowed to overhang the vertical face of the axle seating.
(f) The correct design must first be produced and then if geometrical form alone cannot fully compensate for the effects of stress concentrations, use must be made of one or other of the various types of surface treatments which are available.

Machining

Wheels

The advantages of good design and correct material properties of components are degraded if the machining is not of consistently high quality. Years ago the quality of surface finish and the degree of dimensional accuracy obtained, depended to a large extent on the standard of skill of the individual operator. Now, with the combination of mechanical and electronic principles it has become possible to leave to the machines-with a certainty of consistent quality-the execution of complicated operations.

Press fitting and shrink fitting

Press fitting wheels has tended to surround itself with an aura of mystique. Traditionally it has been controlled by ‘wheel shop practice’-‘know-how’.

It must be admitted the pressman’s skill is often challenged to produce some consistency in final results even with components identical in all respects. There are so many variables governing quantity and quality of lubricant, interferences, speed of pressing and surface finish of mating surfaces that the final press-on pressure is by no means predictable. It is now the practice to specify precisely the interference, the surface finish , the lubricant and maxima for out-of-roundness and out-of- straight in the wheel bore and axle seat.

Despite all precautions having been taken, scoring of wheelsets does occur during pressing for reasons which are not fully known and one can never be certain whether scoring or more serious damage has been caused during the final stages of pressing; consequently, it is possible for wheels to be finally positioned under a false pressure record. If such wheels have never to be removed from the axle, the matter is not serious but in cases where de-mounting is necessary the results are disastrous.

Shrink fitting completely eliminates the possibility of scoring the axle seat on mounting and is to be recommended for assemblies where there is a strong possibility that sometime in its service life the road wheel will be removed from the axle for the replacement of gears or inner bearings etc. the longitudinal stress concentration in the axle seat which results from press fitting is also eliminated by shrink fitting.

The fact that no pressure record is available as evidence of a successful fitting is of no great consequence since the contraction grip can be confirmed ultimately by a predetermined proving thrust if required. In any case, pressure records from press fitting will tend to become less significant as more precise parameters are employed and recorded for the mating parts.

Shrink fitting with a reduced interference achieves an equally tight grip as press fitting with a normal interference so reducing the stresses in the wheel hub.

The heating medium usually is oil or hot air and the wheel is through-heated. Undoubtedly dry shrinking when wheels are heated in a hot air furnace, gives a superior fit to that of oil heated wheels.

Both shrink and press fitting are available on the works depending on what the specification asks for. Shrink fitting can be achieved either by the use of an oil bath or in a recently installed hot air furnace.

However, induction heating is sometimes used in our workshops as an alternative to oil heating for the assembly of components requiring a light interference fit. Roller bearings are assembled onto axles in this way- the bearings being expanded on a Rotary Heater.

Certain precautionary measures must be taken, however, in regard to the heating equipment:

(a) The temperature must be controlled automatically by the pre-setting of a limiting device to control the current supply. Excessive overheating, which is a possibility with manual control, can soften the case hardened layer of the bearing.

(b) A control must be installed in the system to give a degraded power cut-off to reduce any retained magnetism in the bearing to acceptable limits.

(c) Assembly by freezing is carried out to fit the outer cups of roller bearing assemblies into their suspension tube housings. Crank pins, requiring precise circumferential location, also have been successfully positioned by this method.

Liquid nitrogen is the freezing medium, being non-toxic, inert and relatively easy to handle.

Oil injection method for separating joints

The oil injection method of separating pressure joints is a feature which has been in use for many years, and is based on the fact that friction between the contact surfaces in a pressure joint is practically eliminated by introducing oil under high pressure between the surfaces. The principle of the method can be seen from Fig.(2.33).

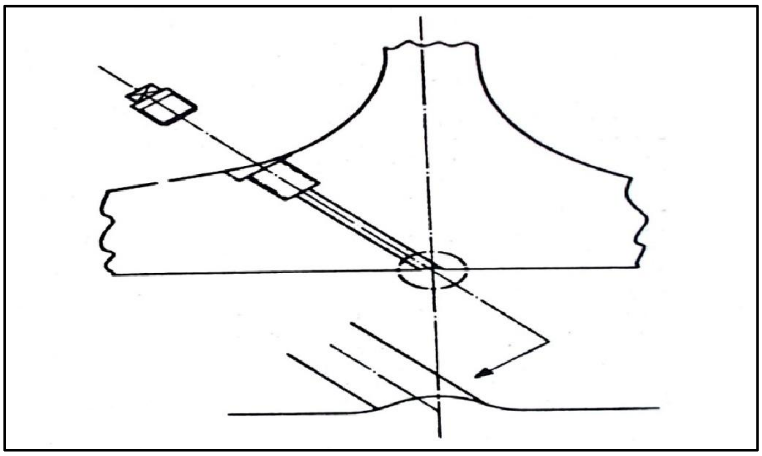


Fig.(2.33) Detail of oil unjection feature.

Ideally, oil should be introduced into that part of the contact area where the thickness of the wheel hub is greatest. This sometimes necessitates two oil ducts in the wheel hub to allow the second duct to take effect when the first becomes inoperative.

Very slight pressure from a wheel press is all that is required to finally remove the wheel and serious axle damage is eliminated.

It is economically justifiable to include the oil injection facility in all wheelsets whether assembled by shrink-fitting or press-fitting. It is of great value in the production of complicated wheelset assemblies where components have to be positioned to precise limits and where the full use of a wheel press would be inadvisable and possibly dangerous.

Wheelset Development

In more recent times, as well as manufacturing conventional wheelsets, emphasis has been placed on the development of lightweight wheelsets.

The advantages offered by lightweight wheelsets are as follows:

(a) Energy saving by a reduction of the dead weight of the vehicle.
(b) A saving of rotational energy built up in a wheelset during acceleration which has then to be dissipated during braking.
(c) A possible improvement in track life.

The concept involves the use of finite element analysis of a new design of wheel thinned at the rim and web since the braking mechanism is remote from the tread.

It also makes use of hollow formed axles.

Various designs have been tested and service trials are to be performed in the near future.

Another development is a trailor road/rail wheel which replaces the existing road wheels of a trailor by a rail wheel using existing stud fittings on the trailor axle.

Conclusion

It is hoped that this study gives the reader a clearer insight on the design and manufacture of railway wheelsets for modern operating conditions.

One can only speculate about the degree of change which the next twenty to thirty years will bring but there is no denying that changes will occur and we shall have to be ready to provide refined manufacturing and design technology to meet the new demands.

2.3.2 Rolling bearing in traction motors and suspension units

Historical .

Rolling bearings were first introduced into traction motors in the early 1900's but it was not until the late 1920's that they became universally accepted. This perhaps is not realy surprising when it is recalled how little knowledge was then available of the capabilities of rolling bearing. Axle hung traction motors were originally mounted on wool packed oil lubricated white metal sleeve bearings but with increased speeds, problems developed due to hot running, sealing and maintenance.

Parallel with the development of traction motors, attention was focused upon the suspension unit system. Plain bearings were used in early designs of suspension units but it was always necessary to match gears and pinions with this arrangement. Slowly, rolling bearings have also been introduced into this suspension unit application.

A very neat arrangement is now possible Fig. (2.34)

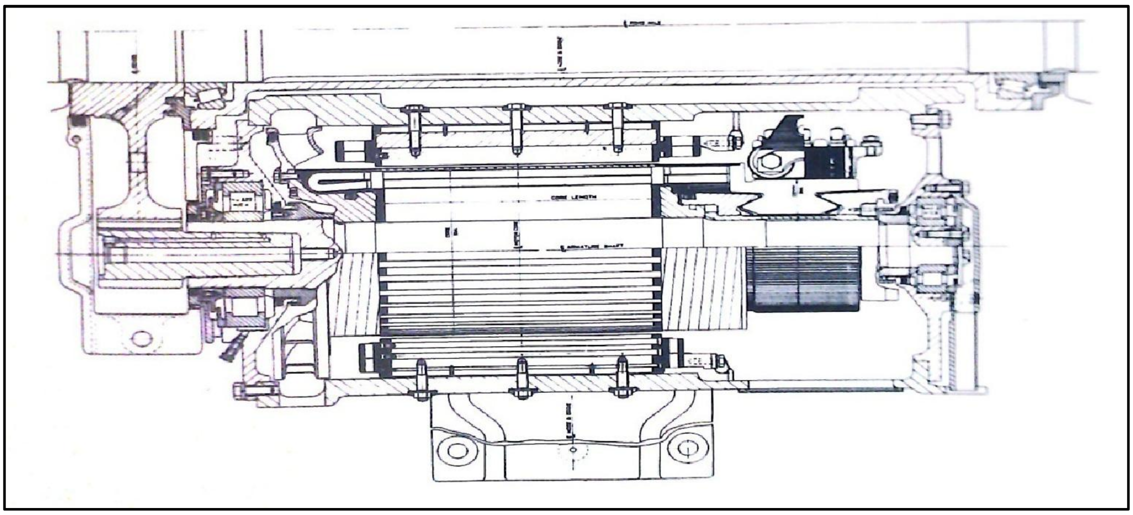


Fig.(2.34) Typical traction motor suspension unit arrangement.

As with most projects, therefore, it was only a gradual build up of experience which proved the suitability of rolling bearings for the arduous conditions prevailing in a railway environment and introduction of these into traction motors and suspension units have made it possible to:[6]

1. Improve the gear meshing due to reduced bearing wear.
2. Improve the gear lubrication and sealing.
3. Have a free interchange of gears and pinions.
4. Increase the running periods with grease lubrication.
5. Remove the motors from the axle without disturbing the suspension bearings.

There are a number of methods of power transmission for electric and diesel-electric locomotive throughout the world and they are basically covered under the following designs Fig. (2.35).

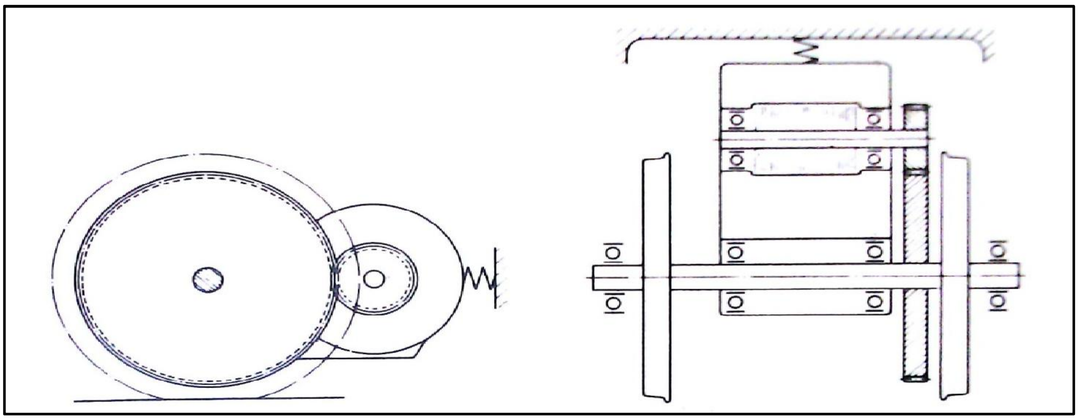


Fig.(2.35. a) Axle hung-suspension bearings.

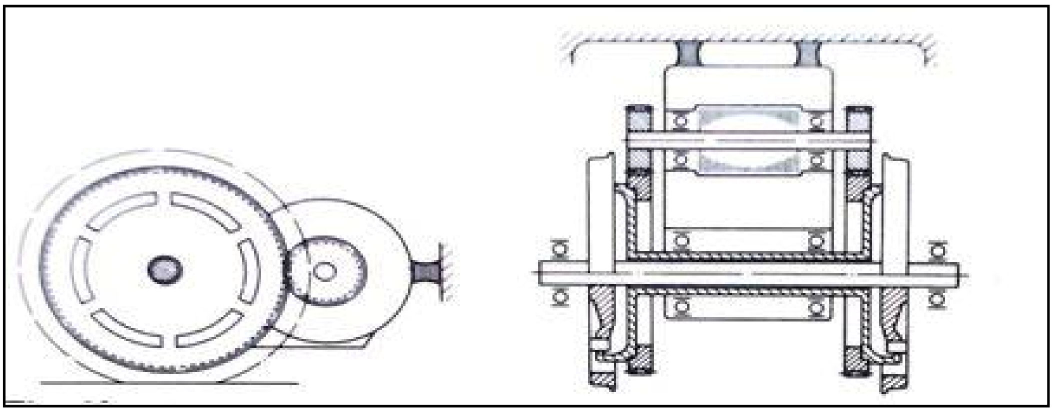


Fig.(2.35.b) Quill drive.

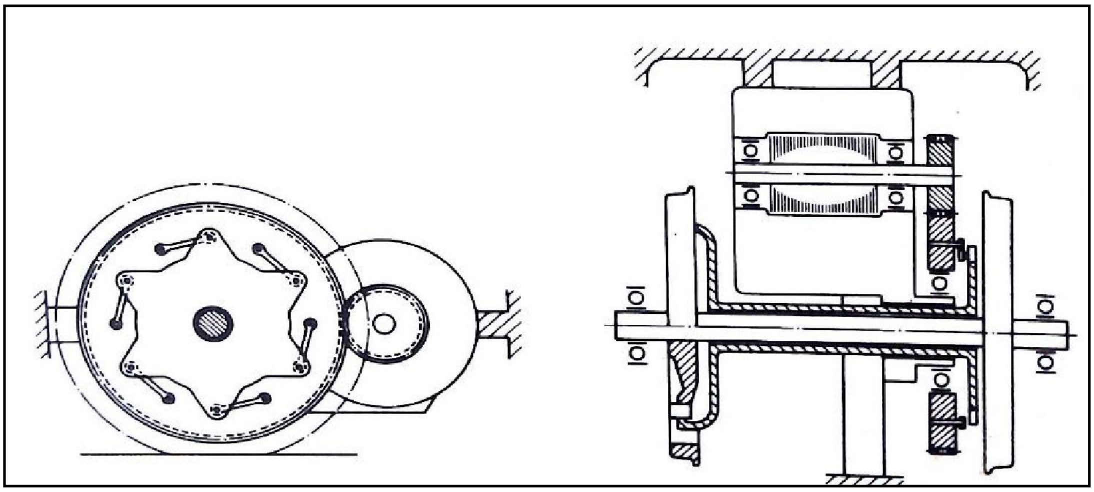


Fig.(2.35.c) Quill drive.

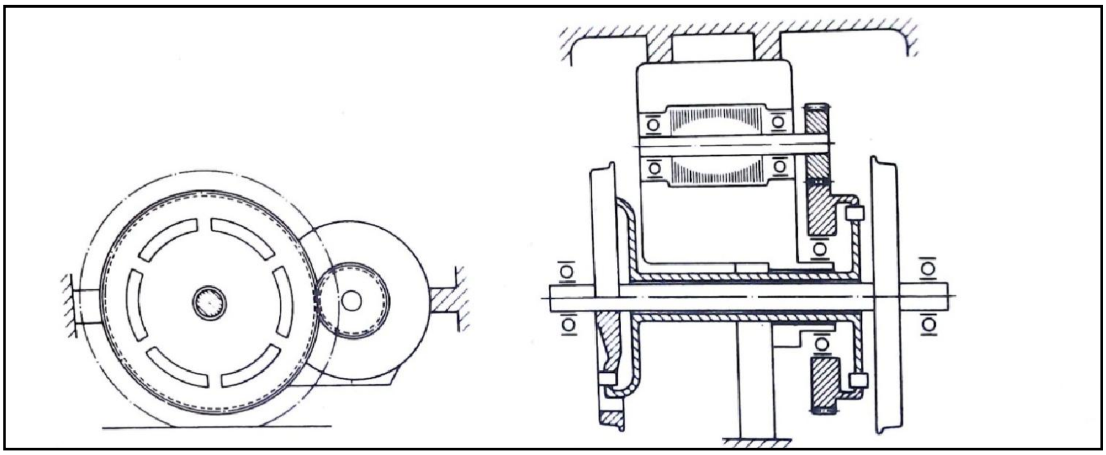


Fig.(2.35.d) Quill drive.

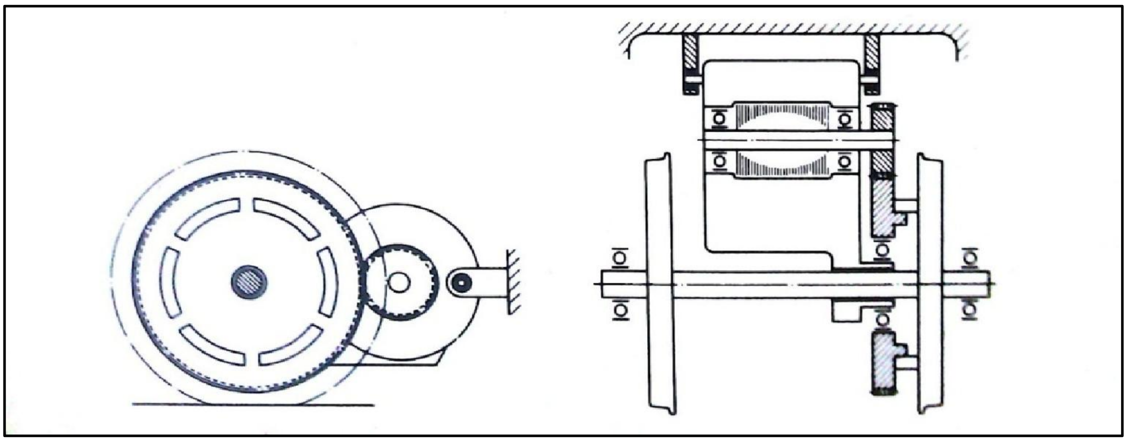


Fig.(2.35.e) Gearbox drive.

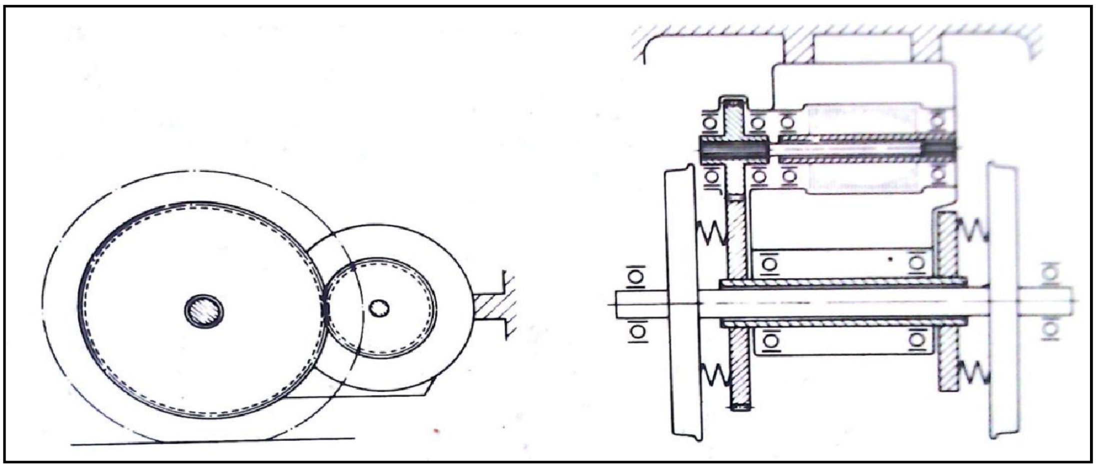


Fig.(2.35.f) Torsion bar drive with quill.

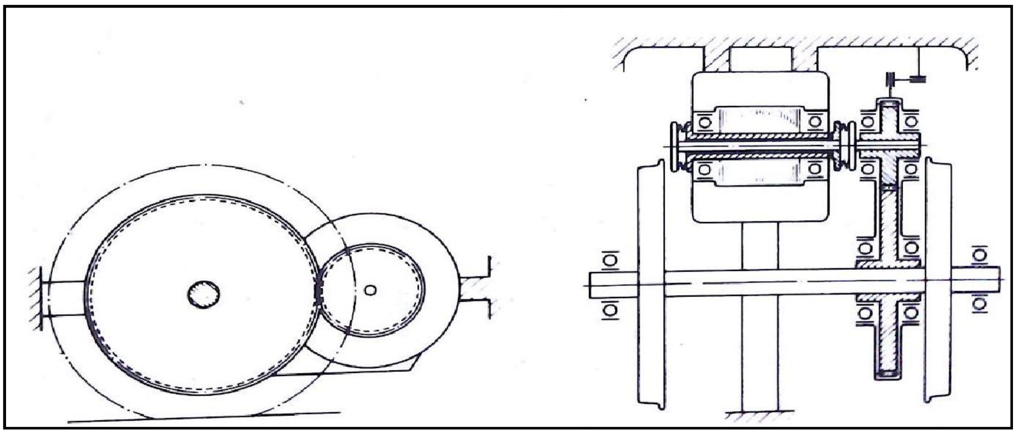


Fig.(2.35.g) Cardan shaft drive.

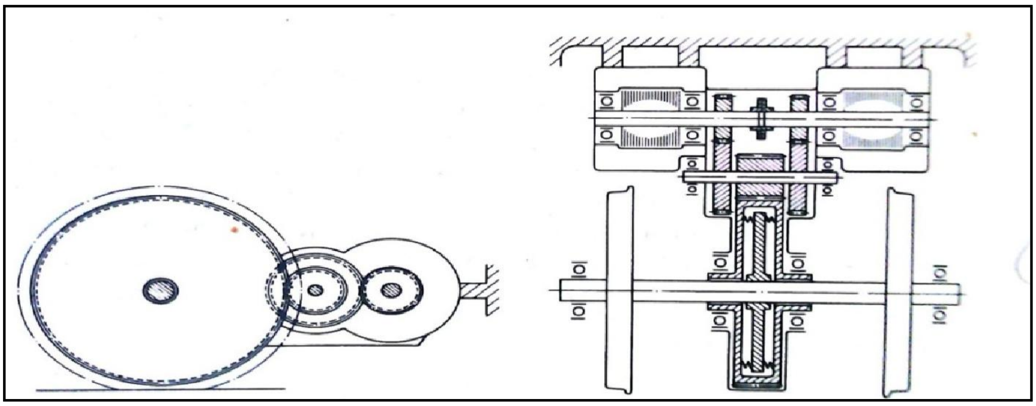


Fig.(2.35.h) Geared axle hung with suspension bearing.

Traction Motors

Very early in the development of traction motors, it became apparent that in order to maintain and indeed obtain increased efficiency, maintaining consistent and reduced air gaps was necessity. In the first place, the size of the gap depended purely upon machining capability of the supplier, whilst maintaining this gap depended upon the ability of the bearing to withstand wear. SKF developments in bearing technology together with the experience gained, showed that the air gap could be maintained by using rolling bearings. It was also proved that loading could be increased considerably and that the bearings were capable of carrying heavy loads for long periods, as well as withstanding shock and overload for short periods.

In the United Kingdom, the standard design has usually consisted of a motor mounted on two rolling bearings, with a pinion at one end direct driving onto a gear on the axle. The motor is supported on the suspension units. [6]

A variety of bearings has been used in traction motors. These have been spherical roller bearings with and without sleeves, ball bearings, combinations of ball bearings and cylindrical roller bearings (the ball bearing being used for location purposes only) and cylindrical roller bearings. Experience has proved that cylindrical roller bearings have been capable of carrying the heavy traction loads and withstanding forces for very long periods. At the pinion end it is normal to use a standard design where the retaining lips are on the outer ring. At the commutator end the cylindrical roller bearing has retaining lips on the outer ring whilst there is one retaining lip on the inner ring and the bearing is fully located by means of an angle ring. A part from the capacity of the bearing, cylindrical are used in traction motors because the mounting and dismounting is relatively uncomplicated when maintenance is required on the motors. The shields may easily be removed complete with the outer assemblies leaving the inner rings visible on the rotor shaft. In addition, much technical development work has been carried out on the capability of cylindrical roller bearings to carry thrust loads in applications where a 7 1/2degree or 15 degree helical pinion is used.

During the early stages of design, operating speeds were moderate but when motor improvements were made the operating requirements became more arduous. As the speed and the loads increased and the output of the motor became higher this resulted in the reduction in space available in which the rolling bearings were to be mounted. Considerable improvements took place in the design of the cylindrical roller bearings and from exhaustive tests, it was established that operating life could be increased by modifying the rolling tracks e.g. crowning of tracks or rollers. This eliminated the possibility of stress concentration at the edge of the roller path. Further improvements were made by reducing ring sections and increasing the roller length and diameter, thereby increasing the carrying capacity without affecting the fatigue life.

Motor design technology has developed with both an increase in speed and the reduction in size relative to output. Bearings have responded to meet this development and now the situation is such that maintenance is reduced to minimum, and in some cases lubrication intervals of 500 000 km are requested and achieved. Consequently, it can confidently be said that technological development in rolling bearings have considerably contributed to the efficiency and reliability now being obtained by traction motors.[6]

Suspension Units

In the early days of design it was necessary to mount the suspension tube into the space previously taken up by the plain bearings, which were an integral part of the motor. The plain bearing halves were in fact joined to the motor body by means of split caps. The earliest tubes were, therefore, secured by means of these caps but like many developments this did cause some problems, due to the stressing when the caps were clamped to the tube body.

Wear does occur in plain bearings and result in gears moving from their true positions and if problems developed with a motor or axle, both gear and pinion were removed and retained as a pair. This wear also made it necessary to match motor pinion and axle gears with the result that reduction in motor maintenance costs was lost. In order to compensate for wear, the practice was developed of filling the gear cases with a bituminous compound which helped cushions the shock of pinion and gear mating on their pitch circle and also assisted in reduction of the noise from the gears. The practice of using this bituminous compound is still current in certain applications but because of its high viscosity, extreme care must be taken to ensure sealing such that it is prevented from entering the bearing housing. In particularly cold conditions this can prevent turning of the rolling elements resulting in skidding and therefore material distress.

Nevertheless, the problems associated with paining gears were eliminated by the use of rolling bearings in traction motors, but other problems still remained. Gears fitted to traction motors are relatively small and accurately ground in order to ensure even distribution loads. Any variation in gear sequence will upset the heavy tooth stresses. In order to counteract this, wear must not occur between the motor bearings or the suspension unit bearings.

The reduction in size of motor pinions was made possible by using the SKF oil injection method. By this means, pinions may be mounted direct onto the motor shaft or plugged into the motor shaft thus eliminating the use of locking keys Fig.(2.36).

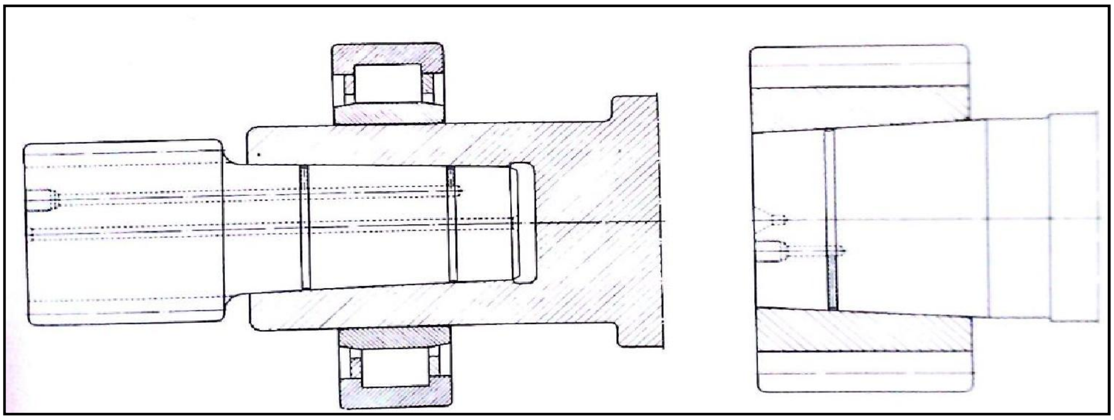


Fig.(2.36) Pinion drives for traction motors showing oil injection grooves.

Rolling bearings have been introduced into the suspension frames using a variety of design: ball bearings, cylindrical roller bearings, taper roller bearings and spherical roller bearings Fig.(2.37). All of them have given satisfactory results and must be related to the type of application or environment in which they are required to operate. With a gradual evolution in the designs, tubes have been designed so that they can either clamp direct onto the motor frame or blot directly to the motor frame feet. The designs, therefore, have either been produced as solid components to spilt halves or of latter years, the design now known as the "U" type Fig.(2.38). The latest development in the suspension unit system is the use of independent housing which clamp onto the motor frame and eliminate the center section of the tube Fig.(2.39). In addition, various materials have been used such as cast steel, forged steel and light aluminum forgings.

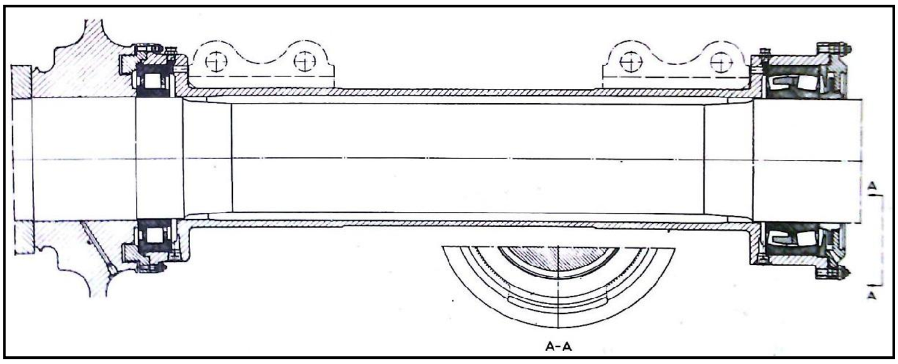


Fig.(2.37) Suspension unit bearing arrangement.

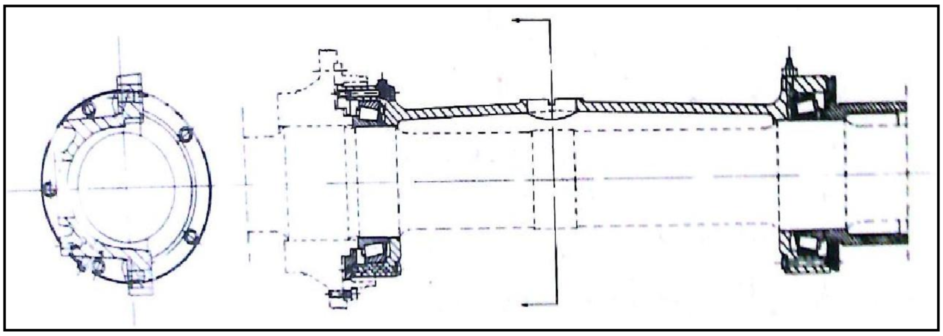


Fig.(2..38) Suspension unit bearing arrangement 'U' type.

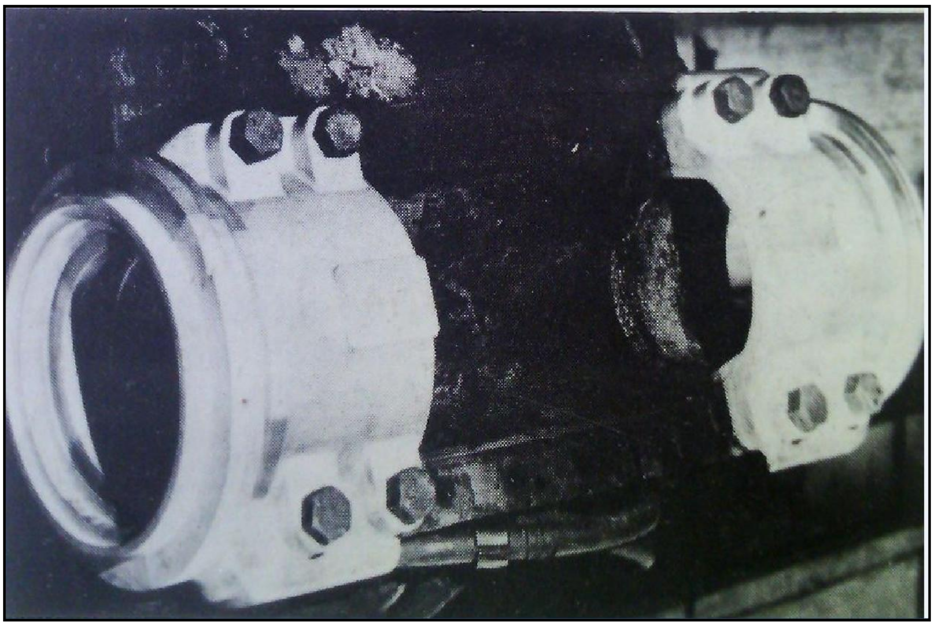


Fig.(2.39) Suspension unit with independent housing clamped onto the motor frame.

CHAPTER THREE

RAIL CARS & FREIGHT WAGONS BOGIE DESIGNS AND PERFORMANCE

3.1 Introduction

This chapter is for the design & development of rail cars (passengers) and freight wagons bogies.

The bogies are very important in safe railway operations &must give good ride comfort by absorbing vibrations resulting from track inegularities.

The freight wagons bogie design evoluated from the conventional plate & cast steel bogie design Fig.(3.1 & 3.2) then development started by incorporating damping systems &development of suspension systems to snub the vibrations and protect the lading, vehicle and the track.

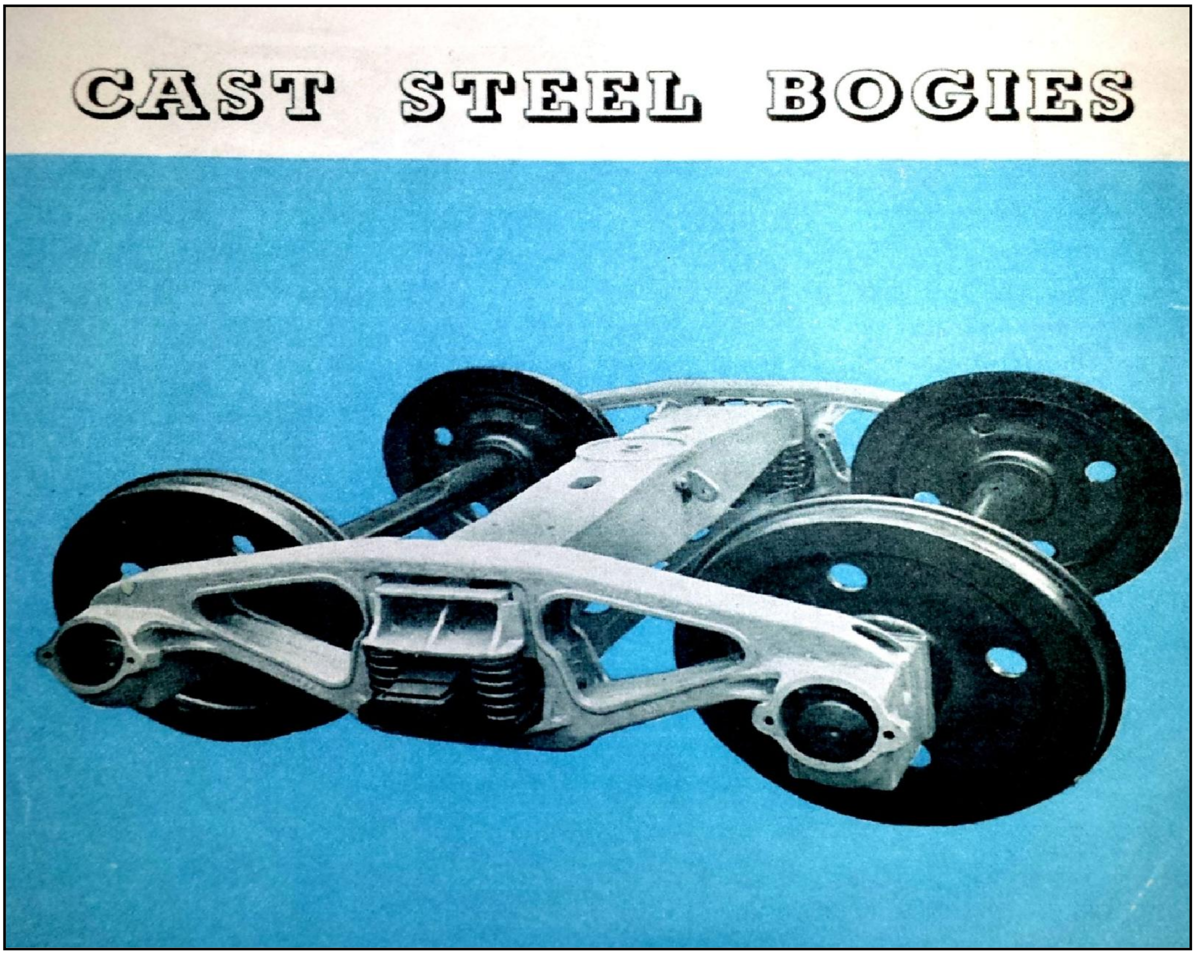


Fig.(3.1) cast steel bogie.

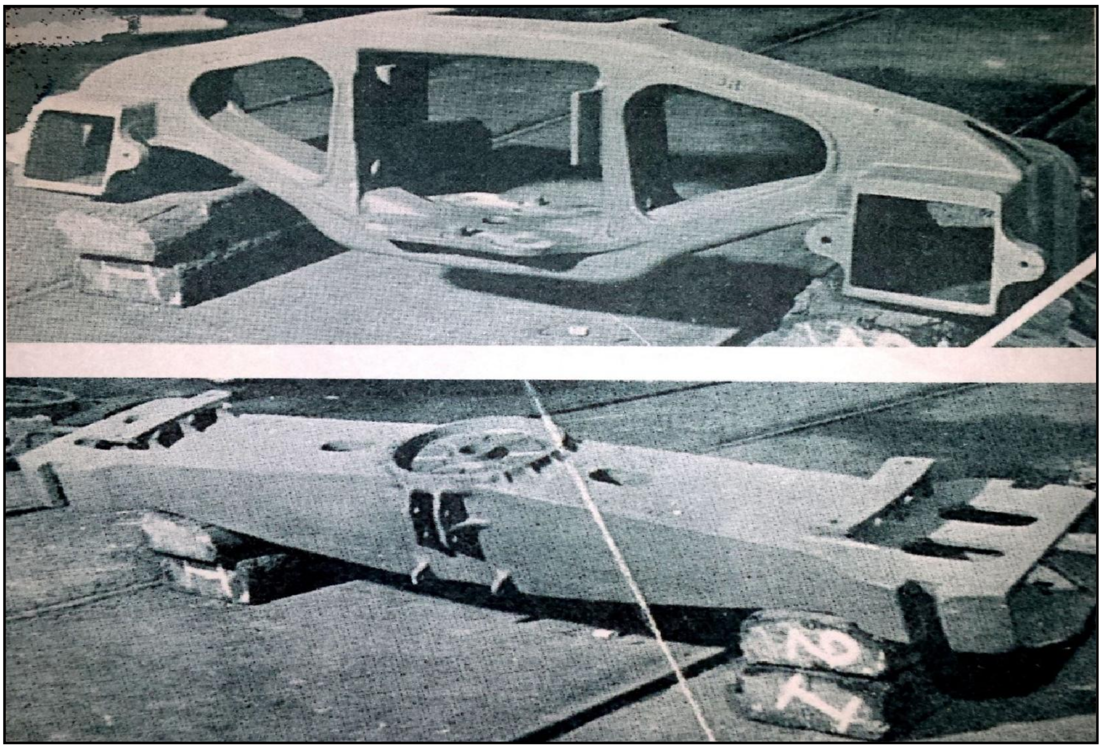


Fig.(3.2) cast steel bogie.

The track constant frictional force (in the ride control bogie design) and proportional to the wagons gross weight frictional force (in the stabilized barber bogie design) mechanisms have been successfully added to the bogie design one after the other.(See figs.(3.3& 3.4) for the ride control & the stabilized bogies).

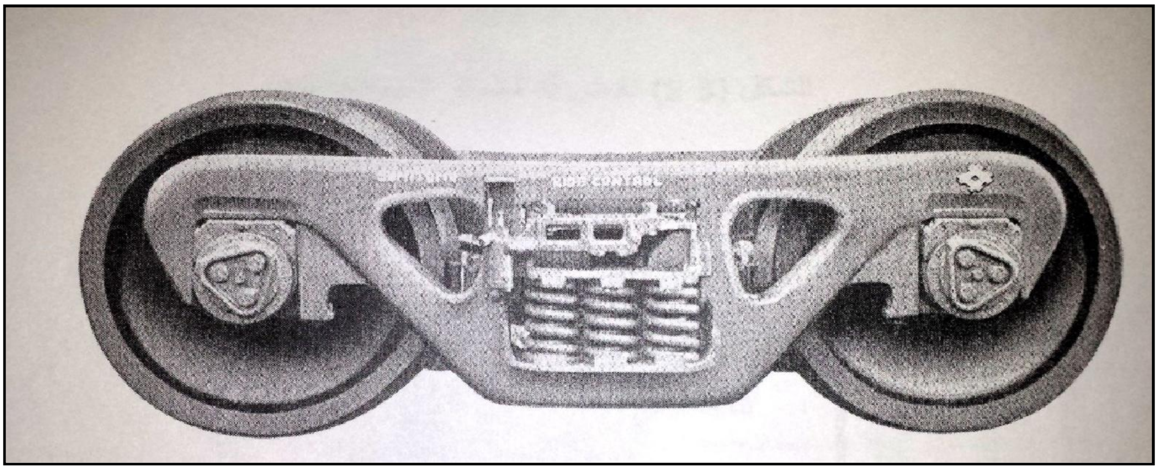


Fig (3.3) The ride control bogie.

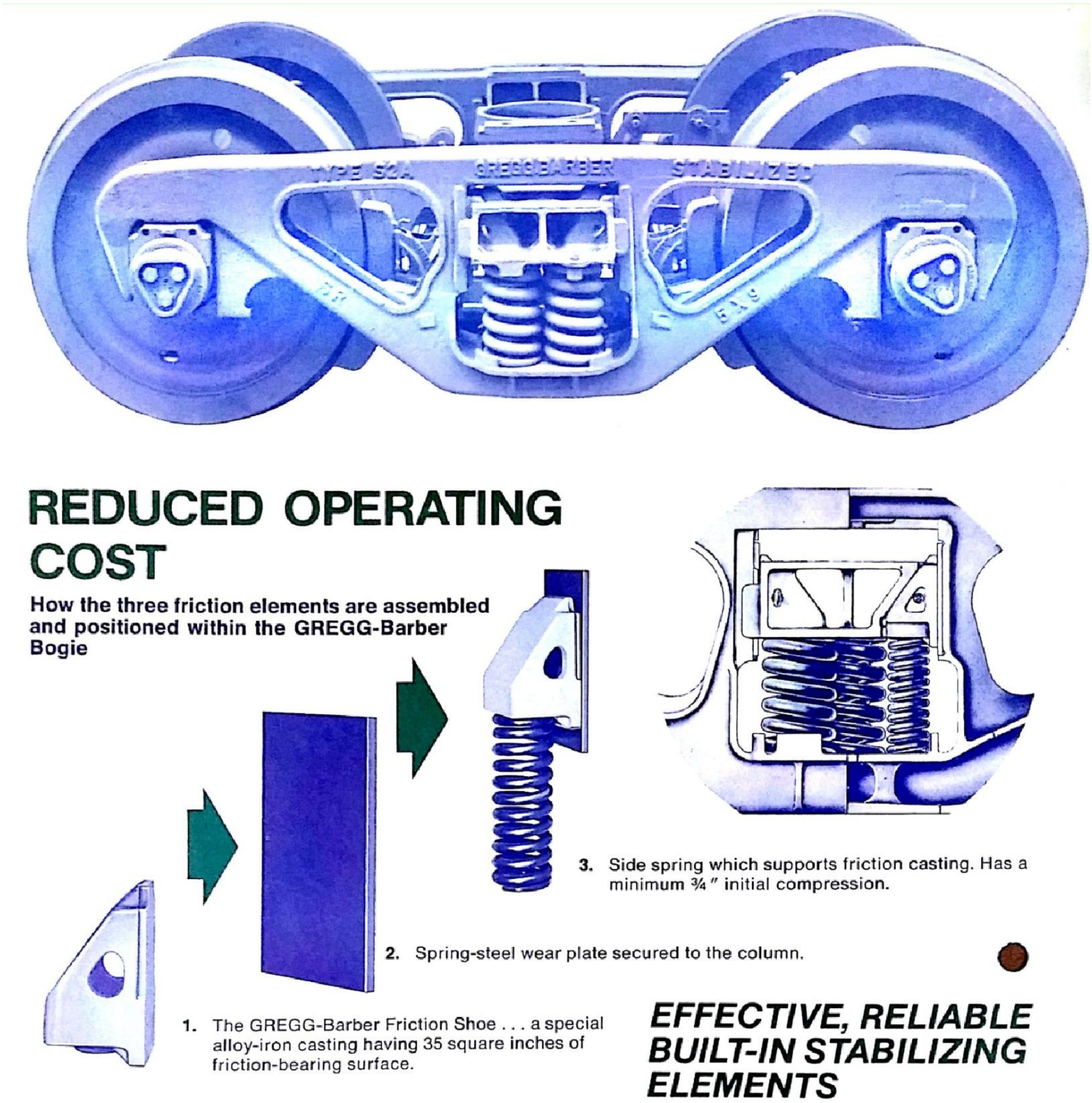


Fig.(3.4) the stabilized bogies.

The stabilized self-steering bogie design is the last development in the freight weagons bogie design up to now Fig.(3.5).

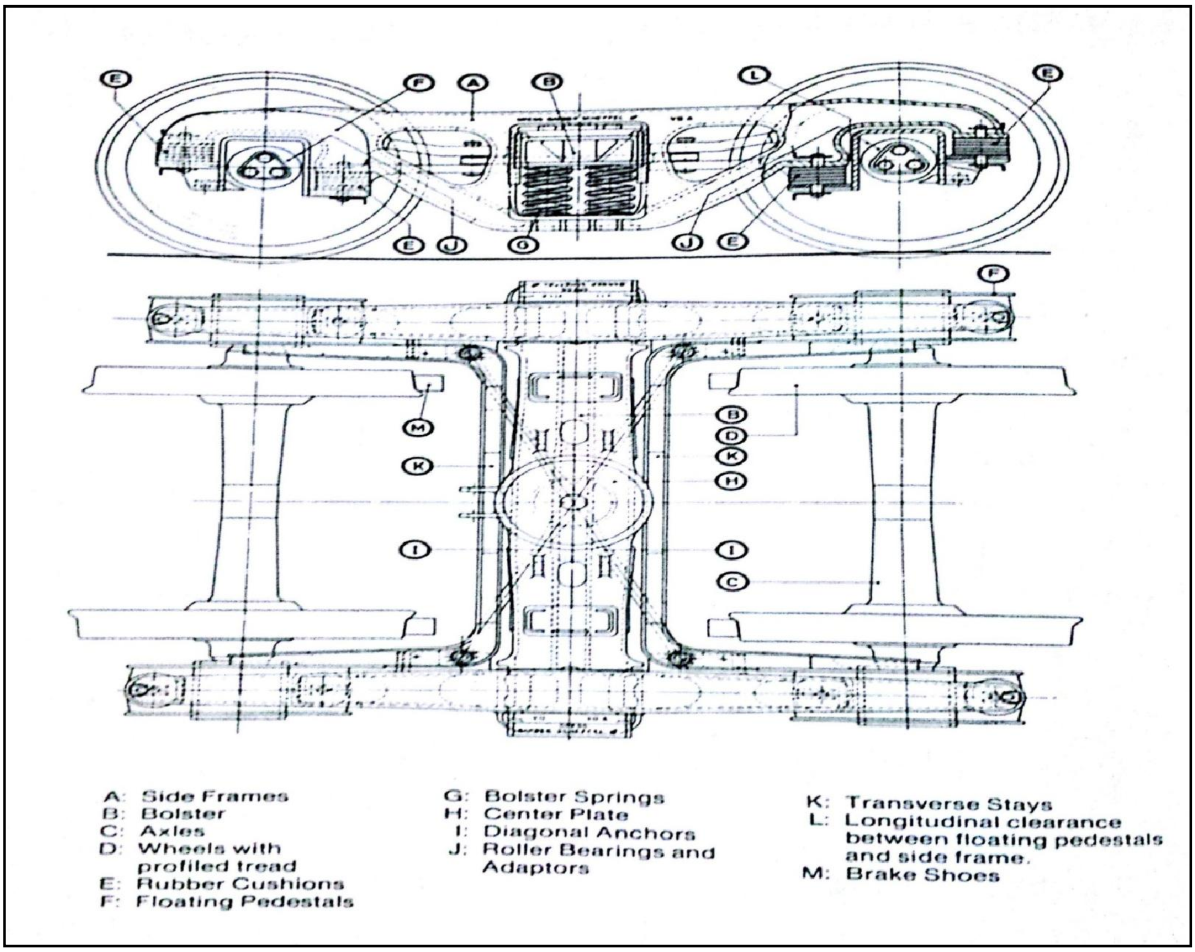


Fig (3.5)The stabilized self-steering bogie.

To show the merits of the proportional friction force mechanism (stabilized bogie), over the constant friction force mechanism (ride control) a comparison of four freight wagon bogies with damped mechanisms shall be displayed in this chapter to show that the variable friction force system is a step foreward in the bogie design.

Also some comments extracted from the American Association of Railroads (AAR) "Shock& Vibration Hand Book-vol.3-1961" will show the superiority of the Barber stabilized design over the ride control design.

The chapter will be closed by a complete study on the design of bogies reached to by use of track dynamics programmes and other aids. Subjects studied are:

Ride quality, latral stability, curving theory, track forces, suspension components, and typesof bogies.

3.2 Rail Cars & Freight Wagons Bogies[7]

a) The railcar (multiple unit) bogie

Railcar bogies usually go unnoticed by rail passenger, but despite their obscurity, they are very important in safe railway operations and perform the following functions.

1) Support railcar body firmly.
2) Run stably on both straight and curved track.
3) Ensure good ride comfort by absorbing vibration generated by track irregularities and minimizing impact of centrifugal force when train runs on curves at high speed.
4) Minimize generation of track irregularities and rail abrasion.

The parts and description of a motored bogie which could be designed for an electric or diesel locomotive or electric multiple units (EMU) are shown in Fig.(3.6).

The bogies, or truck as is called in the US, come in many shapes and size but it is in its most developed from as the motor bogie of an electric or diesel locomotive or an EMU. Here it has to carry the motors, track and suspension systems all within a tight envelope. It is subjected to severe stresses and shocks and many have to run over 300 km/h in the high speed application. The following paragraphs describe the parts shown on the Fig.(3.6), which is of modern UK design.

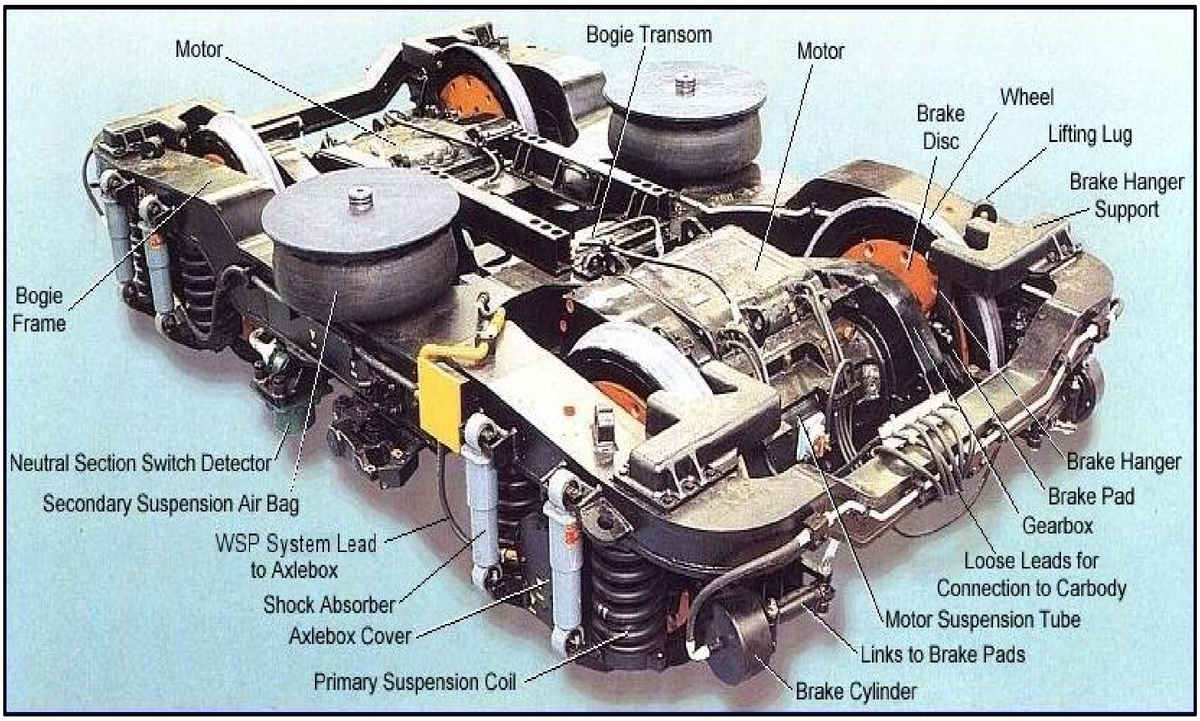


Fig.(3.6) bogie of an electric or diesel locomotive.

Bogie Frame

Can be steel plate or cast steel. In this case, it is a modem design of welded steel box format where the structure is formed into hollow section of the required .shape.

Bogie Transom

Transverse structural member of bogie frame (usually two off) which also support the car body guidance parts and the traction motors.

Brake Cylinder

An air brake cylinder in provided for each wheel. A cylinder can operate tread or disc brakes. Some designs incorporate parking requirements. Each wheel is provided with a brake disc on each side and brake pad actuated by brake cylinder. A pair of pads is hung from the bogie frame and activated by links tothe piston in the brake cylinder. When air is admitted into the brake cylinder, the internal piston moves these links and causes the brake pads to press against the discs. A brake hanger support bracket carries the brake hangers, from which the pads are hung.

Primary Suspension

A steel coil spring, two of which fitted to each axle box in this design. They carry the weight of the bogie frame and anything attached to it.

Motor Suspension Tube

Many motor are suspension between the transverse member of the bogie frame called the transom and the axle. This, motor is called “nose suspension” because it is hung between the suspension tube and the single mounting on the bogie transom called the nose.

Gearbox

This contains the pinion and gearwheel which connects from the armature to the axle.

Lifting Lug

Allows the bogie to be lifted by are crane without the need to tie chains or ropes around the frame.

Motor

Normally, each axle has its own motor. It drives the axle through the gearbox. Some designs, particularly on tramcars, use a motor to drive two axles.

Neutral Section Switch Detector

In the UK, the overhaul line is divided into sections .with short neutral sections separated them. It is necessary to switch, off the current on the train while the neutral section in crossed. A Magnetic Device mounted on the track marks the start and finish of the neutral section. The device is detected by box mounted on the leading bogie of the train to inform the equipment when to switch off and on.

Secondary Suspension Air Bag

Rubber air suspension bag are provided as the secondary suspension system for most modem trains. The air is supplied from the train's compressed air system.

Wheel Slide Protection System Lead Axle Box

Where a wheel slide protection (WSP) is fitted, axle box are fitted speed sensors. These are connected by means of cable attached to the WSP box cover on the axle end.

Loose Leads for Connection to Car Body

The motor circuits are connected to the traction equipment in the car or locomotive by flexible leads shown here.

Shock Absorber

To reduce the effects of vibration occurring as result of the wheel/rail interface.

Axle box cover

Simple protection for the return current brush, if fitted, and axle bearing lubrication.

b) The freight wagon bogie

The most important sub-assembly in the freight wagon is its bogie. A railway could be evaluated by the advancement of the technology used in design and manufacture of the bogies of its freight rolling stock fleet.

From SRC 1996/1997 statistics, its freight rolling stock fleet consists of 6190 wagons and tank wagon 5225 wagons of these (84%) have conventional cast steel bogies. 986 wagons and tank (16%) have enhanced design bogie (ride control & stabilized bogies). Therefore SRC in over 80% of its rolling stock fleet is populated by the 19th century technology. The 84% wagons could be upgraded by changing some components of their bogies.

The research and development in freight wagons bogie design, round the world limited their design in four types up to now (2015).

1) Conventional plate & cast steel.
2) Ride control bogies.
3) Barber stabilized bogies.
4) Self-steering bogies.

The first three types are in use in SRC with number 2 and 3 comprising 16% of SRC wagon fleet.

The conventional cast steel bogie is fitted in 84% of SRC wagons and needs to be upgraded.

SRC rolling stock specification is vague. It should be modified to specify stabilized and self-steering bogies as their minimum requirements in the future.

In the US, cast steel was the most popular material for bogies and simple basic design evolved Fig.(3.6).

In its simplest form, as used under the standard American freight car, spring suspension was only provided for the bolster. The bogie consisted of three main parts the bolster and the two side frames. The basic arrangement provided for a set of steel coil springs provided inside an opening in each side frame of the bogie.

The bogie bolster (truck bolster in US) was mounted on top of these springs and held in place by guides cast into or bolted onto the bolster, the axle boxes were not sprung and merely slotted into the frame, which rested directly on them. The ride wasn't soft but it was adequate. Some later versions of this truck have axle box springs simple coil springs inserted between the top of the axle box and the truck frame.

A comparison of four bogies, three of ride control design & one of stabilized design, shall be presented here under to show the superiority of the stabilized design over the ride control.

3.3 Comparison of four types of Bogies[8]

1. General

There are four typical freight car bogies authorized with AAR (Association of American Railroads) code for high speed use, i.e., NATIONAL C-1, BARBER S-2, ASF A-3 and BUCKEYE C-R.

All these bogies use friction damping force produced by snubber springs in order to control both vertical and lateral vibrations.

Fig.(3.7) and table (3.1) show respectively the snubbing mechanisms and the construction of these bogies.

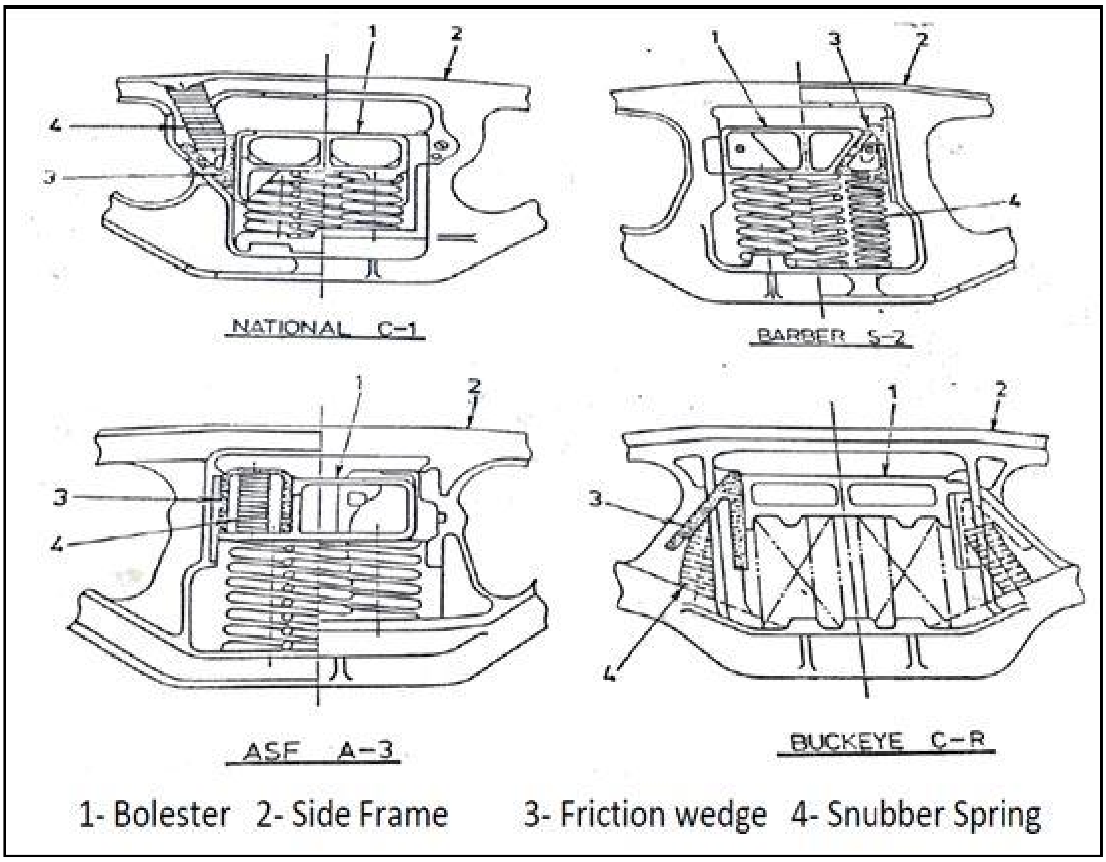


Fig.(3.7) The four bogies.

In Fig.(3.7) , each type include a bolster 1, side frame 2, friction wedge 3, and snubber spring 4. In NATIONAL C-1, ASF A-3, and BUCKYE C-R the force applied by the snubber spring is independent of the relative motion between the bolster and side frame that is weight of the car, thus, the friction damping force is constant. In type BARBER S-2, the snubber springs carry a portion of the weight of the car, thereby increasing the friction damping as the weight increases.

Table (3.1) Construction of the four bogies:

Item / Type	National C-1	Barber S-2	ASF A-3	Buckeye C-R
Damping friction force (against load)	Constant	Variable	Constant	Constant
Friction wedge position	Side frame	Bolster	Bolster	Side frame
Snubbing spring Position	Side frame	Side frame	Bolster	Side frame

It is considered to be desirable that the friction damping force produced by the snubber spring is variable according to the change of the load conditions such as empty, half-loaded or full-loaded state.

Among the said four types of bogies, only BARBER S-2 BOGIE has an excellent mechanism to produce such variable snubbing force (friction damping force) according to the load change. Meantime, as snubbing force of the remaining three types of bogies is independent of the load change, it can be said that those bogies are liable to be overdamping or underdamping.

2. Performance

According to the road test carried out by AAR about twenty years ago (1949-1950), of which results were published in title of "AAR-Cooperative Freight

Trucks and Snubber Research Program" each Lading Damage Index of four types of bogies are found a little different respectively. Although many improvements has been performed so far as three CONSTANT SNUBBING FORCE type bogies, the optimum value of L.D.I. has not been obtained. Meanwhile, BRBER S-2 BOGIE has constantly improved as shown by following best L.D.I. value.

Table (3.2) L.D.I values in different conditions

RAIL LOAD	**169,000 *Lbs.* (FULL-LOAD)**	**145,000 *Lbs.* (80% LOAD)**	**60,000 *Lbs.* (EMPTY)**	**AVERAGE**
L.D.I. RATIO (VERTICAL)	0.56	0.34	0.42	0.44
L.D.I. RATIO (LATERAL)	0.48	0.69	0.63	0.60

The following description extracted from "Shock and Vibration Handbook – vol. 3 – 1961" shows the excellence of BARBER STABILIZER. "BARBER STABILIZER has a wedge loading spring mounted in such a way that supports some of the weight of the car; therefore, the force it exerts on the wedges is proportional to the weight of the car.

Tests indicate that the principle embodied in BARBER STABILIZER is desirable.

3. Construction

I. NATIONAL C-1 and BUCKEYE C-R

As shown in Fig.(3.8) and table (3.3), NATIONAL C-1 and BUCKEYE C-R type bogies have friction wedge and snubber spring installed in the pocket part of the side frame in respect of stress, and need to be reinforced. In addition, these bogies are disadvantageous in respect of complex construction, rather heavy weight and higher production cost.

II. BARBER S-2 and ASF A-3

BARBER S-2 and ASF A-3 have friction wedge installed in the end of bolster, where bending moment is nearly equal zero. ASF A-3 has snubber spring at the same place as friction wedge, which makes the bolster structure of this bogie complicated. From the above, it can be said confidently that BARBER STABILIZED BOGIE is the most excellent bogie in respect of structure, strength and weight.

4. Maintenance and inspection

AAR Interchange Rule mentions that maintenance cost for BARBER STABILIZED BOGIE is not exceeding half of that for the other three types of bogies, which shows that the simplicity of construction of BARBER STABILIZED BOGIE contributes to low maintenance cost.

For BARBER STABILIZED BOGIE, no special tool is required in assembling and/or disassembling of the bogie, but friction wedge (friction castings) are set and/or detached at the special place on the bolster with special cotter before usual operation of bogie. Snubber springs will be easily set at the same place as bolster springs. And it is also one of the characteristics of BARBER STABILIZED BOGIE to inspect the bogie easily from the outside. For reference we attach the copy from Code of Rule byAAR as table (3.3)

Table (3.3) from code of rules by association of American railroads

Item –code No.	Repairs Made	Charge (1)
	wedges, friction blocks, or shoes truck bolster or side frame for built in truck stabilizing or lateral motion devices including jacking of car, not including spring or wear plate for same, net, applied, each.	
	(1) Renewed separately:	
4515	(a) ASF Ride Control type, any size	36.30
4516	(b) Barber Stabilized type, any size	17.85
4517	(c) National C-I type, any size	34.60
4518	(d) Buckeye C-R type, any size	38.20
	(2) Renewed in connection with R. & R. or R. (2) of side frame R. & R. (3) of truck bolster, exchange of wheels or after one wedge is applied the same truck at same time (material only):	
4520	(a) ASF Ride Control type, any size	6.80
4521	(b) Barber Stabilized type, any size	4.95
4522	(c) National C-I type, any size	5.15
4523	(d) Buckeye C-R type, any size	8.20
	Spring for stabilizing wedge or friction block of built-in truck stabilizing or lateral motion devices, including jacking of car, net, applied, each.	
	(3) Renewed separately:	
4530	(a) ASF Ride Control type, any size	30.45
4531	(b) Barber Stabilized type, any size	14.90
4532	(c) National C-I type, any size	30.75
4533	(d) Buckeye C-R type, any size	33.20

NOTE

(1) Represents the amount one railroad may charge another for the described work done on the other's truck.

(2) R. & R. or R. means: Removed and replace same article or removed old article and apply new one.

(3) R. & R. means: Remove and replace same article.

5. Supply Record

BARBER STABILIZED BOGIE has so many excellent characteristics that it is extremely popular in the leading railways in U.S.A. Canada and many other countries. BARBER STABILIZED BOGIE often called the best bogie in the world occupies about 80% share of newly-built freight cars in U.S.A. and approximately 100% in Canada.

Supplies Record of S-2 BARBER STABILIZED BOGIE is attached for reference.

Table (3.4) Supply Record of S-2 BARBER STABILIZED BOGIE

DELIVERY	Q'TY/	CUSTMER	REMARK
1975	200	KOREA NATIONAL RAILWAY (KNR)	50 GONDOLA CAR
1976	10	KOREA NATIONAL RAILWAY (KNR)	3-AXLE BOGIE
1977	200	SAUDIARABIA (SGRRO)	50 FLAT CAR
1977	100	SAUDIARABIA (SGRRO)	50 TANK CAR
1977	100	SAUDIARABIA (SGRRO)	50 GONDOLA CAR
1977	100	SAUDIARABIA (SGRRO)	50 BOX CAR
1977	30	SAUDIARABIA (SGRRO)	SPARE PART
1977	600	KOREA NATIONAL RAILWAY (KNR)	50 GONDOLA CAR
1979	330	KOREA NATIONAL RAILWAY (KNR)	50 BOX CAR
1979	462	KOREANATIONALRAILWAY(KNR)	50 GONDOLA CAR
1979	60	KOREANATIONAL RAILWAY (KNR)	50 BALLAST HOPPER CAR
1979	56	HONAM OIL REF. COM. (KOREA)	50 TANK CAR
1980	76	SUDANRAILWAY (SRC)	"
1981	50	SUDANRAILWAY (SRC)	50 CONTAINER CAR
981	50	SUDANRAILWAY (SRC)	35 FLAT CAR
1981	100	SUDANRAILWAY (SRC)	35 GRAIN HOPPER CAR
1982	60	KOREA NATIONAL RAILWAY (KNR)	50 BULK CENTER CAR

3.3.1 S-2 Barber Stabilized truck

3.3.1.1 Introduction

The stabilized design, pioneered by Barber – the USA Company, is a step forward to introduce a damping system design with a damping force proportional to the state or dynamic load of a railway vehicle to have superiority over the constant Ride Control system design.

This Barber stabilizer bogie is produced and supplied by GREGG USA Company with minor changes and additions which made it carry the title “GREGG – BARBER STABILIZED BOGIE”.

We mention this, because the first fright wagons to be supplied to SRC with stabilized bogies were from GREGG Company of Belgian.

3.3.1.2. S-2 Barber stabilized truck

This description has been extracted from AARS “Shock and Vibration Handbook – vol.3 - 1961”.

Efforts have been made to make S-2 BARBER STABILIZED TRUCK as possible in construction Fig.(3.8) , taking into consideration the fact that the reduced time and cost of maintenance are the most indispensable requirements for fright cars which are predominantly larger in number than other types of railway rolling stock.

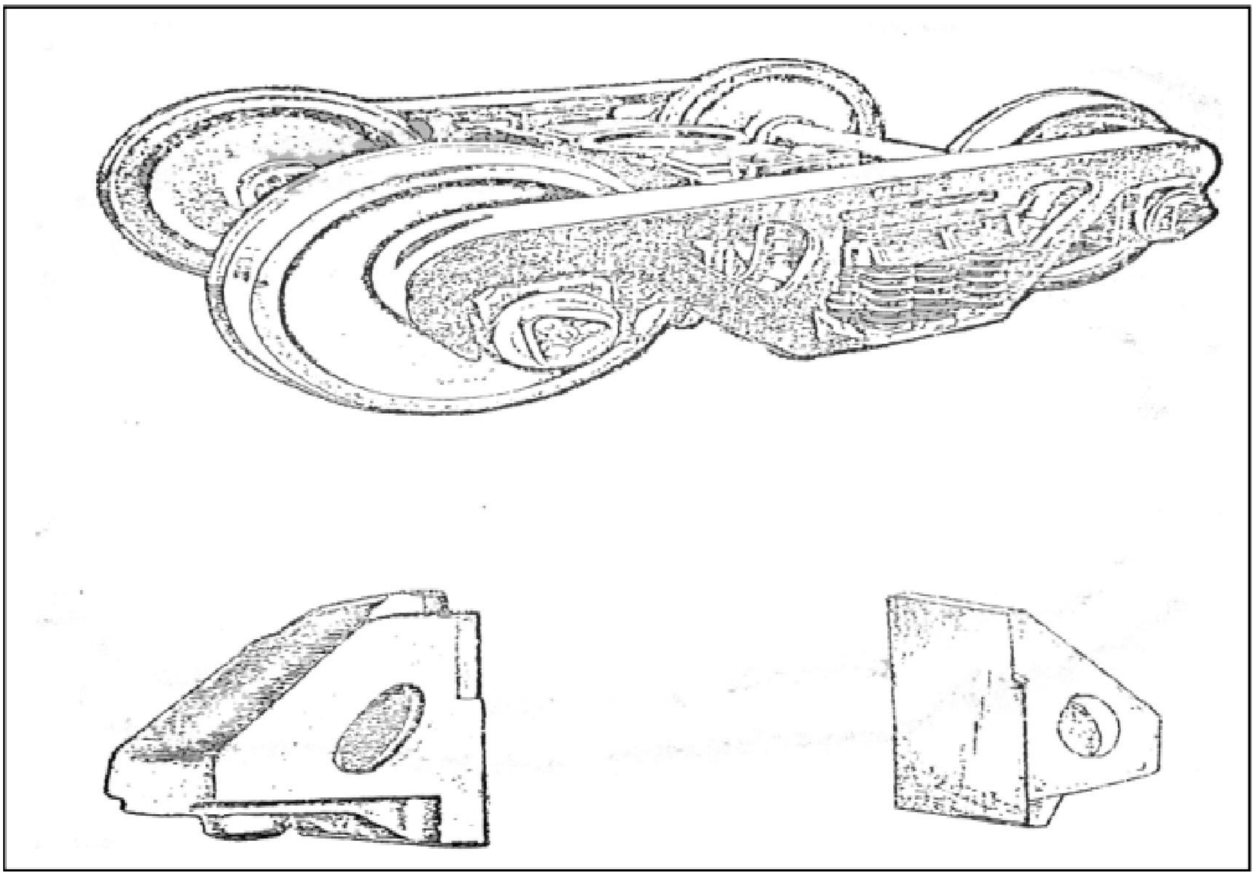


Fig.(3.8) S-2 barber stabilised truck.

For the bolster spring of the truck, a set of bolster spring having different combinations of spring arrangement for different magnitude of load and with a longer travel are prepared and specially the stabilizer device liable to become a complicated construction is designed extremely simple while employing a unique spring supporting system capable of exerting the most optimum running performance.

The side spring for the stabilizer of S-2 BAREBR STABILIZEDTRUCK is set in parallel to the bolster springs to let it support part of the load, so that it can obtain a friction matching the load variation and exert anidealdamping effect Fig.(3.9).

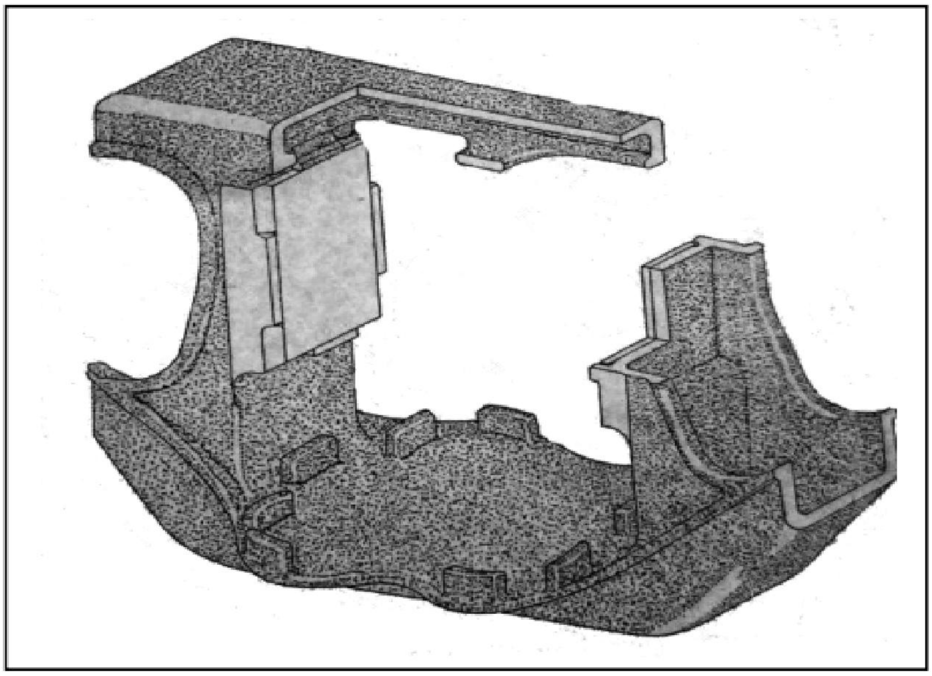


Fig.(3.9) Side frame.

Therefore, there can be no over damping in empty car condition or under damping at a load over what is fitting the constant friction value, often seen with the trucks of constant friction system having no special mechanism against the load variation.

S-2 BARBER STABILIZED TRUCK capable to obtain friction force completely following all possible load variations contrary to conventional constant friction trucks as above can be said as the most ideal truck for railway rolling stock from the logical point of view, too.

STABILIZED Advantageous for truck strength

The stabilizer of S-2 BARBER STABILIZED TRUCK is, as illustrated in the accompanying figures Fig (3.10) and Fig (3.11), installed in a reasonable construction to make it even more advantageous to reduce the weight of the truck, with no unreasonable construction from the point of view of the maintenance of strength of the bogie frame and bolster and requiring no need of increasing the thickness for maintenance of strength.

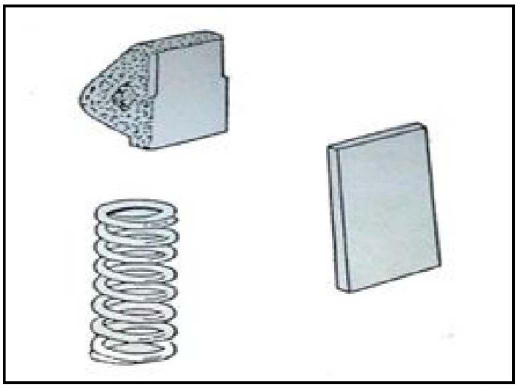


Fig (3.10) friction wadge, friction plate & snubbing spring.

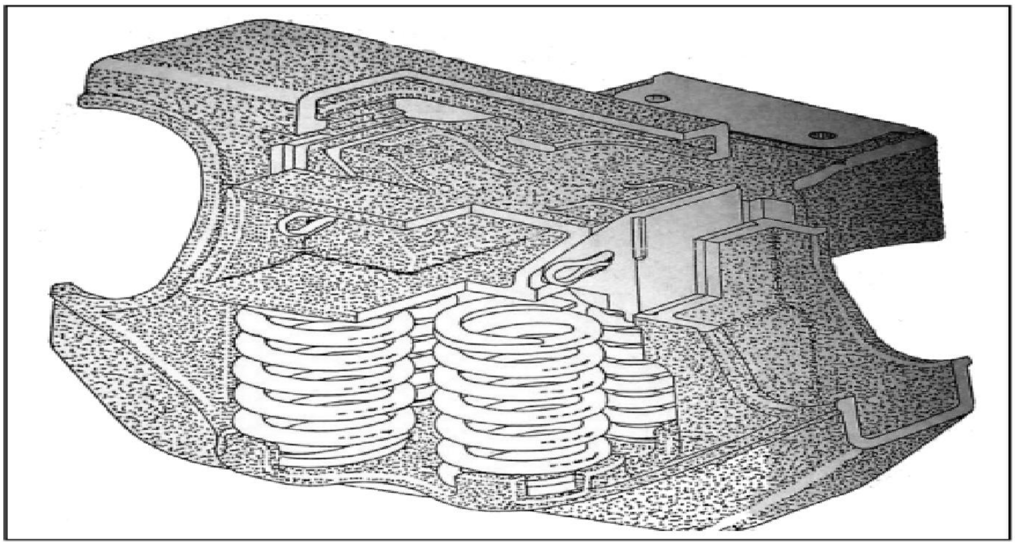


Fig.(3.11) side frame & bolster spring.

The stabilizer is designed in a very simple construction so that large savin in labor cost can be obtained from the reduce time required for assembling and disassembling and quantity of parts requiring exchange.

Unlikeothertrucks.S-2 BARBERTRUCK can be easily disassembled and Reassembled without requiring any special tools for the removal and installation of the stabilizer device Fig.(3.12).

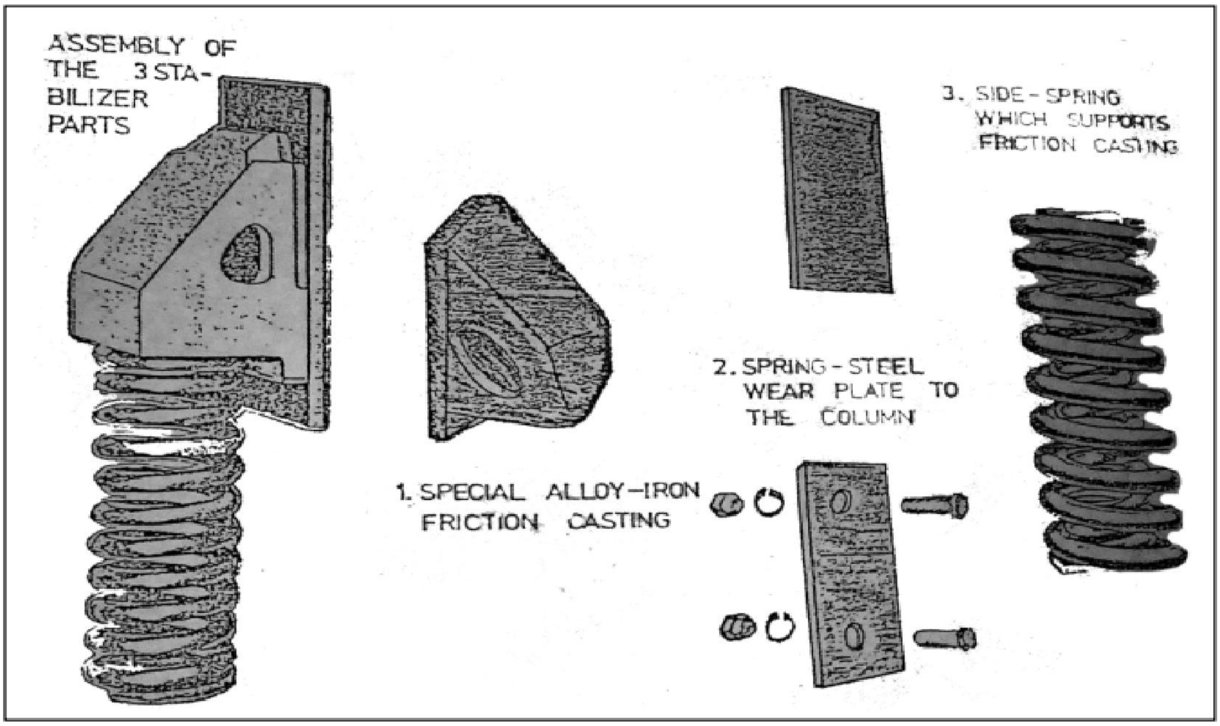


Fig.(3.12) Damping device for stabilised bogie.

To be more precise, disassembling of the truck can be made simply by lifting the bolster, and the side spring can be removed simultaneously together with the bolster spring. The friction casting is simply installed on the bolster spring by a special spring cotter.

Further, the exceedingly superior running performances of BARBER STABILIZED TRUCK can protect even easily damageable goods transports From destructive frictions and dangerous rolling for decreased claim for damaged goods and an increased credit on the railways using it. At the same time, it protects the car body, bogie frame and cast steel parts to allow them last longer. Thus the overall cost of maintenance of the car can be reduced from this side, too. Of course, the force working on the rails can be reduced to a great extent. For minimized rail damages and great saving in the cost of maintenance of the tracks concerned.

In addition to above, the fact that the working condition and damages of the truck can be observed easily from outside allows easy checking of the rolling stock for an effective safety administration of the train running, meaning a large saving obtainable in labor and cost in the face of railway operation, too.

Quick and easy assembling of truck

To assemble the truck..

1. First, place the friction casting at the specified position on the bolster, and then insert the special spring cotter to fix the friction casting to the bolster Fig.(3.13).

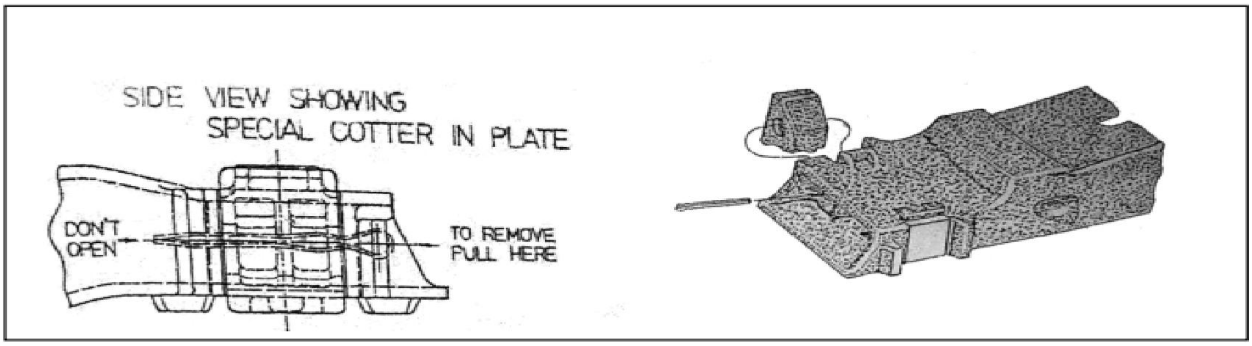


Fig.(3.13).Special spring coller.

2. Assemble the bolster with the friction casting fixed on it, to the side frame of the truck Fig.(3.14).

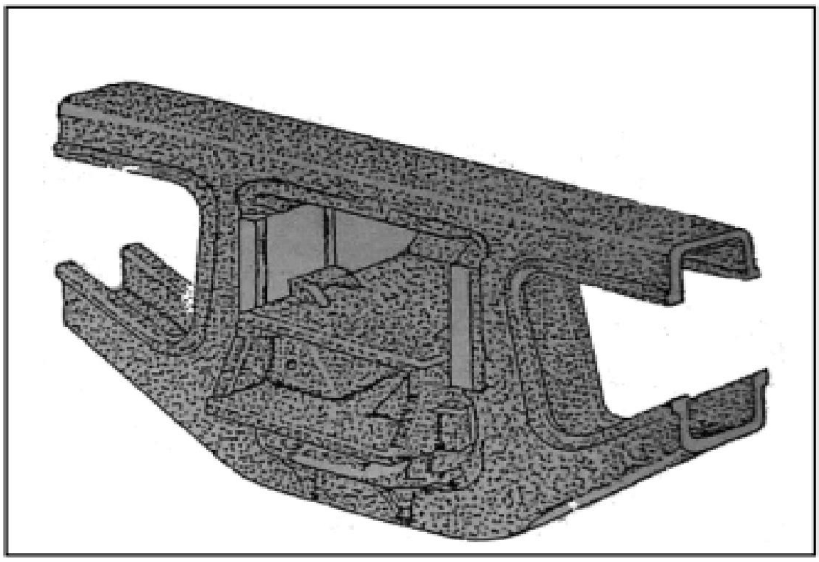


Fig.(3.14). Bolster with wedge into the side frame.

3. Raise the bolster up to the full length of the side frame and, then, after inserting the bolster spring and side spring, lower the bolster down on the spring to finalize assembling Fig.(3.15).

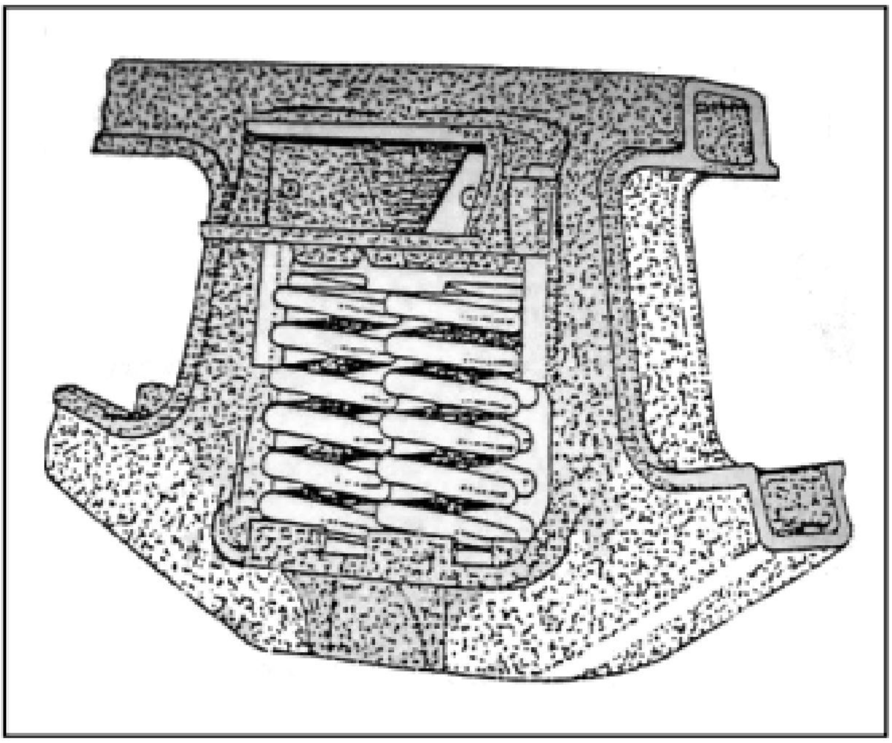


Fig.(3.15) Bolster spring nest allembled.

NOTE

The special spring cotter required for assembling and disassembling of the truck can be left unremoved from the truck assembly. Or it may be removed with no effects to the running of the truck Fig.(3.16).

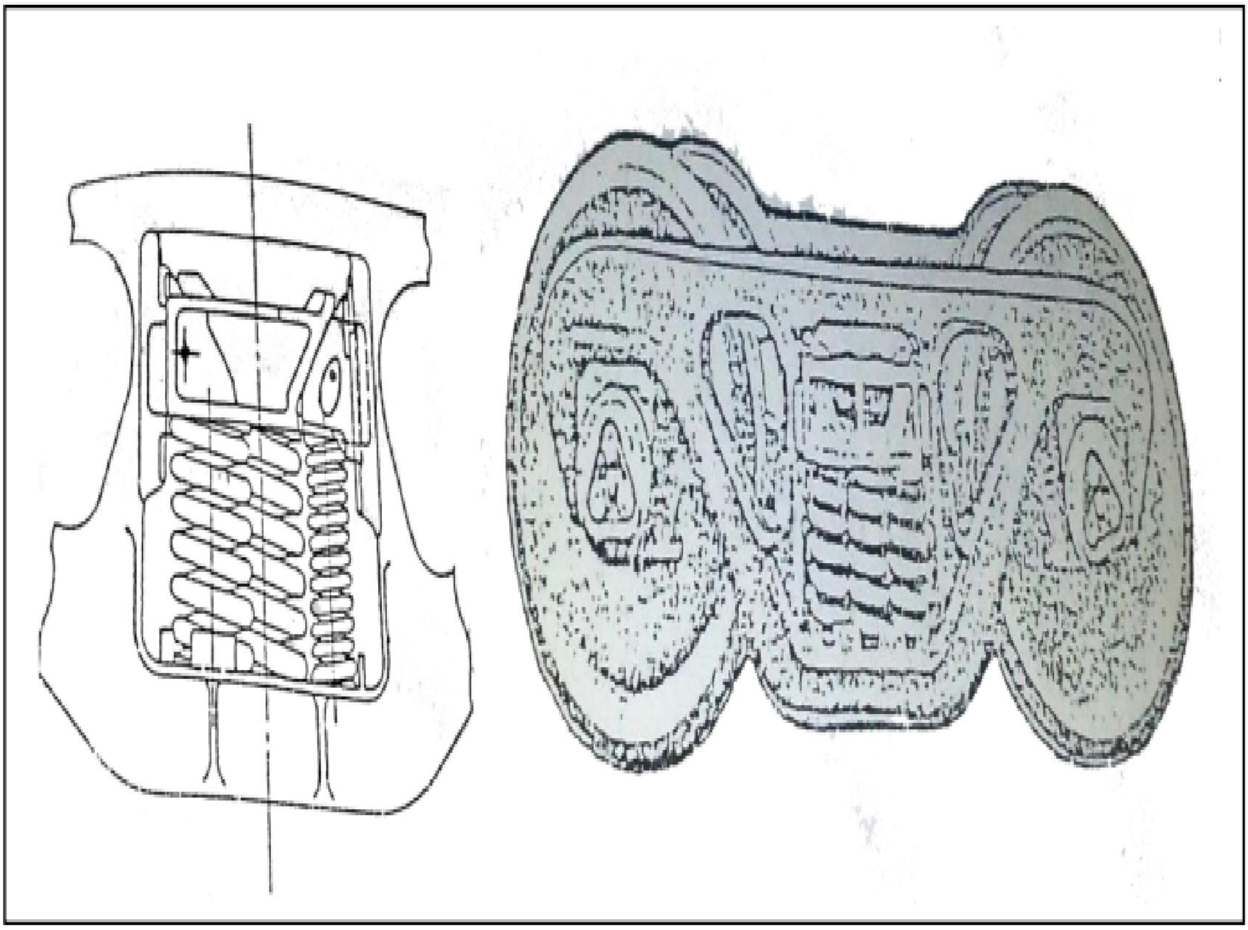


Fig.(3.16) Coller in its place.

As S-2 BARBER STABILIZED TRUCK is equipped with the coil spring of a longer travel, it can be bended in the lateral direction too, within the range of the stopper. And with the action of this spring and control by the stabilizer, the shock in the lateral direction can be absorbed to a large extent, contributing much to the improvement of the running performance of the car. As the flat surface of the bolster guide columns at four points per truck and the flat surface of friction members of the stabilizers are kept pressed against each others with the same pressure at the same time by the side springs Fig (3.), the swing bolster can be kept in a balanced condition at all times to avoid shake motion of the truck Fig.(3.17).

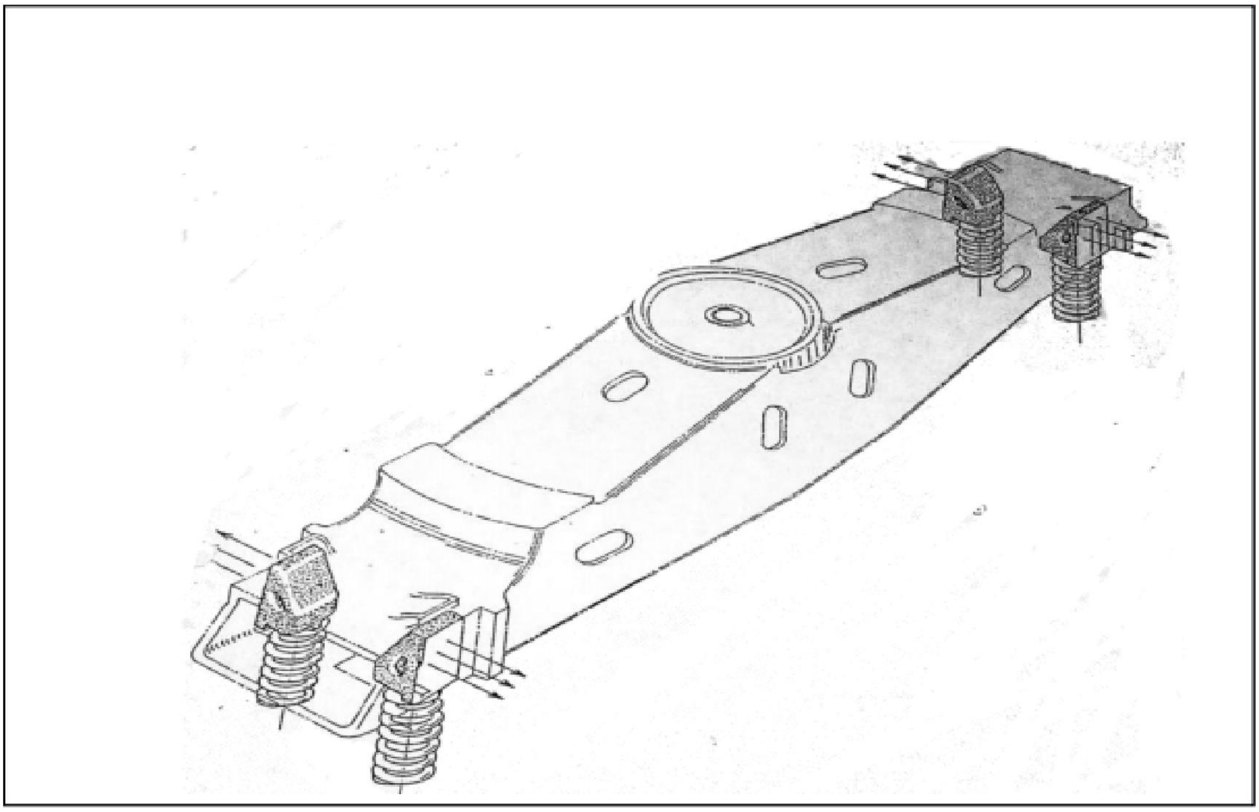


Fig.(3.17) Bolster balance.

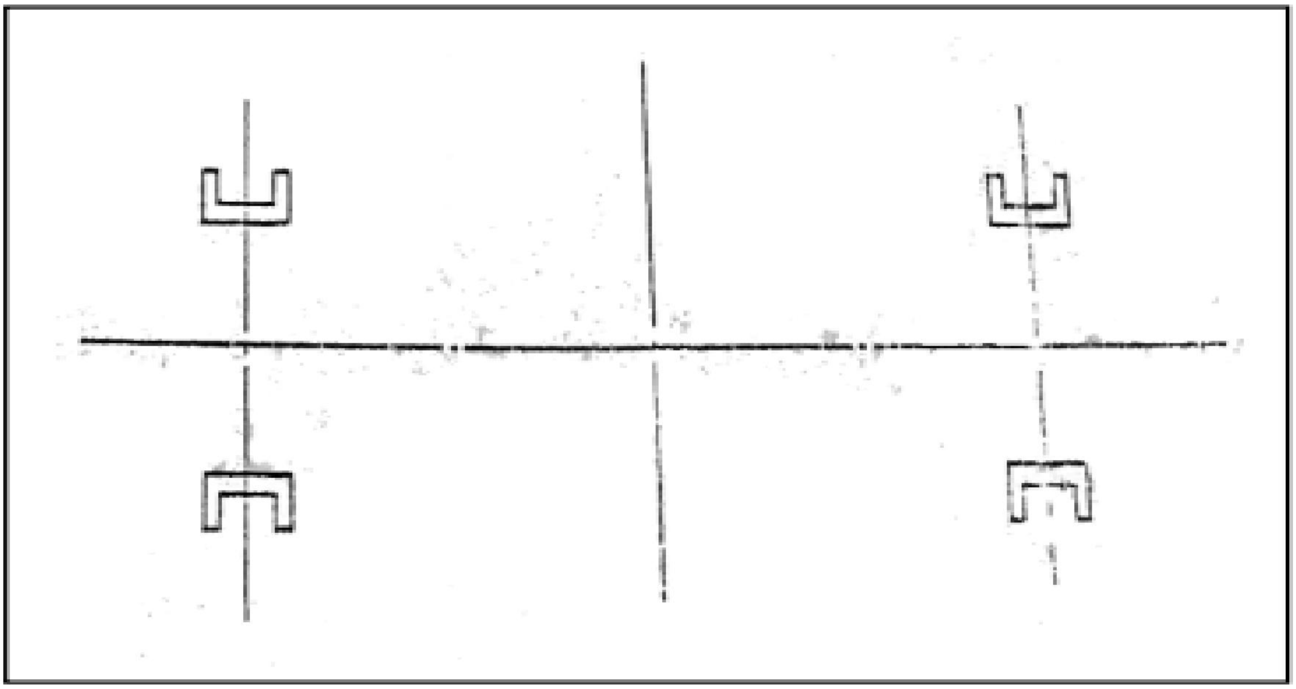


Fig.(3.18) Squareness of bolster.

3.4 Bogie desig[9]

Synopses

In the last two decades teams of dynamicists have played a leading role in the development of fundamental theories, supported by a complementary advanced analogue and digital computer techniques which now permit the quantitative assessment of many important aspects of vehicle dynamic performance not previously amenable to calculation. This study briefly describes the current position and outline the effect these new theories and techniques have had on bogie and suspension design, where possible comparing existing and previous practice.

3.4.1 Introduction

Consistent evolution pursued by a number of railways over the past decades has shown that speeds of 200 km/h are economically viable and even speeds of300 km/h are feasible. This development has placed a great responsibility on the bogie and suspensions engineers to produce hardware that is safe to operate, gives a high quality of vehicle ride and minimizes track forces.

Impetus to this evolution has been given by some railway setting up teams of dynamicists to explore the fundamentals of vehicle suspension performance. These teams have played a leading role in the development of theories which permit the quantitative assessment of many important aspects of vehicle performance not previously amenable to calculation. The development of these theories coupled with the complementary advance in analogue and digital computer techniques and the collection of much statistical track roughness data has provided the engineer with a new tool to explore new suspension design of advanced concept and performance.

3.4.2 Ride quality

ISO 2631 (1974) “Guide for Evaluation of Human Exposure to Whole Body Vibrations” has been accepted in many countries for passenger carrying vehicles. This international standard defines and gives numerical values for limits of exposure for vibrations transmitted from solid surfaces to the human body in the frequency range of 1 to 80 Hz. Three limits are defined comfort, working efficiency or fatigue, and safety or health. The primary quantity used to describe the intensity of a vibrational environment is the rms acceleration and this is weighted according to the frequency of oscillation.

At low frequencies and low amplitudes the human body may be represented as a linear lumped parameter system the important masses being: head, arms, upper torso, abdomen, hips and legs. Research shows that the body is least susceptible to vibrations in the 1-2 Hz range since this is experienced all the time when walking. The abdomen has a pronounced resonance in the 3-6 Hz range and this makes efficient isolation of a sitting or standing passenger difficult. Further responses occur at 20-30 Hz (head-neck-shoulder system) and 60-90 Hz (eyeball resonance) but it is the abdomen/thorax resonance which is shown to have the great effect and this is reflected in the ISO weightings, see Fig.(3.19 and 3.20) BR have extended the ISO weightings to cover motion sickness generally attributed to oscillation frequencies of less than 1 Hz. An instrument known as ajacobmeter is commercially available to measure in-service weighted rms accelerations.

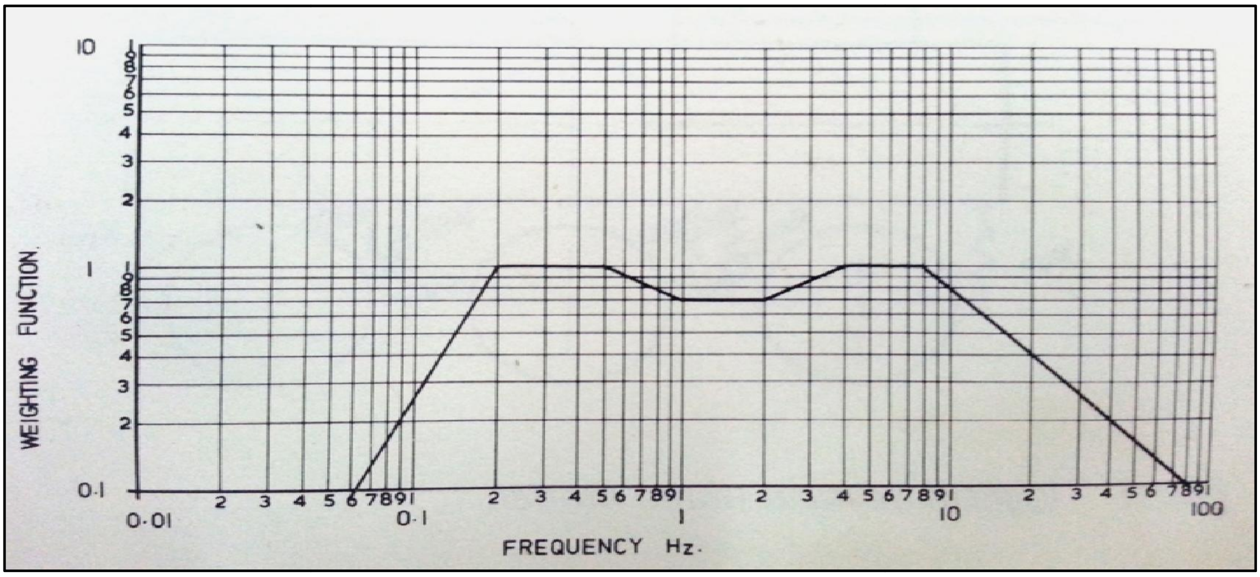


Fig.(3.19) ISO/BR acceleration weighting functions−vertical.

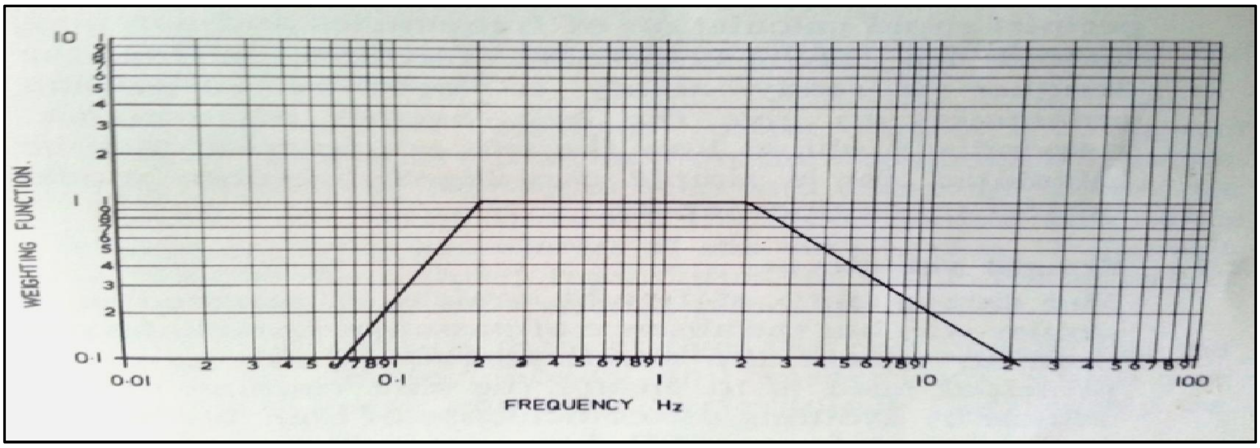


Fig.(3.20) ISO/BR acceleration weighting functions−horizontal.

The Jacobmeter is equipped with two accelerometers arranged to measure in the vertical and lateral planes; the output of the transducers is filtered according to ISO/BR weightings. The meter calculates the rms over 10-second periods and displays the reading which is logged by the operator. Usually the log is converted to a histogram and journey average rms calculated. An improved from of the instrument is currently under development incorporating a paper print-out facility and a small computer to continuously calculate journey average.

To determine the response of vehicle to track irregularities the designers generally use the random process theory where the track is characterized by its power spectral density (psd) which specifies the content of the irregularities (track deflection)^{2}as a function of spatial frequency (cycle / metre). Rail spectra can be used in special circumstances but generally a smoothed spectra is used which can be represented by some mathematical formula. The vehicle response can then be calculated using the well-known formula which states that the power spectral density of the vehicle response co-ordinate (acceleration, say) can be obtained by multiplying the track psd at each frequency by the square of the vehicle frequency response functions—the area under the acceleration response psd leads directly to the rms value of vehicle acceleration.

To calculate the response of a vehicle to random track roughness the designer performs an Eigen vector analysis using a digital computer and a linearised set of equation of motion, see Fig.(3.21) for typical vertical plane model 12 degrees of freedom. The computer permits rapid calculation of frequencies and damping factors for each suspension freedom, giving the designer a ready ability to optimize suspension parameters to give the best overall performance.

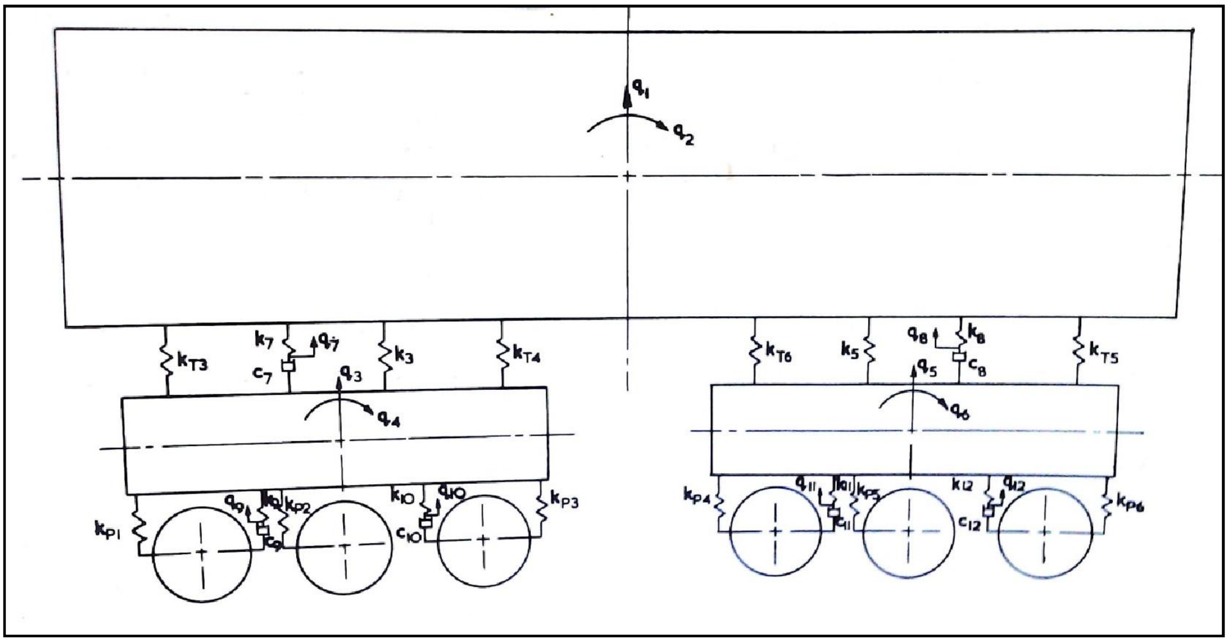


Fig.(3.21)Typical vertical plane model 104.

3.4.2.1 Freight vehicles

The ride quality of freight vehicles is assessed in a similar way but the absence of passengers precludes the need to weight the accelerations. In this case the principal need is to ensure the safe running of the vehicle by avoiding the coincidence of high lateral and low vertical forces. BR have recently introduced a procedure involving the running of vehicle under test over a predetermined route some 260 km long which includes track of all standards. The test vehicle is fitted with accelerometers measuring the oscillation in the vertical and lateral planes, the output of these transducers is subject to peak count zero crossing analysis and the resulting histogram compared against a predetermined limit, see Fig.(3.22 and 3.23) which show typical test results compared against the limit curve for two planes.

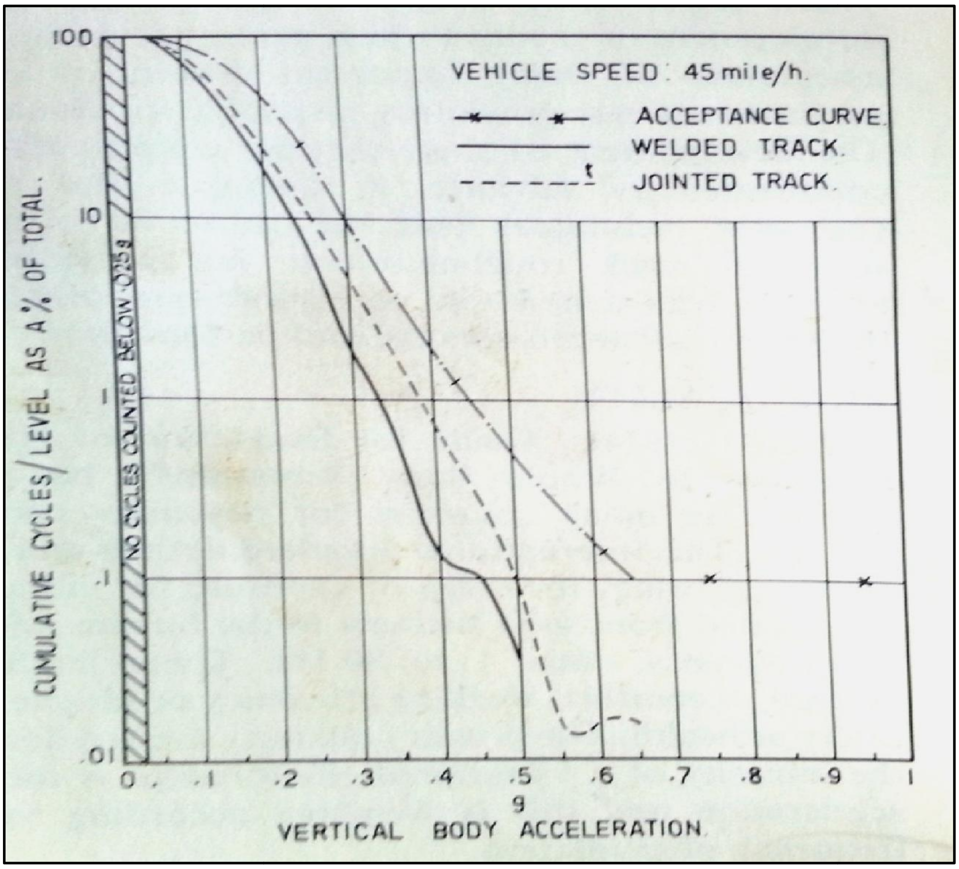


Fig.(3.22) Typical test results of test vehicle accelerometer reading – vertical – compared against the limit curve.

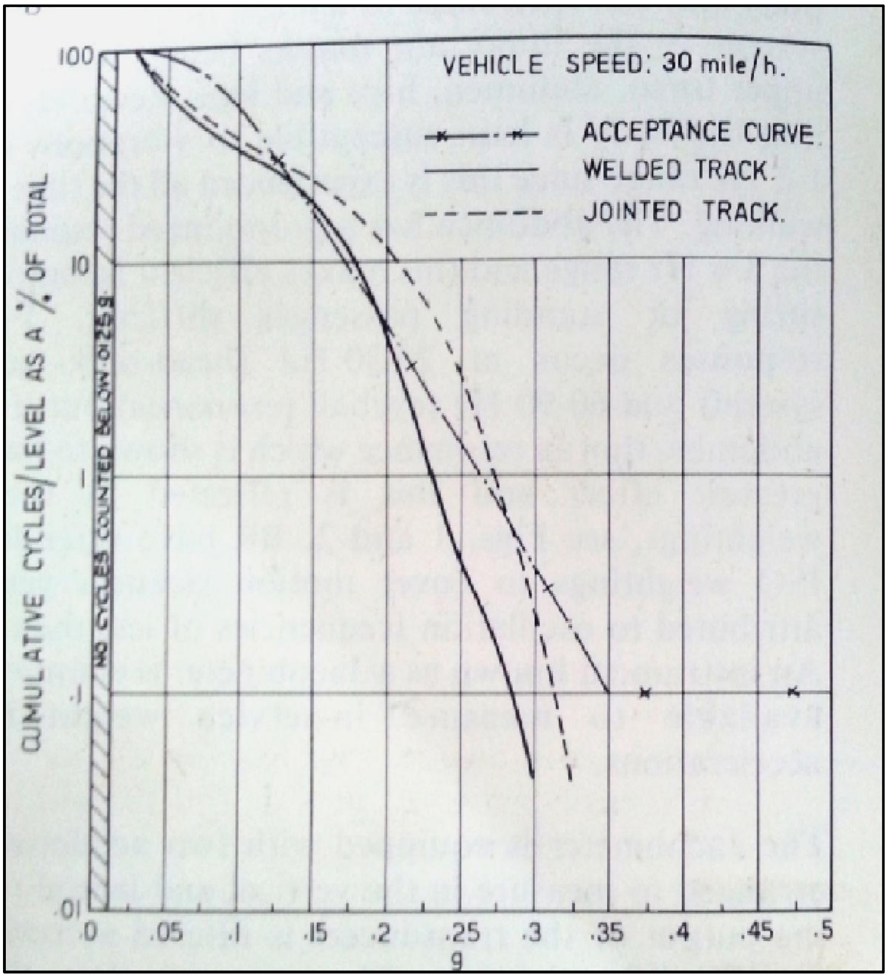


Fig.(3.23) Typical test results of test vehicle accelerometer reading – lateral – compared against the limit curve.

3.4.3 Latral stability

Sustained lateral oscillations experienced by some vehicles, usually referred to as hunting, and has presented problems to engineers from the early days of the railways. Empirical approaches to the problem usually ignored certain important factors. The development of creep theories and a greater appreciation of the importance of the mutual wheel/rail geometry has led to new theories which incorporate all relevant factors.

The linearised theory is most widely used for stability work but for curving calculations a non-linear method must be used. The linear method consider conventional railway wheelsets in which two wheels are rigidly mounted on a common axle. Each wheel has a coned or otherwise profiled tread and a flange on the inside of the rail. A flange way clearance is provided so the wheelset has available a lateral movement before flange contact occurs. Assuming rigid track for motion at constant speed the only other wheelset freedom is yaw.

The important geometrical features relate to the behavior of the contact point on each wheel as the wheelset is displaced laterally. For a purely coned wheelset the contact angles are substantially constant and the rolling radii of each wheel are directly proportional to the lateral displacement of the wheelset. For a hollow wheel profile, however, they become functions of the mutual geometry between wheel and rail and computer methods have been developed to determine the basic linearised relationship for such condition.

To determine the wheelset tangential forces and their effect on the motion, creepage theories are used. Many workers have contributed to such theories which generally assume that the contacting bodies (wheel and rail) are elastic without interlayer–coulomb friction behavior being assumed in the slipping region. For small creepage and spin the force/creepage relationships for longitudinal, lateral and spin creep are linear but at large creepage the whole contact degenerates to the case of pure sliding. BR use Kalkers creep coefficients but these are scaled down since experience has shown the full coefficients can only be obtained in practice where the contacting metal surface are scrupulously clean. In reality surface contaminations reduce the force levels.

Linearised Equations of motion for the wheelsets have been derived. For a simple bogie analysis some six equations of motion are involved–yaw and lateral motion of each wheelset and also of the controlling bogie frame. For more complex analysis used particularly on high speed or novel vehicle a full vehicle may be represented by some fourteen equations although usually many more freedoms are considered (see Fig.(3.24) which shows a typical twenty-seven degree freedom model), each equation derived considers the various masses to be connected by triaxial spring and damper elements.

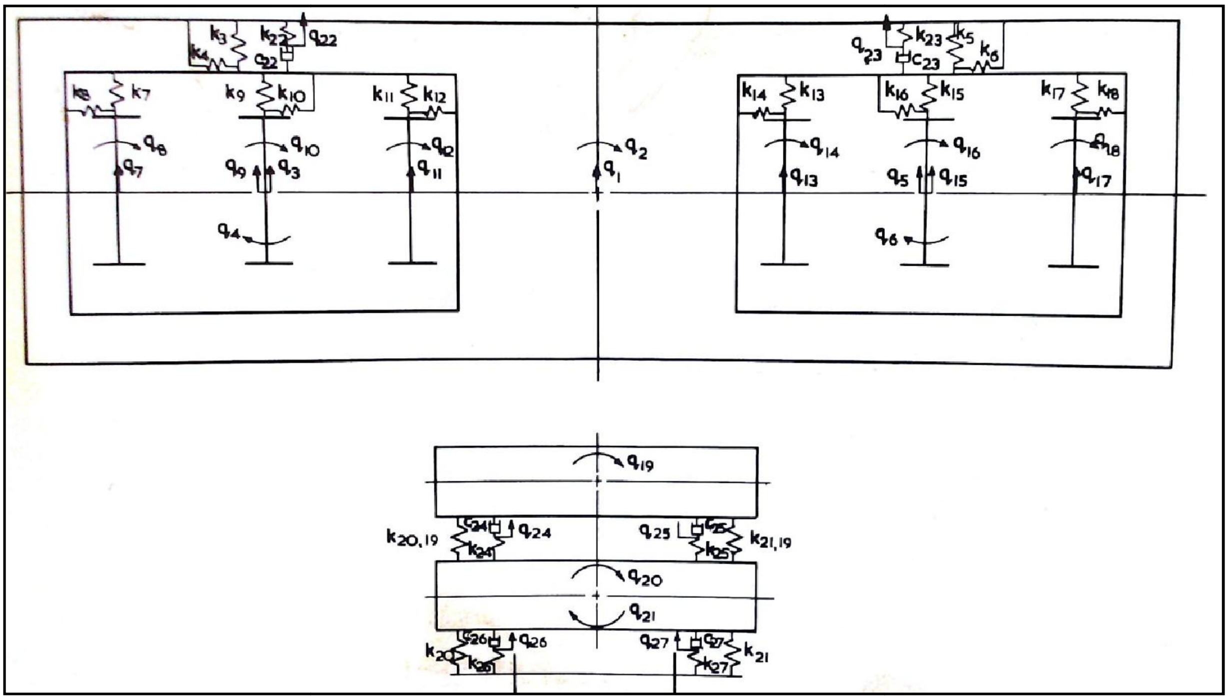


Fig.(3.24) Typical twenty-seven degree freedom model (lateral and roll)

Eigen vector analysis using digital computer programmers is performed on the equations of motion to determine the frequencies and damping factors for each suspension freedom, the mode shapes and hence the stability boundaries of the vehicle (Fig.(3.25)shows plot of the stability boundaries for two extreme values of creep and two extreme values of conicity).

The use of such theories over the past decade has led BR to develop bogie and suspensions which are largely linear and thus performance can readily be predicted.

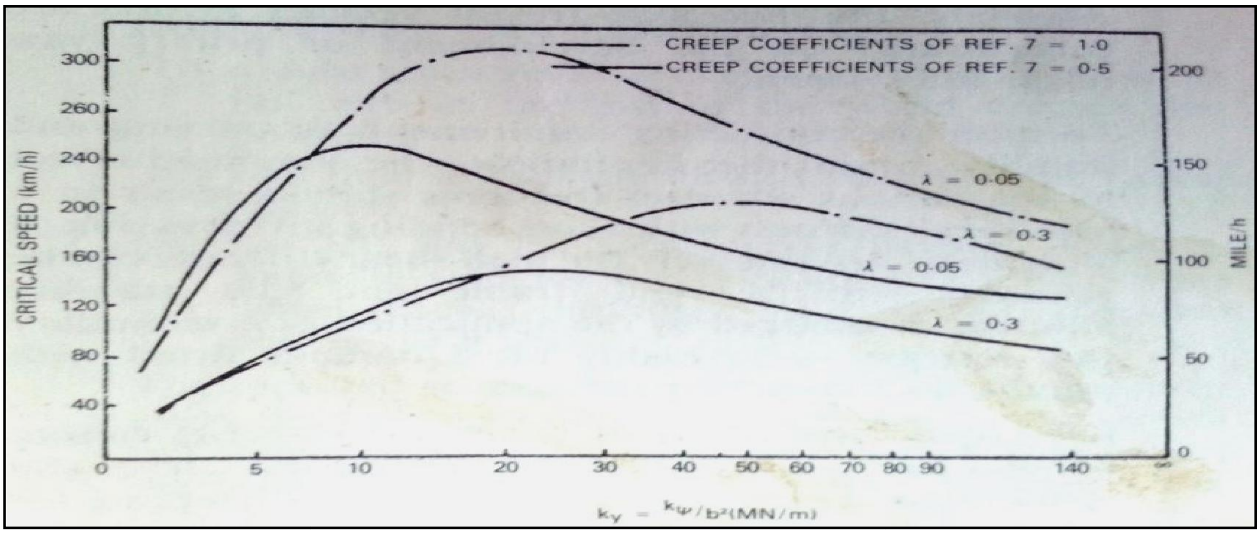


Fig.(3.25) Critical speed of bogie hunting with variation of creep coefficients.

3.4.4 Curving theory

The concept of creep, conicity and suspension flexibility outlined above are also fundamental to the behavior of railway vehicle in curves. Clearly, if a flexible vehicle can be provided with stable guidance on straight track the flexibility can be used to improve performance on curves. In practice however the requirements for dynamic stability at high speed conflict with those of slow speed curving on small radius curves. Stability of conventional linear bogies and suspensions generally requires stiff wheelset yaw freedom while curving demands gross yaw freedom and operates in the non-linear range of wheel/rail contact condition.

Early linear steady-state curving theories clearly demonstrated that a free wheelset with coned wheels can traverse a circular curve in pure rolling by aligning itself radial to the curve and displacing outwards so that the conicity provides the required difference in rolling radii between the wheels to avoid longitudinal creepage.

With wheels coned within the normal range used on railway vehicle, however, the minimum curve on which pure rolling could be achieved was rather large the linear theory is therefore, conceptually, only correct for modern vehicle traversing main line curve.

For small radius curves BR have developed a non-linear theory which can handle the variations in conicity of modern or worn wheel profiles as the wheelset moves laterally across the rails and adjust the creepage conditions applications applicable accordingly. The theory is still subject to refinement as more experimental information becomes available but it has provided the impetus for many administrations to develop bogies and suspensions with improved curving characteristics.

The theory has assisted in the development of the three piece bogie – the addition of controlled wheelset freedom and intercouplers has greatly increased the stability margins of such bogies and greatly reduced the wheel/rail lateral forces in curves.

Many administrations are developing bogies for rapid transit and suburban type applications with improved curving characteristics. Two basic types are under evaluation: (1) a three piece bogie with inter coupled wheelsets and (2) a rigid frame bogie with soft wheelset yaw control− stability being achieved by a suitably designed secondary yaw restraint.

In the locomotive field engineers are concentrating on a primary yaw relaxed suspension and inter coupled wheelsets. For two axle freight vehicles efforts are concentrated on the development of primary yaw relaxation systems.

To meet the conflicting requirements of curving and stability current theories indicate the combined effect of the various wheelset freedoms should result in a high shear stiffness with a low bending stiffness, Fig.(3.26) illustrates. Alternatively the shear stiffness can be achieved with rigid frame and the stability boundaries enlarged by the application of a secondary yaw restraint– a highly rated damper fitted with suitable end mounting stiffness.

Even with these forms of bogies the wheelset cannot adopt a true radial position on the smallest curves and some researchers are investigating forced steering for rapid transit applications where the number of very small radius curves is significant.

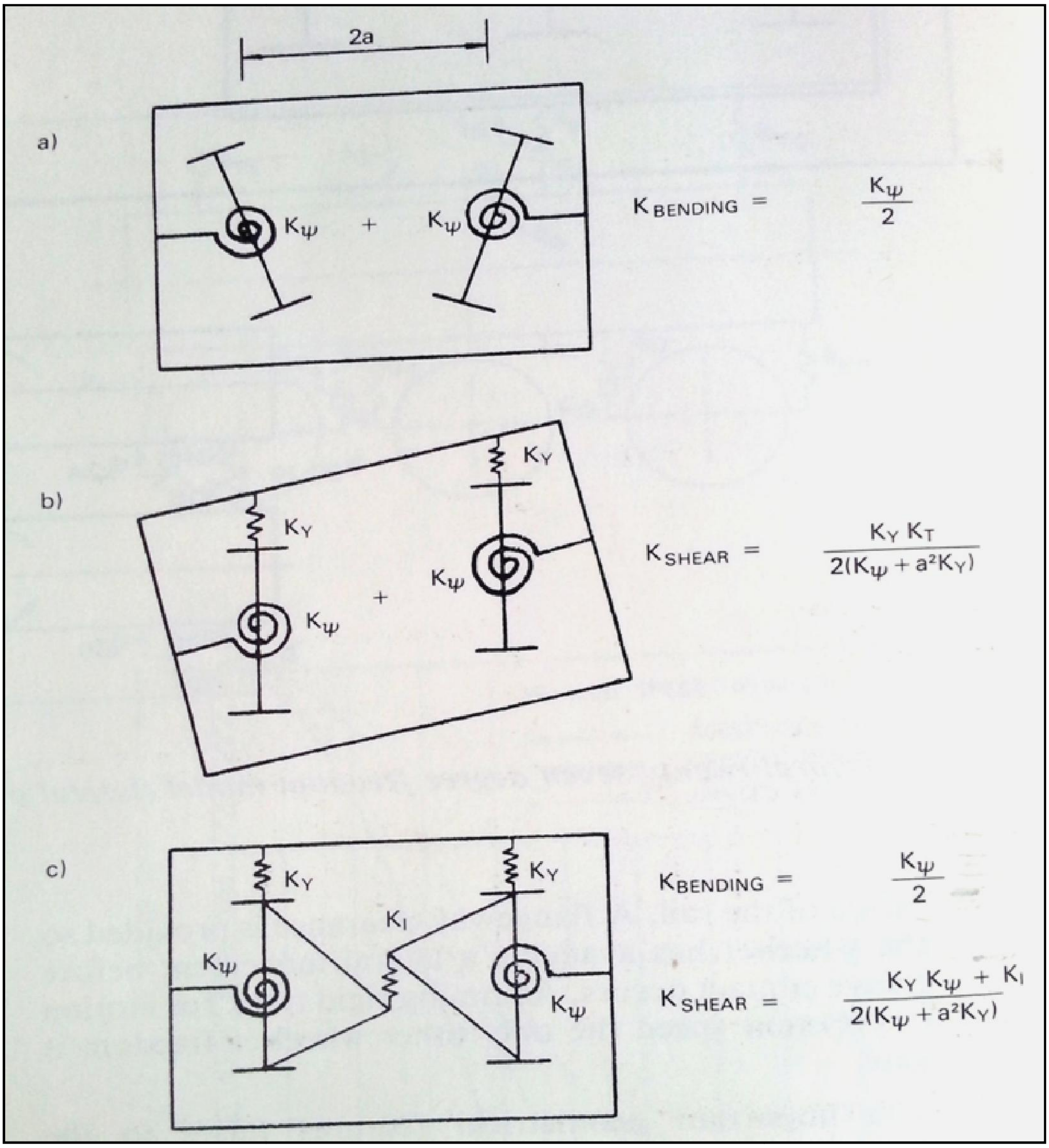


Fig.(3.26) Diagrammatic illustration of Bending and Shear stiffness of a freely pivoting bogie.

3.4.5 Track forces

The major causes of increases in the vertical force between wheel and rail are:

i. Isolated irregularities in the rail running surface which occur by default at joints and welds and by design at points and crossings.

ii. Periodic irregularities such as corrugation on the rail surface on the repetitive effect of sleeper spacing.

iii. Random variations in track top level, i.e. track top roughness.
iv. Defects in the vehicle such as wheel flats and wheel eccentricity.
v. Random variations in sleeper support stiffness such as hard spots or voids.

Computer programs have been developed to cover all these force mechanisms but the spectrum of forces experienced by the track covers a wide range of frequencies and it has been necessary to develop different methods of analysis with differing complexities of the vehicle and track models to calculate the response to the various forms of excitation. At very low frequencies− below 10 Hz – the variations in track loading are due almost entirely to variations in suspension forces caused by the various bogie and body motions. The track system has a relatively small effect and may be modeled simply in many instances the track is assumed to be rigid with a particular geometric form. At intermediate frequencies, 20-100 Hz, the track system plays a significant part in determining force levels while only the vehicle unsprungmass and its primary suspension have some influence. At high frequencies, 500 Hz to 2 KHz, encountered during impacts the response is largely dependent on the wheelset and rail masses, and the elasticity of the wheel/rail contact zone.

3.4.5.1 Dipped rail joints

Despite the continued increase in the mileage of continuous welded rail (CWR) the increasing number of rail end failures on BR in the early 1960s gave rise to serious concern. It is not surprising, therefore, that the forces and stresses to a dipped rail joint was one of the first features to be studied. The idealised form of the rail top profile at a dipped joint is shown in Fig.(3.27) together with the usual model used in the calculations.

Results of typical calculations are shown in Fig.(3.28) (a) to (d). The force/time history shows that a peak occurs some ¼ to ½ms after crossing the joint. This peak is of high frequency and is a result of the unsprung mass colliding with the rail mass and vibrating on the hertzian contact stiffness between wheel and rail. This peak force has been designated P1. It is this force which is directly responsible for rail-end batter, i.e. the spreading of the rail-head contact area immediately adjacent to a joint.

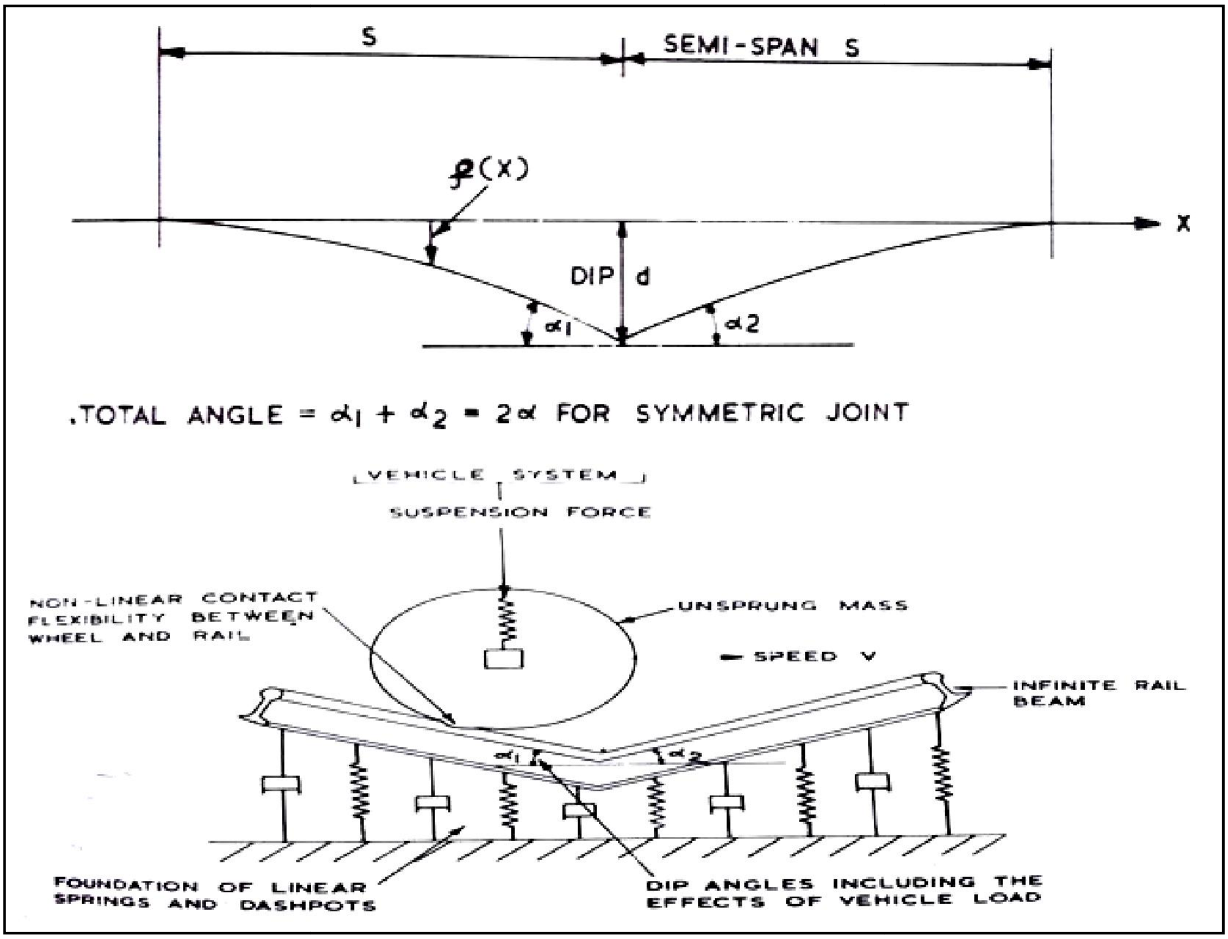


Fig.(3.27) Idealised from of dip used for calculations.

The force/time history also shows a second peak, designated P2, which occurs several milli-seconds after crossing the joint. At 45 ms (100 miles/h) this peak coincides with the first running on sleeper. Unlike the P1 force which is largely reacted by the rail inertia, the P2 force peak is transmitted via the sleeper to the ballast producing a corresponding peak track displacement as shown in Fig.(3.28) Thus the P2 force is responsible for the increased sleeper loads at rail joints and is the cause of ballast damage and track up to deterioration.

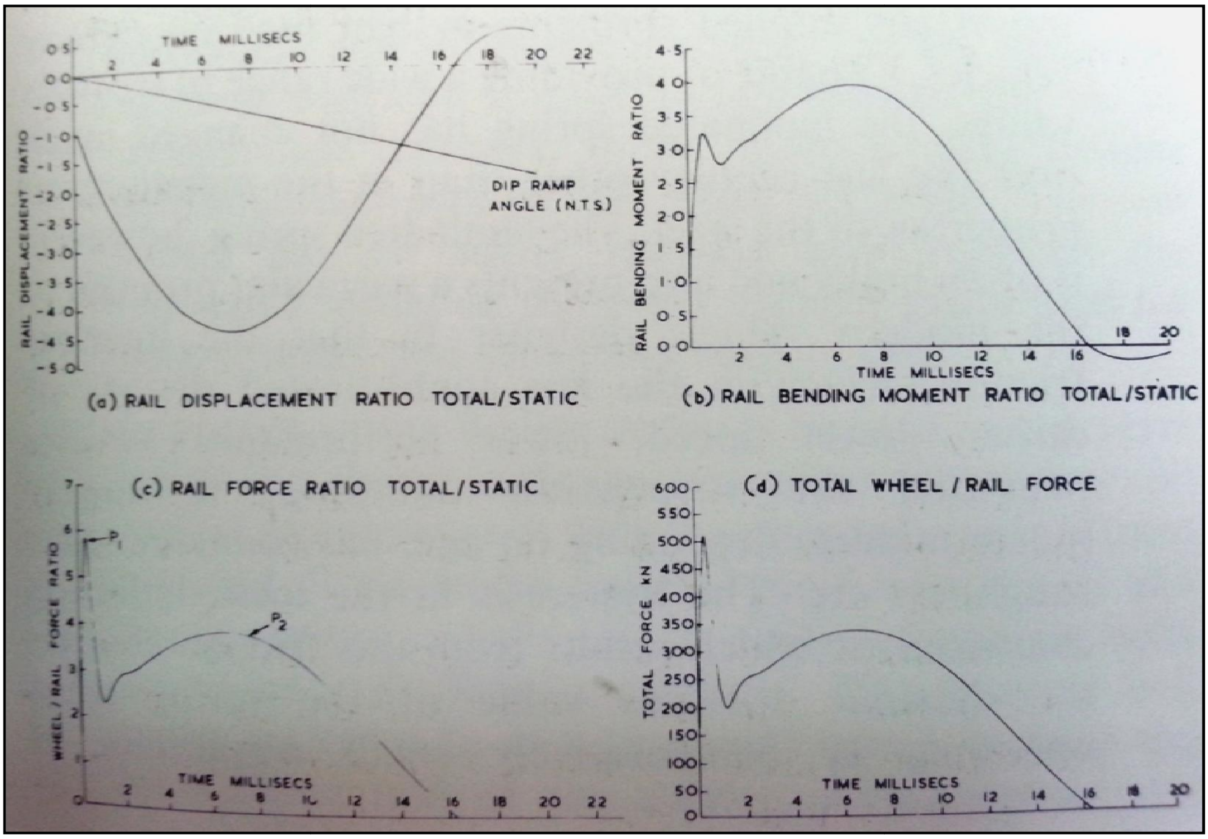
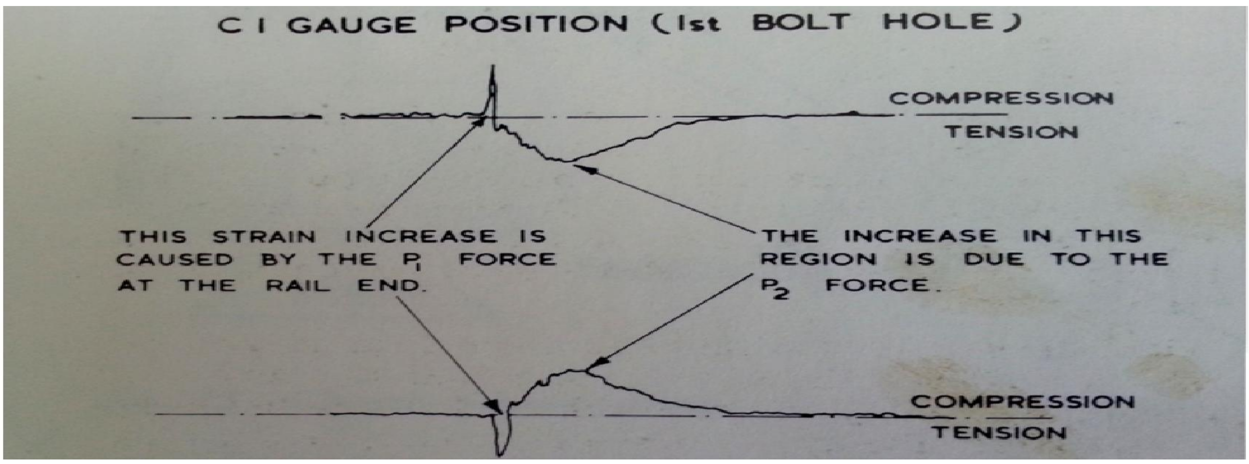


Fig.(3.28) Typical responses to dipped rail joint.

The combination of the P1 and P2 forces govern the stress range experienced by the rail web adjacent to the fishplate bolt holes and hence largely determine the fatigue life of the rail ends. At the first bolt hole the stress range depends stongly on the combination of the P1 and P2 forces while at the second bolt hole the range depends almost entirely on the P2 force alone.

Experimental measurements of the strains at the bolt holes of a joint are shown in Fig. (3.29).



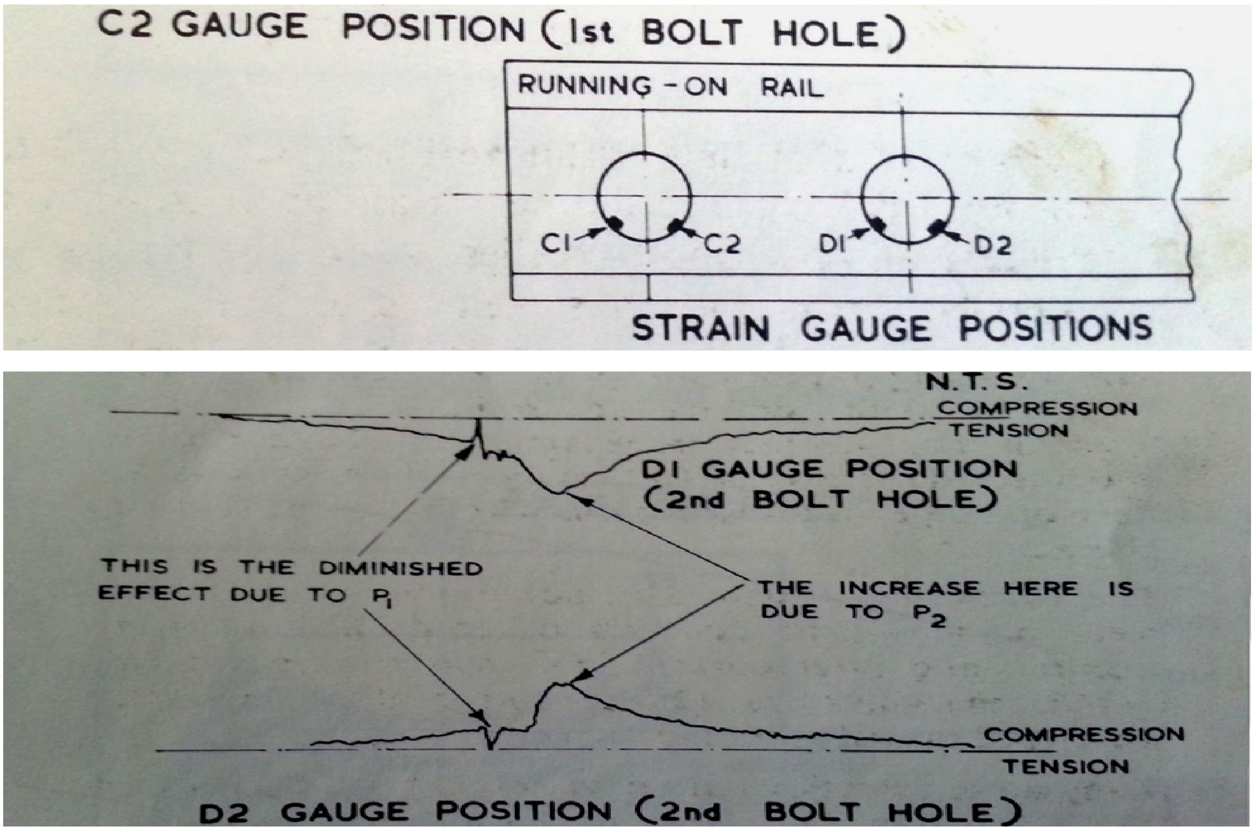


Fig.(3.29) Bolt hole strains.

Although fatigue crack growth is governed by the total stress range, final fracture of a rail end is due solely to the levels of tensile stress. Reference to Fig.(3.29) shows that the P1 and P2 effects produce bolt hole stresses of opposite signs. This is because:-

i. A shear stress reversal occurs at any position in the rail web at the instant the wheel crosses that position;
ii. Bolt hole stresses are directly related to web shear stresses;
iii. The P1 and P2 force peaks usually occur on opposite side of the bolt holes (P1 before the wheel reaches the holes, P2 after the wheel has passed the holes).

The repeated sequences of P1 and P2 loadings under every train are undoubtedly responsible for the growth of fatigue cracks in the bolt holes. These forces are not necessarily the sole cause of the final fracture which occasionally occurs in undetected cracked rail ends.

The occasional vehicle with a large wheel flat striking the rail end may be responsible for at least some of these fractures.

3.4.5.2 Parametric studies for dipped rail joints

The general purpose program has been employed to study the effects of various track and vehicle parameters on the levels of the P1 and P2 force peaks, the rail bending moments and the peak rail head contact stresses. Some results are presented in Fig.(3.30 & 3.31). Fig.(3.30) shows that, to a close approximation, the P1 and P2 force peaks are proportional to the product of the total joint angle (2 α) and the speed (v). This is also true for bending moment. The angle (2α) is defined in Fig.(3.26) as the sum of the angles between each rail end and horizontal. In practice this angle consists of two components – one due to permanent deformation of the rail end, the other due to deflection of the joint under load. It is this total angle and not the total dip of the joint which governs the magnitude of the track forces and stresses produced in the immediate vicinity of the joint. Dip depth in important, but only from the vehicle response aspect. Fig.(3.31) illustrates the P1 and P2 force dependence on vehicle unsprung mass, track stiffness and track mass. Suspension stiffness and damping play a large part in limiting low frequency body and bogie motions but their influence upon the P1 and P2forces is negligible.

Although the foregoing has concentrated on the P1 and P2 forces generated by rail joints, similar forces in form and magnitude are developed at weld irregularities particularly stepped or raised welds where a short ramp is effectively built into the rail.

To reduce the P1 force at such irregularities the vehicle designer has only two courses open to him – reduce the unsprung mass or fit resilient wheels – effectively reducing the rail support stiffness. BR have used both approaches: all new traction vehicle intended for operation at speeds in excess of 160 Km/h are fitted with frame mounted traction motors and whole fleet of Class 86 Locomotive are currently being fitted with resilient wheels.

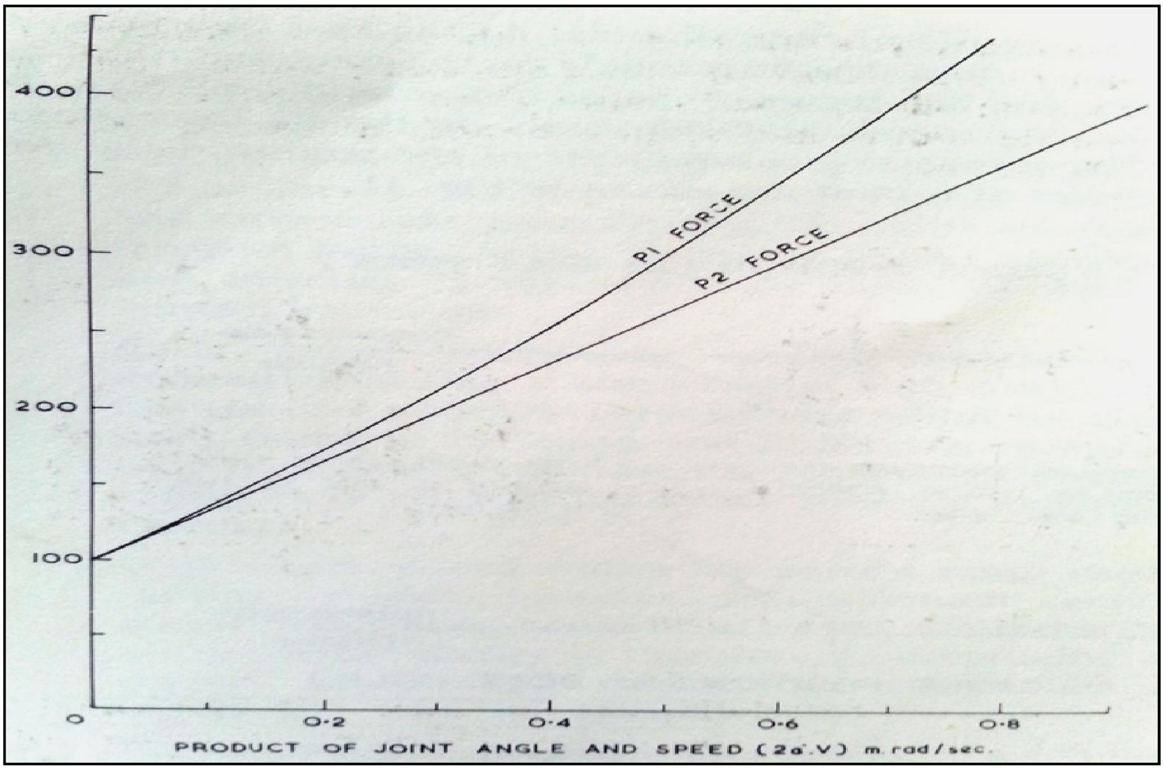


Fig.(3.30) Variation of P1 and P2 force peaks with ($speed \times joint\ angle$).

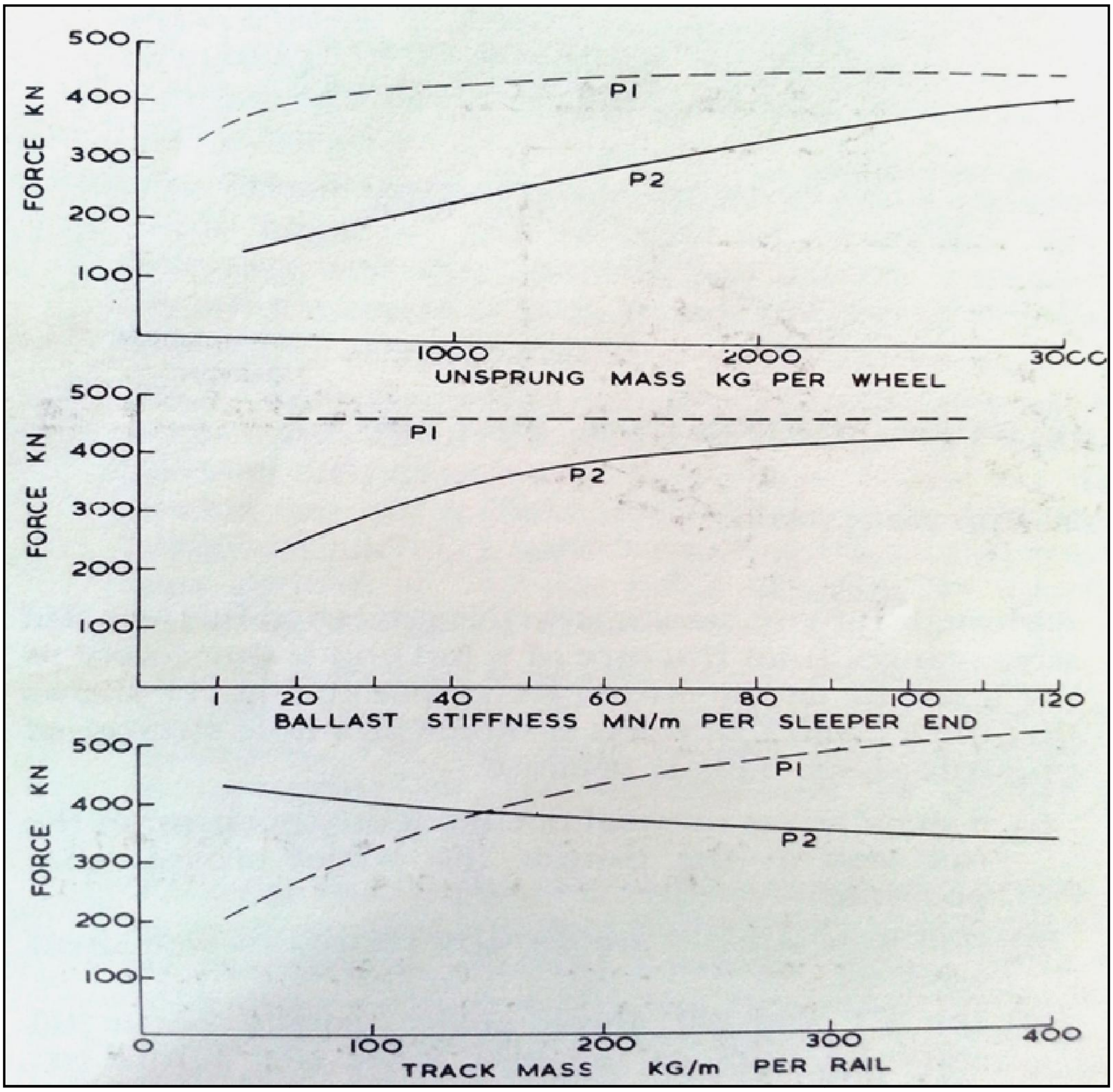


Fig.(3.31) Variation of P1and P2 force peaks with unsprung mass, track stiffness and track mass.

3.4.6 Suspension components

To engineers with knowledge of bogie types the number of different combinations of suspension components will appear very large, indeed to cover all combinations would lead to the presentation of all extensive catalogue without really clarifying the true purpose, advantage and disadvantage of each component. The development of the theories and computer techniques covered in the first part of the study, however, has enabled engineers to see more clearly the real advantages etc. of the various individual components currently available.

3.4.6.1 Springs

Springs are an essential part of any suspension, their ability to store energy providing the necessary shock absorbing properties. Various forms have been used by engineers but the following are the most widely used for railway applications:-

Laminated springs

Laminated springs are historically probably the most universally applied springing system used on railway vehicle. Capable of providing a wide range of stiffness values, the laminated spring has not changed much over the last century other than in the metallurgical properties of the steel; the laminated spring , however, is often bulky and also presents a particular problem to the modern vehicle designer in that the interleaf friction, which in the less sophisticated designs of earlier, lower speed, lower performance vehicles provided the suspension damping, is largely indeterminate, depending on age, environment, plate roughness etc. the hysteresis in the load/deflection characteristic which results from this friction change the dynamic stiffness value of the spring when operating at comparatively small amplitudes of deflection.

Tapered leaf spring

Recognizing the limitation of conventional laminated springs some engineers have devoted some effort to develop alternatives. Two types are available in the market today — both are aimed at the freight vehicle market.

The Taperlite spring is an improved from of leaf spring: each 'leaf' is tapered and rolled to fine tolerances to a good finish, each leaf is subject to strain-peening, i.e. shot peening with the leaf in a state of strain; the springs are also pre-set to introduce favourable residual strains. The net result is a construction which is much lighter than conventional leaf springs due to the higher permissible working strains. Where multi-leaf taperlite springs are required care is taken to avoid interleaf contact except at the essential load points. This method of construction gives the spring a low hysteresis characteristic and it is usual in suspensions employing these springs to install additional damping. The tapered leaf spring introduced by Bruninghaus follows similar lines but the important feature here is the pre-loaded clamp across a number of leaves which introduces a controlled level of interleaf friction. Such designs are now in service without the need for additional installed damping.

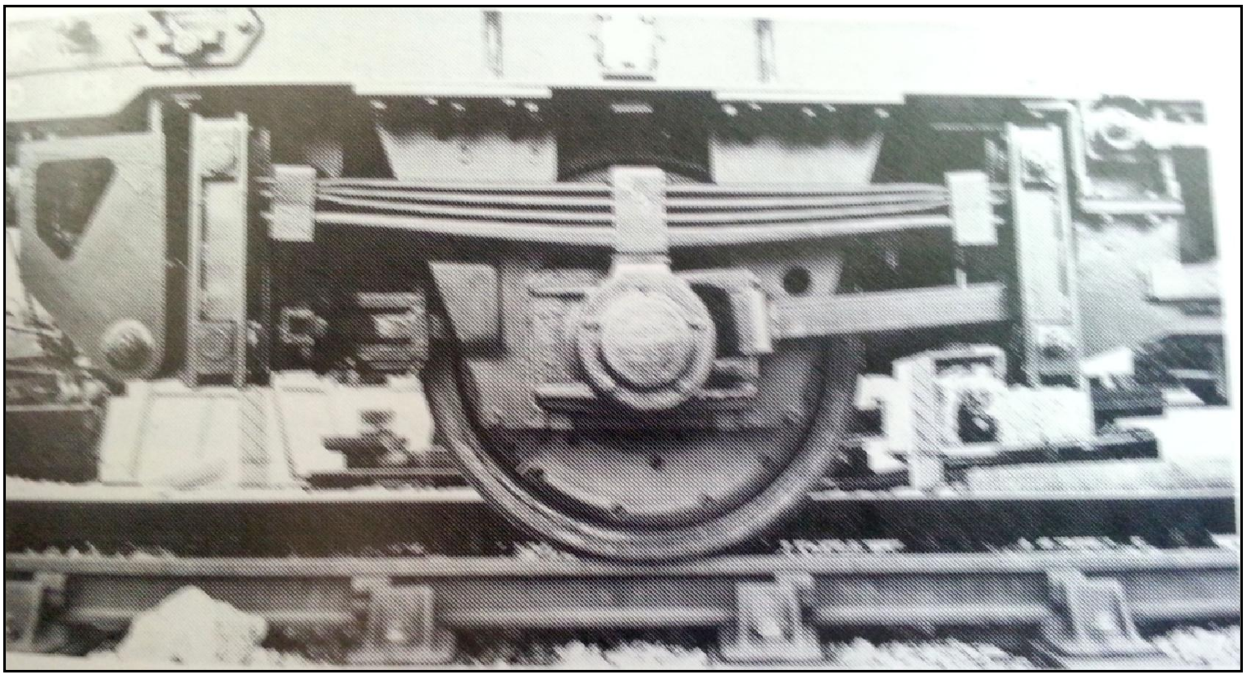


Fig.(3.32) Tapered leaf/long link suspension as applied to BR Hopper Wagon.

Coil springs

The coil spring is compact and accurately reproduces the characteristics required by the designer. Modern design methods usually result in springs which have a very long life and low incidence of failure. This form of spring is usually utilized to provide resilience in the axial direction only. Where coil springs are used to provide the vertical resilience in secondary suspensions, it is therefore usual to have a swing-link bolster, to permit the lateral body-to-bogie displacements, and centre or side bearers for rotational movement. In primary suspension, where lateral and longitudinal displacements are smaller and the required stiffness in these planes very much greater, axlebox guidance systems which provide suitable movement and stiffness are used. The low hysteresis inherent with coil springs usually requires the use of independent dampers, in the more modern designs hydraulic dampers usually being preferred. The coil spring has on many railways now replaced the laminated spring on passenger vehicle and is used for both primary and secondary suspensions.

Improved spring design methods have in recent years led to the ability to design secondary suspension coil springs to provide not only the required axial stiffness rate and displacement but also required axial stiffness and displacement radially, i.e. to control bogie to body lateral and rotational movement. This form of coil spring, referred to as a ‘flexicoil’ spring, permits the total elimination of the bolster, with a consequent considerable reduction in bogie weight and wearing components.

Rubber spring

Rubber springs are becoming an increasingly popular solution to primary suspension designs, especially for suburban and rapid transit vehicles. Numerous forms of rubber springs may be designed, in many of which it is possible to stress the rubber elements simultaneously in shear and compression. Thus, by arranging the rubber in a particular manner, it is possible to obtain the required stiffness values in two and sometimes three planes, unassisted by other means.

One such suspension (Clouth) is shown in Fig.(3.33) where a combination of compression and shear of the rolling rubber element provides the vertical stiffness rate and compression of the element by radial displacement of the suspension provides the wheelset lateral and yaw stiffness – in this instance at the same value. This form of rubber primary suspension has been applied experimentally on a number of BR vehicle and successfully tested at speeds up to 200 Km/h.

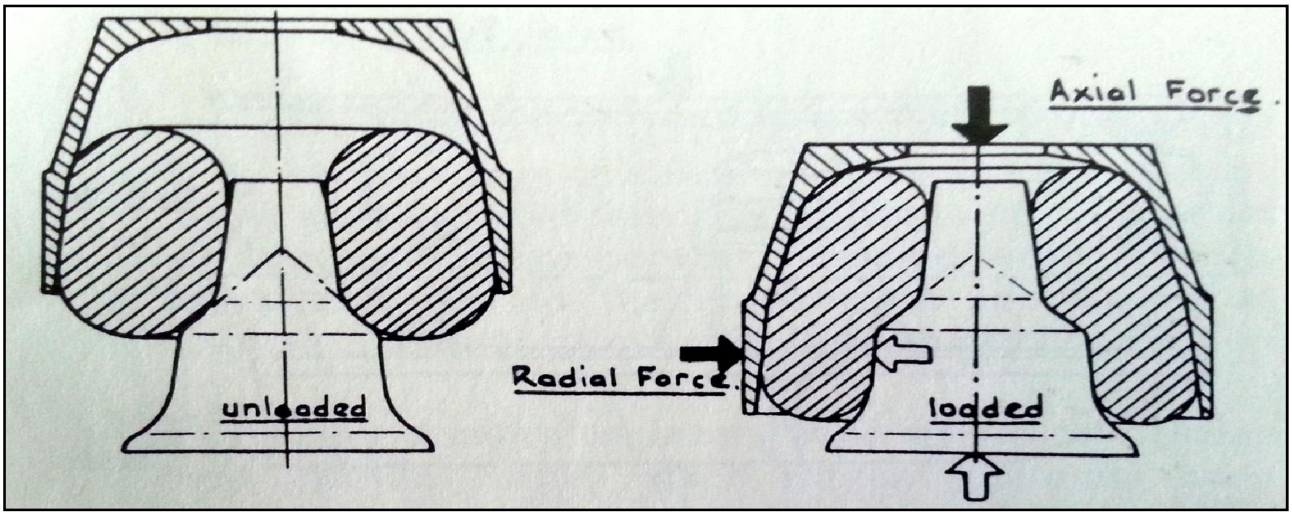


Fig.(3.33) Clouth rubber spring.

Space considerations, or particular stiffness values, may dictate the use of an independent means of locating the axle longitudinally and providing the spring rate in that direction. Such an arrangement, shown in Fig.(3.34), is used on the latest BR suburban stock. The vertical and lateral stiff nesses are provided by a chevron rubber unit positioned directly over the axlebox bearing; the axlebox is provided with a trailing arm, the pivot of which has a rubber bush providing the axle yaw rate.

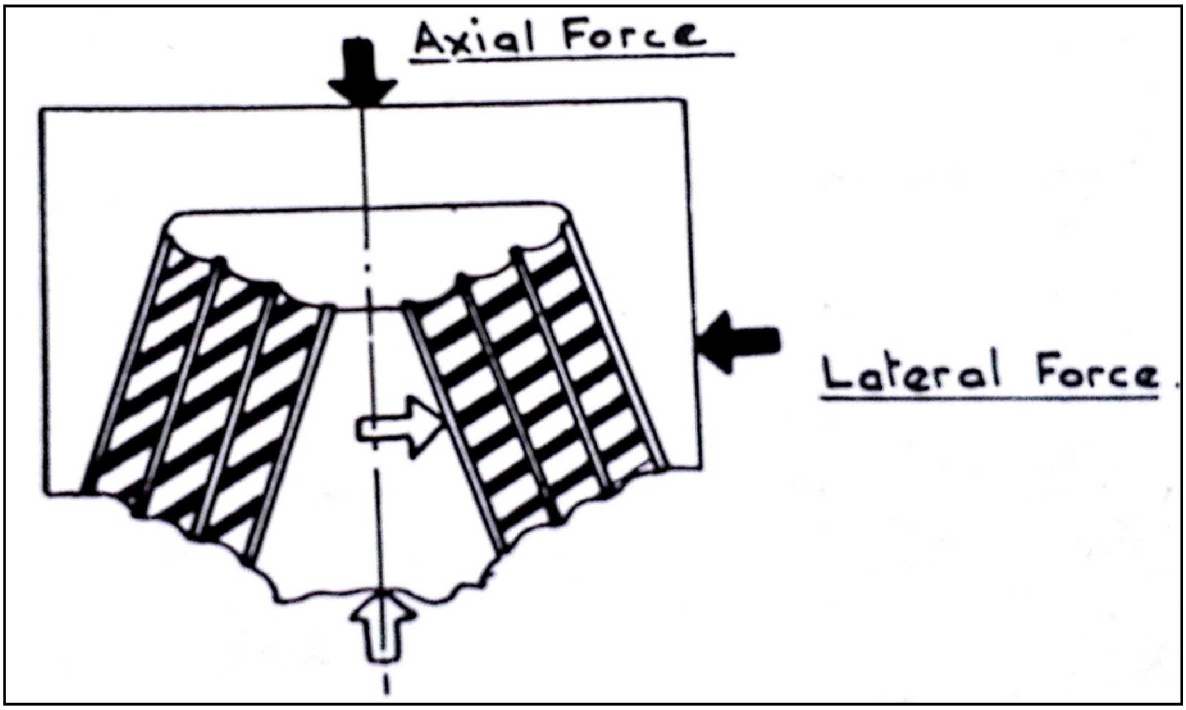


Fig.(3.34) Rubber chevron spring.

Although the rubber elements have inherent hysteresis, this is usually insufficient for the vertical displacements and therefore external damping is normally applied.

A further advantage of rubber suspension units, becoming more important in modern vehicle designs, is noise reduction by elimination of a principal transmission path.

In any rubber suspension the designer must make due allowance for creep effects due to temperature and the constant static load.

Air spring

Air springs are another increasingly popular suspension means. A number of forms of air spring are possible; in Britain, two forms are now in common use as secondary suspensions.

The first form, of a convoluted design, is used in conjunction with a bolster; in other words, it provides only the vertical deflection, the lateral movement and stiffness being by use of swing links and rotational displacement provided by side bearers with a friction interface of stainless steel on PTFE pads. Fig.(3.35) shows a convoluted air spring as applied to the latest BR locomotive-hauled passenger coaches as well as the trailer vehicle in the High Speed trains.

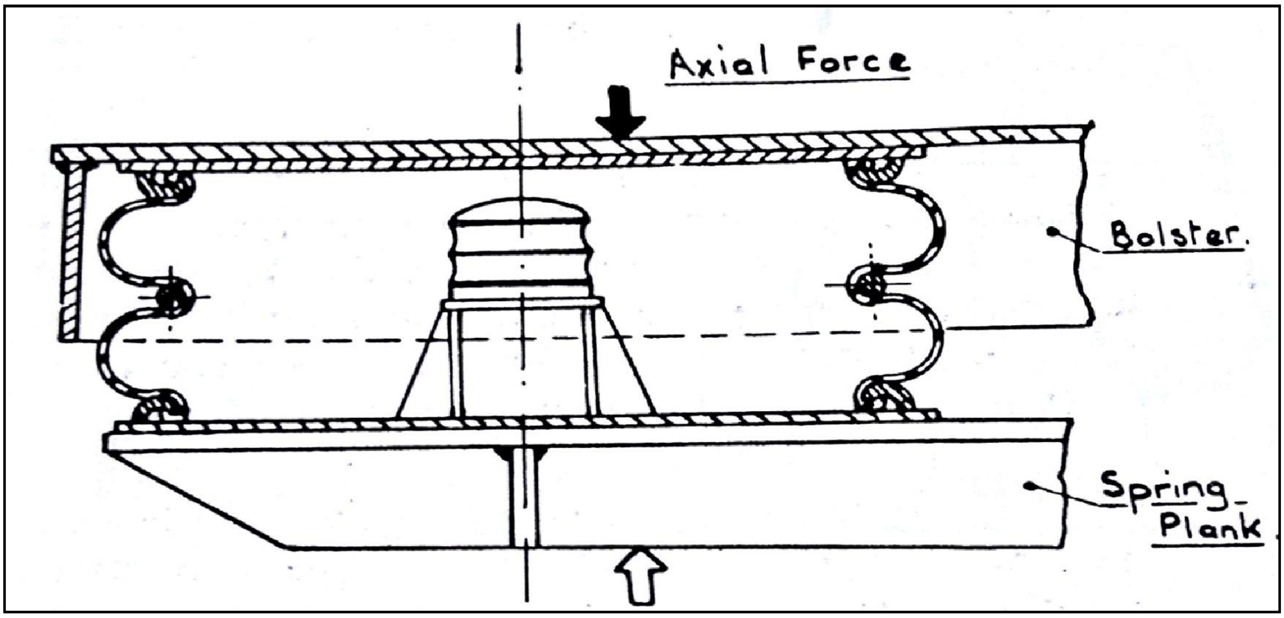


Fig.(3.35) Air spring – convoluted type.

The second form is the diaphragm type; this form of spring can be designed to act as a ‘flexicoil’ spring, i.e. providing all the relative displacements between bogie and body and thus eliminating the bolster. This arrangement, now the standard form of suspension on new design of BR suburban stock, is shown in Fig.(3.36).

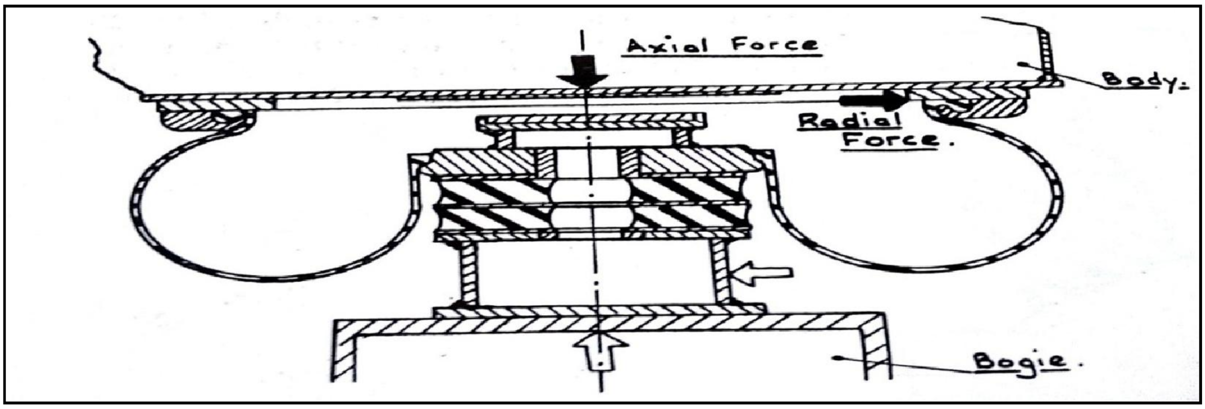


Fig.(3.36) Air spring – diaphragm type.

The principal advantage of air secondary suspensions is an ability to accommodate a very wide range of rate to overload conditions (important in any high density rapid transit vehicle) at an almost constant natural frequency, yet at the same time, because of control of spring height by a leveling valve, there is little change of vehicle height throughout the whole load range. Thus, a soft suspension can be achieved without large static deflections.

Hysteresis in the spring rates of the air suspension is quite low, necessitating the use of additional damping. Early applications of air suspensions therefore incorporated hydraulic dampers; however, the provision of an auxiliary air reservoir to soften the air spring axial rate gives a means whereby a suitable choke in the connection between the spring and reservoir can be made to give a satisfactory damping characteristic for the spring. Following developments, both forms of air suspension now operate quite successfully using only the air orifice for vertical damping.

Further, as with rubber springs, the air spring eliminates a principal noise path to the vehicle body.

3.4.6.2 Damper

A damper is any device which dissipates energy from a vibrating system. The damper generates a force, the damping force, which resists the motion and dissipates and depletes the mechanical energy of the vibrating system.

Friction resistance, which occurs naturally in most suspensions, is the most common form of damping; where this is too low, or undesirable, then additional damping by a device specifically for this duty must be employed.

The most usual form of friction damper is that in which two or more flat or curved surfaces are clamped together, sufficiently tightly to resist relative movement but not prevent it, across the spring system. The damping force across the suspension is the product of the coefficient of friction between the surfaces, the clamping force and the number of contact face. To obtain a consistent force, it is usual to use springs to provide the force, as this permit the setting of a known load and take up of wear. The damping force obtained from friction dampers is substantially a constant and is independent of relative velocity or displacement; it is therefore optimum for only one particular set of operating conditions.

Although used widely, on most modern passenger vehicle suspension designs the friction damper is giving way to viscous dampers — the most usual being the hydraulic damper. With this form of damper, the damping force opposing the oscillations may be made to be proportional to the velocity of the latter. Suitable valving can be incorporated to cut off the rising damping force at a predetermined value to prevent transmission of very high forces under extreme suspension accelerations. The near proportionality of damping force and velocity means it is suitable over a whole range of suspension displacements and that at very low velocities there is little resistance, eliminating the stiction effect of the friction clamper.

Fig.(3.37&3.38)show the primary suspension of a modern high speed coach fitted (a) with friction dampers and (b) with hydraulic dampers.

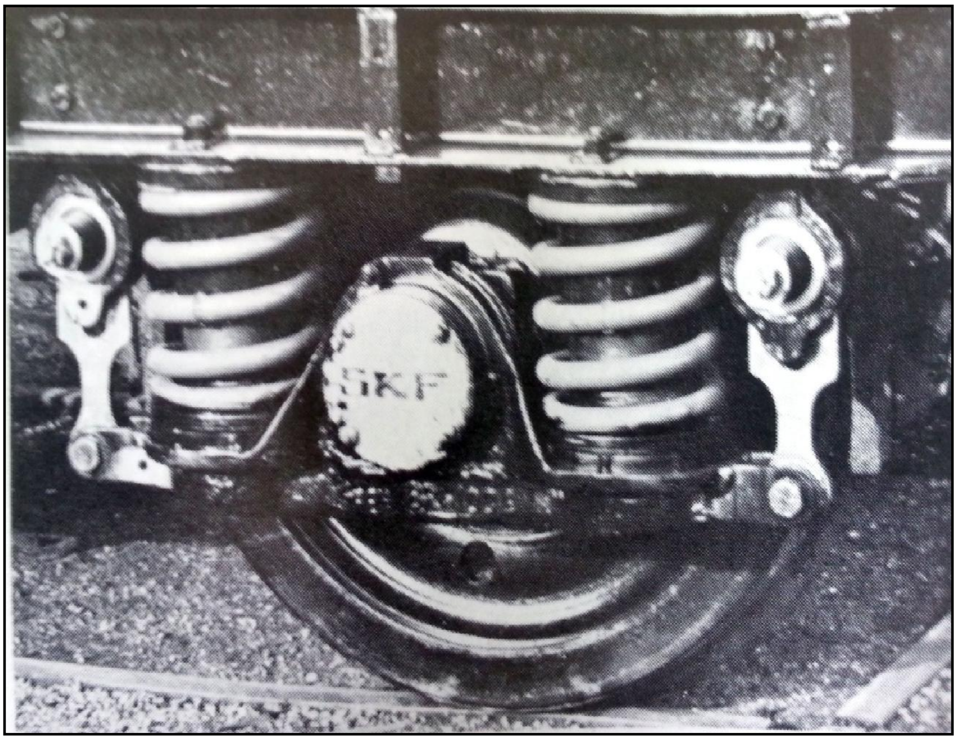


Fig.(3.37) Bogie primary suspension – friction dampers.

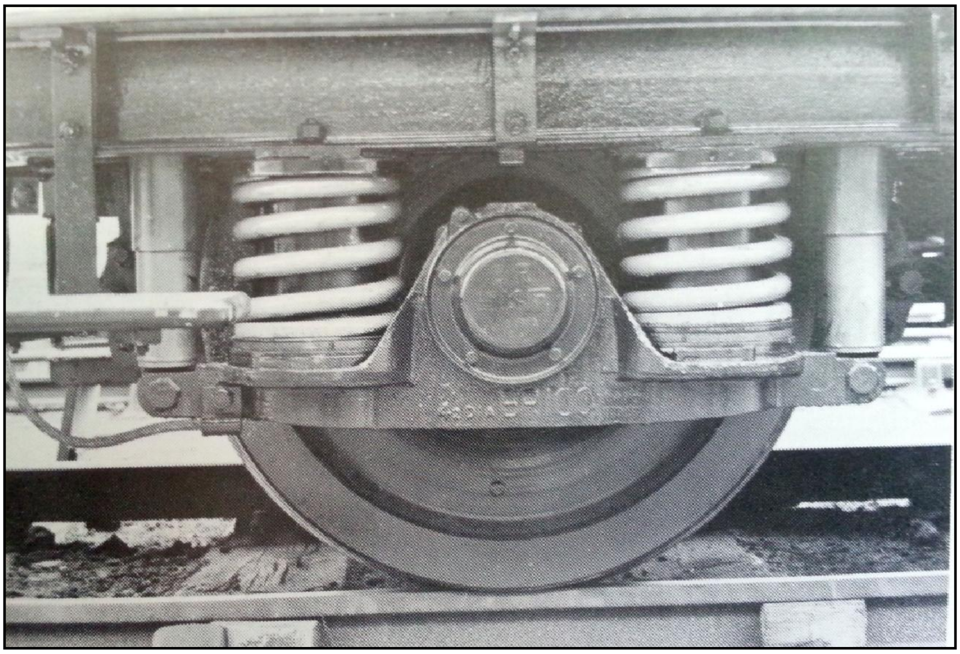


Fig.(3.38) Bogie primary suspension−hydraulic dampers.

3.4.6.3 Axlebox guidance

Whilst European practice has generally turned away from the conventional horn guide type of wheelset guidance it has remained almost standard in America and most other areas of the world. Links and guide *posts* with suitably designed rubber bushes or pads are now commonly used in Europe to achieve controlled wheelset freedom. Fig.(3.39) shows a recent innovation on this theme currently being developed by BR; in this application the rubber rolling ring provides thewheelset yaw and lateral stiffnesses, vertically the rubber contributes a little to the vertical suspension stiffness. Such systems of axlebox guidance are regarded as essential to ensure stability of coach and multiple unit bogies etc. at- speeds above 130 km/h but experience has shown that locomotives can operate in many cases at 160 km/h with conventional hornguides.

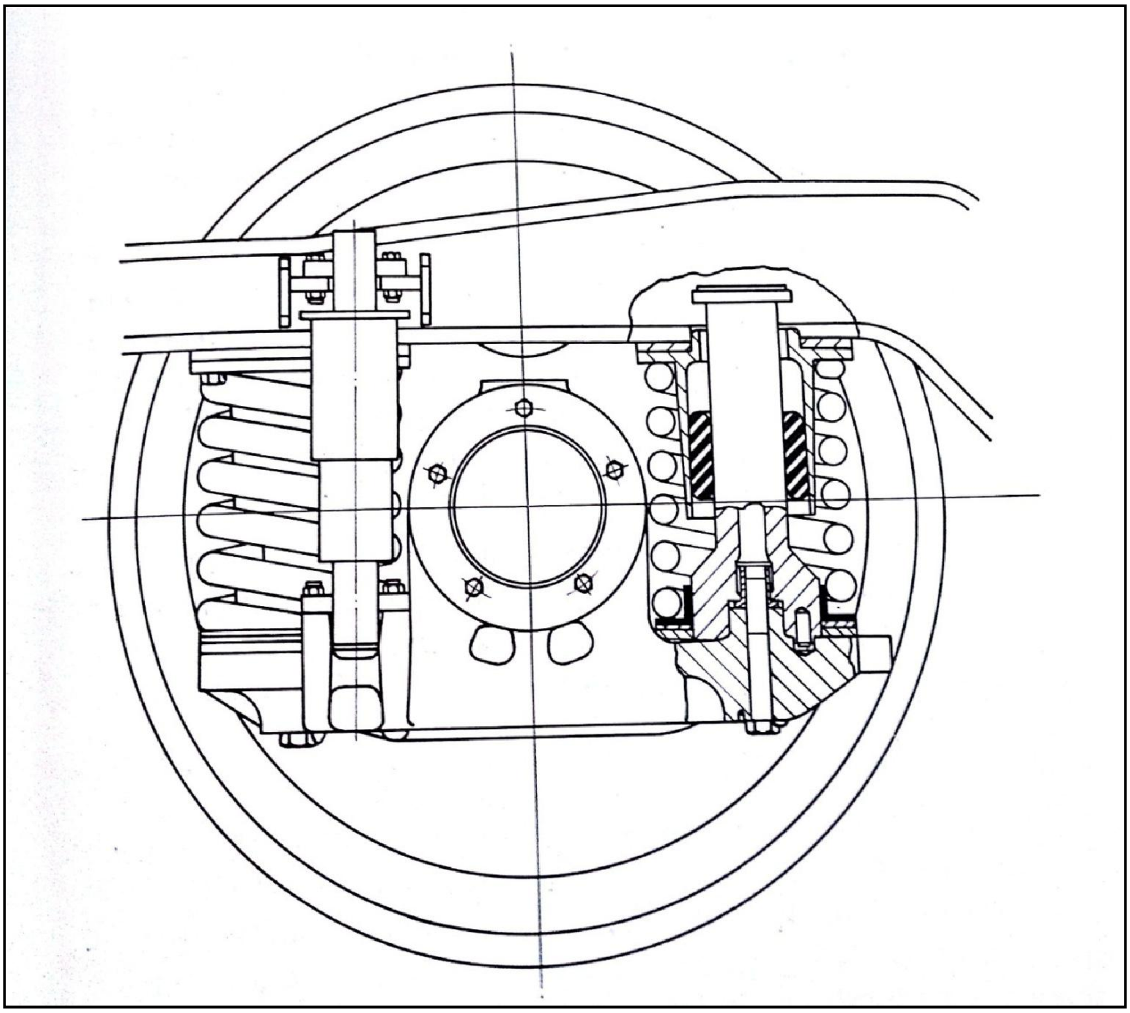


Fig.(3.39) Rolling ring axle guide.

3.4.6.4 Wheels and axles

Roller bearings are now almost universally used for traction motors and axleboxes. The longer life achieved by these bearings has concentrated engineers’ minds on achieving longer axle life. Care is now taken to ensure a good stress distribution in axles, particularly at the wheelseat.

With the increased speeds of operation many operators are moving over to monoblock wheels. Excessive tread braking on tyred wheels can lead to loose tyres and potentially hazardous derailments.

3.4.6.5 Tyre profiles

As indicated earlier tyre profiles have a profound effect on the vehicles' performance in the lateral plane. Early engineers soon found that a slight coning of the wheel tread and the mutual inclination of the rails gave stable running on straight track with reduced flange/rail contact. With moderate operational speeds stability could be maintained even with worn wheels provided the axles were guided to remain essentially parallel to each other. The action of wear on the wheel tread normally exaggerates the tyre profile shape adjacent to the flange, giving rise to an increase in the effective conicity.

As conicity increases the Stability margins decrease and it necessary to re-profile the wheels at some stage of tread wear to avoid the onset of instability. The gross difference in shape between new and worn profiles however, leads to the requirement to remove an excessive amount of material from the tread.

For many decades engineers have striven to develop new profile forms which largely maintain their essential geometry as wear takes place. Unfortunately, such profiles usually give rise to high effective conicities even when new and are unsuitable for many existing vehicles, reducing the speed at which instability occurs to an unacceptable level. Such profiles, however, do generate the largest wheel differences as the wheelset is moved laterally relative to the rails and are, therefore, most suitable, for vehicles which have to negotiate many curves, Fig.(3.40) shows two typical BR profiles. The P1 profile has been the standard for many years but its low concicity when new (0.05 — see Fig.(3.41)) is not maintained as the wheel tread wears. P8 is the BR profile which is superseding P1 on all modern vehicles specially designed to maintain stability on high conicity. Fig.(3.41) shows that effective conicity when new is 0.18 and this increases only moderately with wear, to about 0.25 Even the BR HSTs which operate at 200 km/h are fitted with this P8 profile.

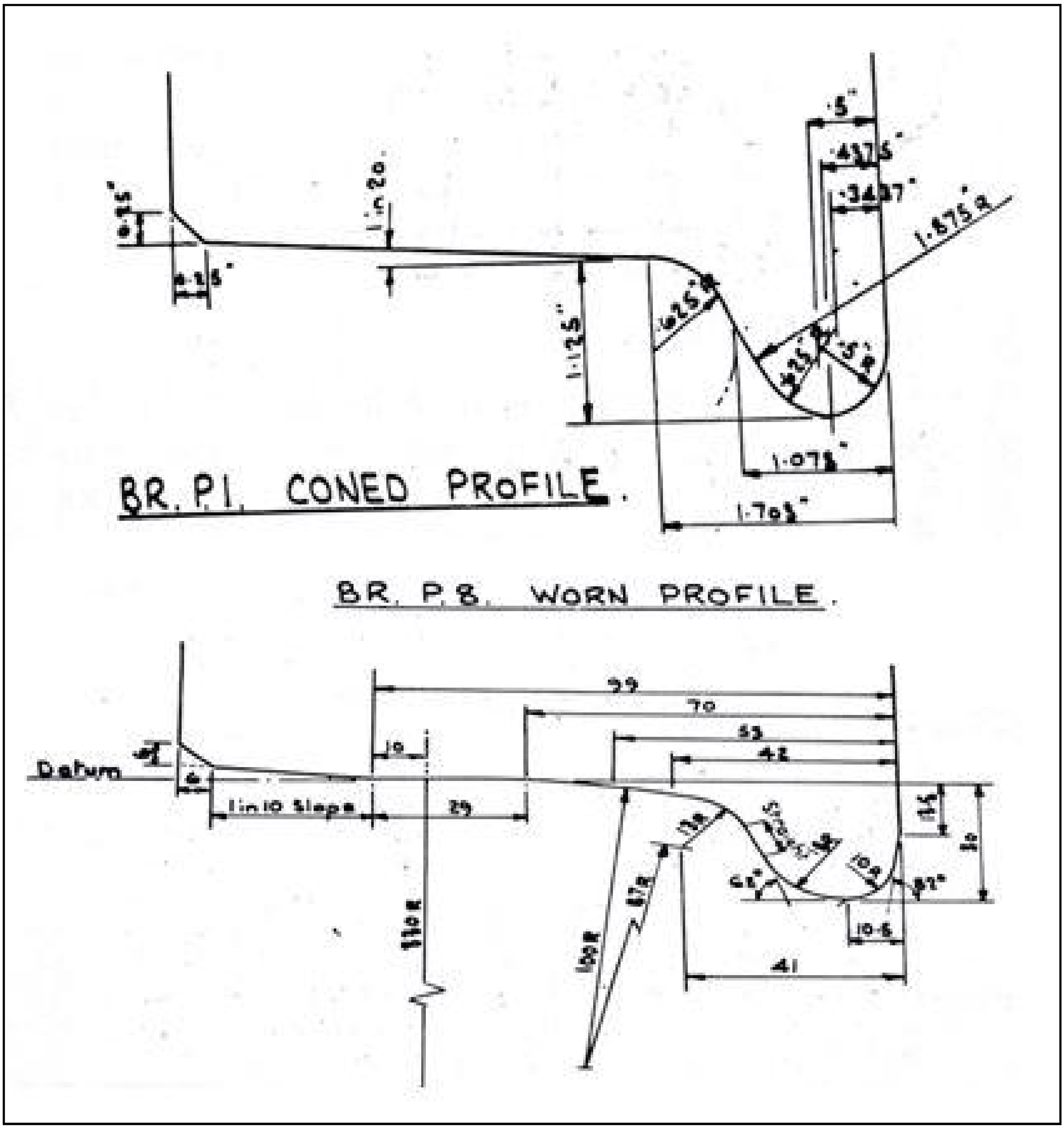


Fig.(3.40) Tyre profiles – coned and worn.

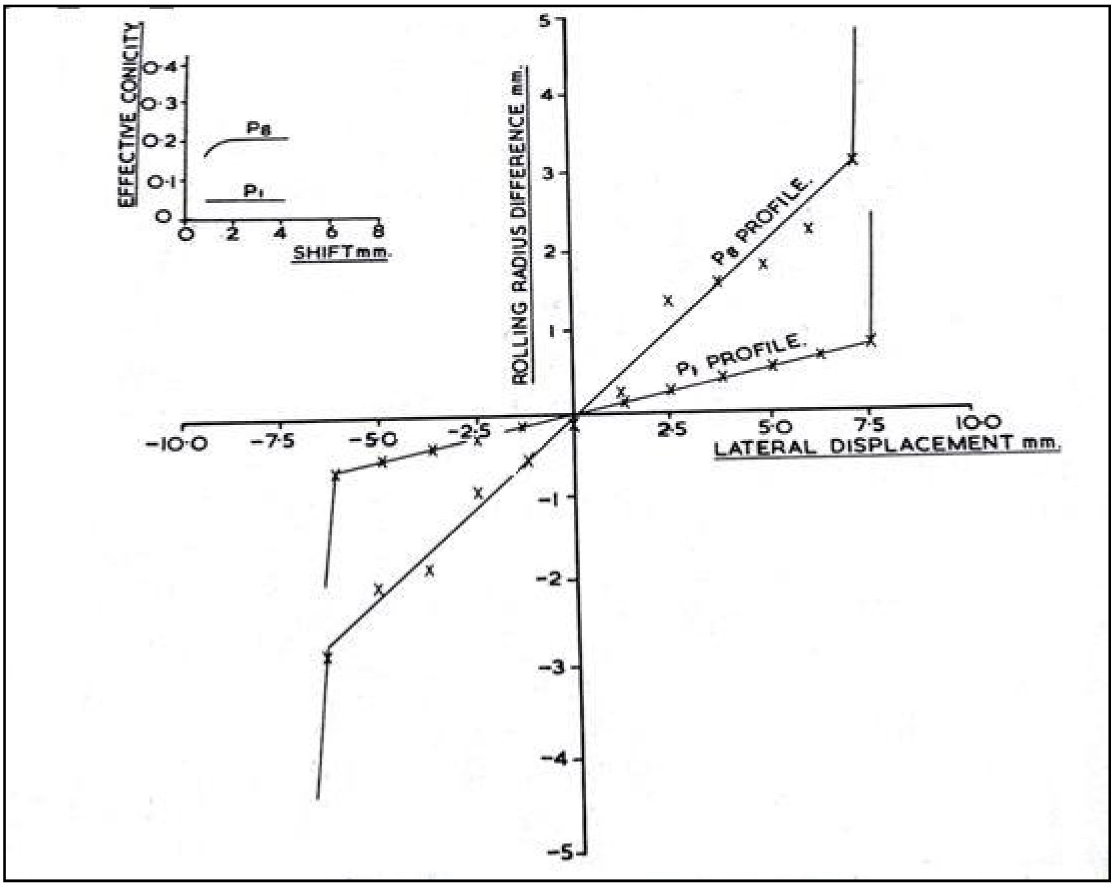


Fig. (3.41) Tyre profiles – P1 and P8.

3.4.6.6 Bogie frames

In recent times bogie frame construction has reduced to two basic forms castings single or three pieces or fabricated designs, the riveted form of construction having largely been replaced by the latter. Much research in welding processes and the fatigue of welded sections has resulted in modern trouble-free designs.

Cast frames tend to be a little heavier and costlier than their fabricated counterparts but in many instances the differences are small and the choice of a particular builder may depend on the local circumstances. BR, with a large investment in welding equipment and machines to prepare plate have favored the fabricated design. Many recent designs, however, have incorporated castings in areas of the frame which would be difficult to fabricate. It seems we are now in the era of the hybrid frame.

3.4.7 Types of bogies

To conclude the study two recent designs of bogies have been selected to illustrate how engineers have used the current technology. All the bogies are BR designs and they are representative of current thinking for multiple units and freight vehicles.

3.4.7.1 BXl bogie for electric multiple unit Fig.(3.42)

This bogie dates from about 1974 it and its many variants are now the standard bogie for all inner and outer suburban units built for BR. A narrow gauge version has been supplied to Taiwan.

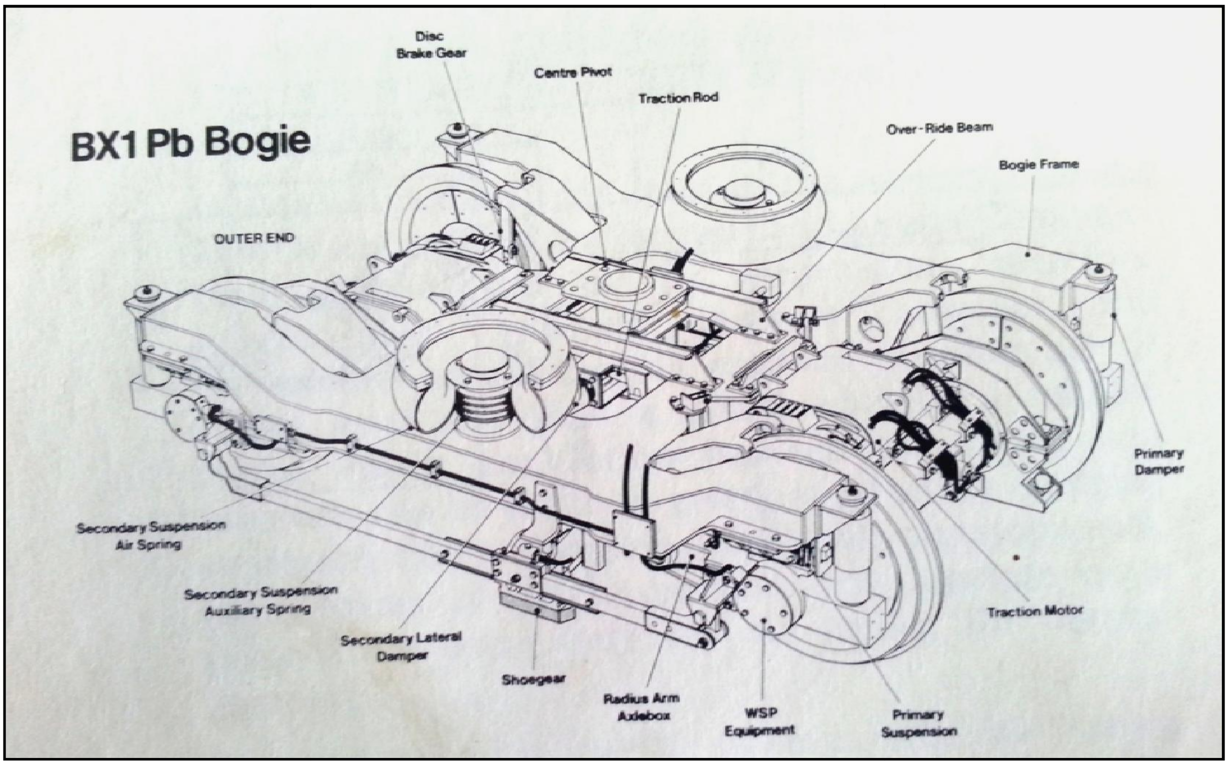


Fig.(3.42) BXI bogie for electric multiple unit.

The bogie frame is a basic "H" form without headstocks fabricated from mild steel plate into box sections. The frame incorporates a number of small steel castings welded into the basic box sections in areas of complex geometry which would be difficult to fabricate.

The primary suspension consists of rubber springs arranged in Chevron formation above a radial arm type of axle box. Wheelset guidance is controlled by the rubber bushed axle box/frame connection. Hydraulic damping is provided.

The secondary suspension consists of air springs which have freedom in all three planes. The suspension is very soft in both vertical and lateral planes with basic rigid body modes of less than 1 Hz. Body height is controlled by a pair of leveling valves arranged at each side of the bogie. Damping in the lateral plane is by hydraulic dampers fitted between the bogie side frame and traction centre. In the vertical plane air orifice damping is used; this is simply achieved by fitting a suitable sized choke in the pipe connection between the air springs and their auxiliary reservoirs.

Disc brakes are used to give 9% gretardation in intensive inner-suburban operation. Each axle is equipped with wheel-slide protection.

The motors are axle hung nose-suspended.

The bogie is built in motor and trailer forms using the same frame. Versions are in service incorporating shoe-gear for third rail electrified systems and trip- cock gear for underground tube style operations.

3.4.7.3. Three piece cross-braced freight bogie Fig.(3.43 and 3.44)

This bogie, recently developed by BR and supplied to a number of customers in narrow gauge form follows the basic three piece bogie form but in this case the frames are specially designed Steel fabrications. The “secondary suspension” is conventional helical coil springs with load-sensitive friction damping. Each side frame is mounted on the wheelsets via rubber pads which are designed to give low lateral and yaw wheelsets tiffness to promote passive steering in curves. Stability of the bogies is maintained by the cross- bracing between the wheelsets see Fig.(3.44).

This type of bogie is particularly useful for freight vehicles which are required to operate on routes with a significant population of small radius curves, the improved curving performance giving significant reductions in flange wear and rail side cutting compared to conventional bogies.

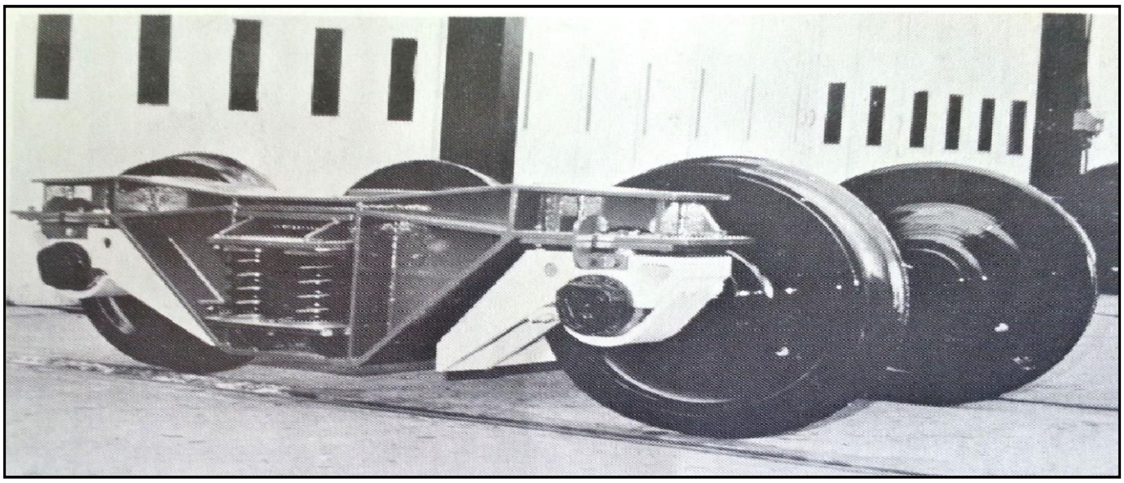


Fig.(3.43) Three piece cross- braced freight bogie

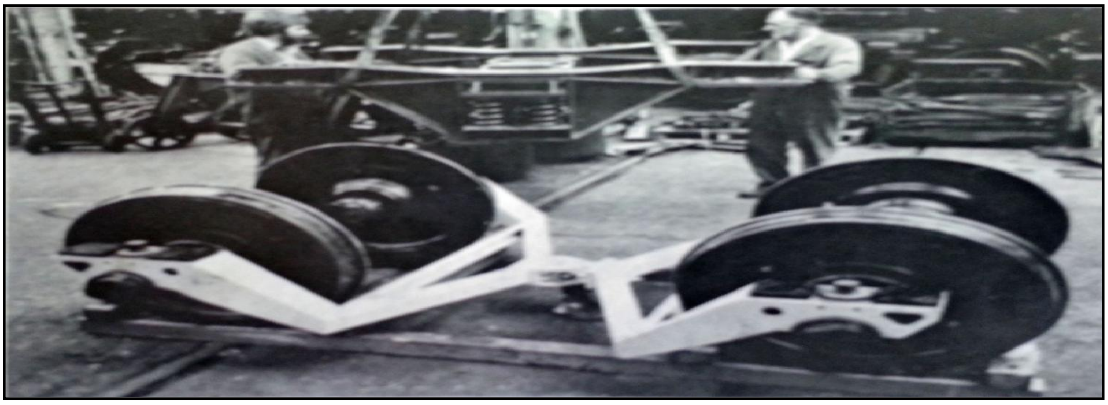


Fig.(3.44) Three piece cross-braced freight bogie showing wheelsets, cross-bracing and axle box shear pads.

CHAPTER FOUR:

RESEARCH WORK

UP-GRADING OF SRC TRACK FOR HIGHER SPEEDS, LOADS, STABILITY & SAFETY

4.1 Introduction

The general objectives of the project involve mechanical, electrical & electronic engineering applications in motive power (locomotive), rolling stock (rail cars & freight wagons) and in plants & machinery serving these equipment.

Also the railway (civil engineering) has a lot to do with the study of the motive power and the rolling stock because of the interaction between wheel/rail which involves design, construction, maintenance & safety for both the track and units running on it.

The components of the permanent way; rails, fastenings, base plates, fishplates, sleepers, etc. are all of mechanical engineering design & nature.

The train track dynamics studies which helped in redesigning for enhancement of both motive & rolling stock and the track are almost done exclusively by mechanical engineering dynamicists.

The above subjects have been displayed, studied &criticized in chapters two and three.

The specific objective of this project is a PRELIMINARY TECHNICAL FEASIBILITY STUDY for some of the existing main line routes of SRC to be up-graded to a higher speed of 120 km/h& a higher axle load of 20 tons.

This feasibility study, to be presented in this chapter (4), shall touch on the relationship between the three components of running trains i.e. motive power (locomotives), rolling stock & the track when in operation. The interactions and the train track dynamics computer programmes are used for simulation of all design, construction, maintenance, operation, etc problems to have quantitative results which help in the enhancement of products of the railway industry.

This preliminary technical study CAN NOT be taken as a substitute for a professional technical and financial feasibility study to be conducted by professional personnel.

This chapter shall include studies on:-

- Up-grading of some SRC main line routes for higher speeds.
- Up-grading the same lines for a higher axle load.
- Stability & safety.

Every railway must have a philosophy to achieve its main goals through successful business. Some pillars of a railway philosophy are:-

To start with, track selection may come on top of priorities. Who sets the objectives for selection?

There is no hesitation in identifying general management as the body responsible for this decision. Only they can identify the traffic to be moved in terms of gross tonnage per annum, the speed at which it is to be moved and the maximum axle loading. In addition they must indicate the anticipated life of the project.

With these four pieces of information it is possible for the engineer to undertake his part of the task, namely providing a minimum cost track to meet the business needs.

We emphasize most forcibly that the objective must come from the general management in clear unambiguous terms. Without a clearly defined objective the engineer is likely to provide a track having an overlong life and capable of taking any traffic thrown at it, possibly the only realistic consideration would be speed limitation.

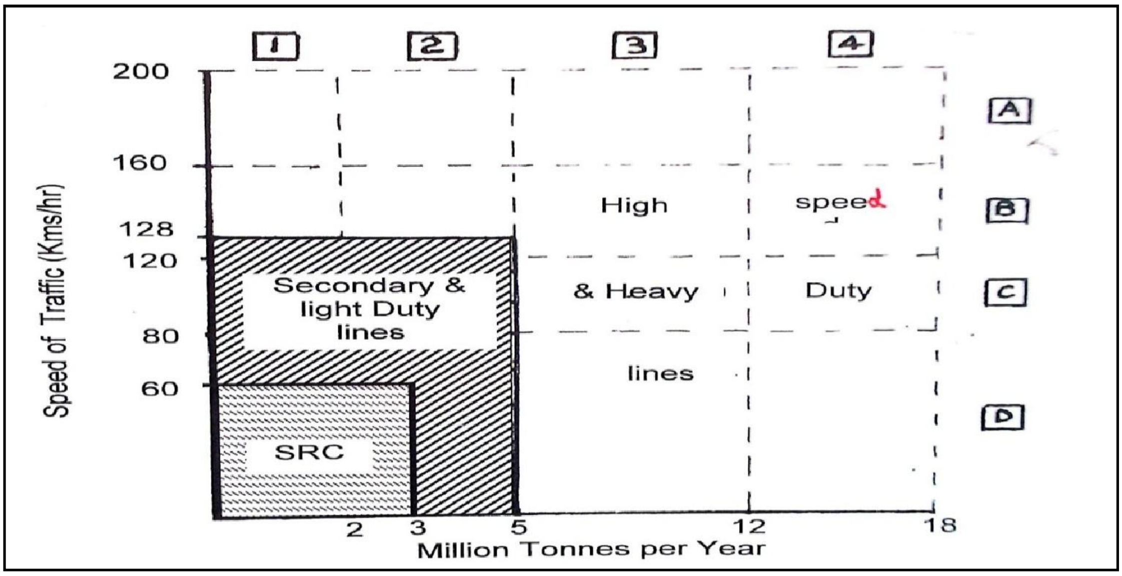
In such a situation the probability is that a too expensive track would be the end result.

Motive power and rolling stock dictate the design of the track or vice versa. Highly qualified and experienced railway engineers are the personnel entrusted the job of preparing the specifications and deciding on the selection of motive power and rolling stock suitable based on the philosophy of the particular railway.

Since now SRC is an existing entity with its rail routes, motive power and other facilities, it is expected that its development and upgrading shall be based on scientific studies rather than random decisions.

If we look at fig. (4.1) which is a method of classifying the status of lines, we see that the system shown in figure is based on speed & tonnage.

For British Railway (BR) secondary & light duty lines are taken to be those with route speed of 128 km/h or less & annual tonnage of 5 million tonnes or under.[10]



Fig(4.1) Classification of railway track.

If we look to the classification of SRC status of lines in Fig. (4.1) (60 km/h speed, three (3) million tonnes per year) we can see that it is a very modest railway & it needs to be put in the intensive care then to be up-graded in the near future to the capacity of the secondary light duty lines of BR in the 60s of the 20^{th} century at least.

4.2 The track[11]

4.2.1 Introduction

General information on track, the track structure, individual track components, rails, rail fastenings, the sleepers, ballast, track formation, the subsoil, together with drainage systems shall be overviewed.

1. General information

The track consists not only of individual componentsviewed separately, but the "railway wheel-track" system as a whole.

The track must

• guide vehicles without risk of derailment,

• take up vertical and horizontal vehicle forces,

• off-load these forces via the track grid and ballast bed into the subsoil,

• ensure high passenger comfort and

• high availability for train traffic

The railway wheel transmits vertical and horizontal forces onto the track. Furthermore, the long weldedrailway track is subject to the influence of longitudinal forces arising because of changes in temperature.

The track is stressed by quasi-static (low-frequency) and dynamic force components of higher frequency. Fig. (4.2) schematically represents theWheel-Track system.

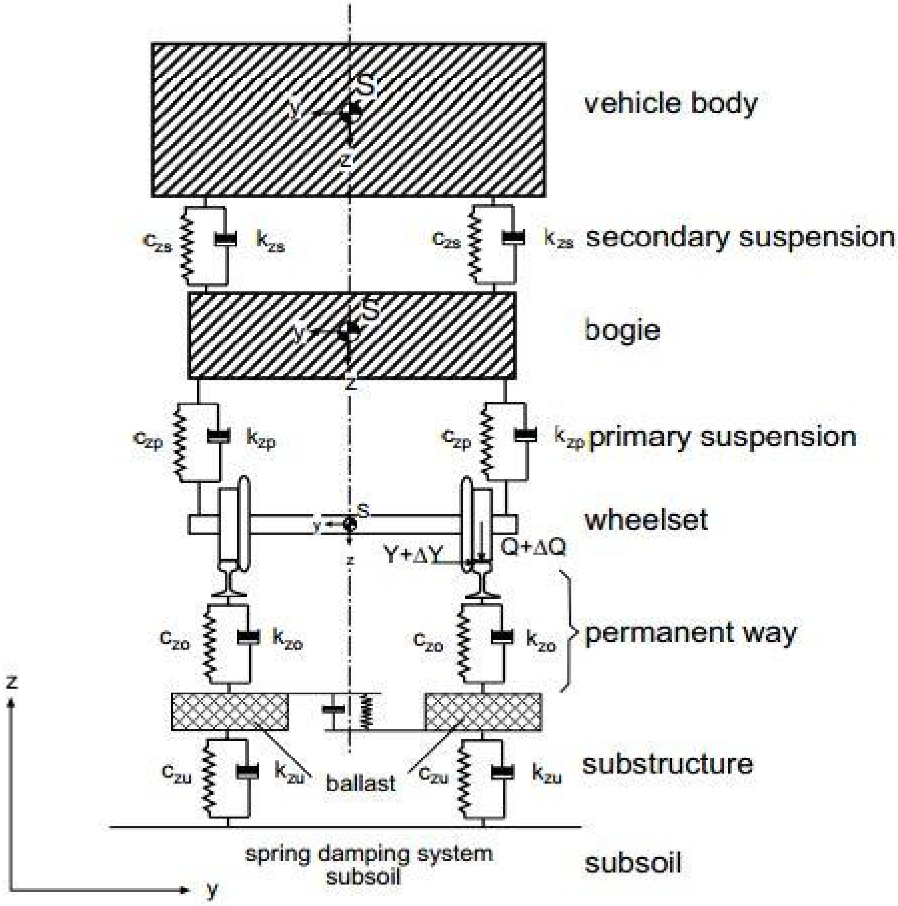


Figure (4.2) Schematic representation of the Wheel- Track system.

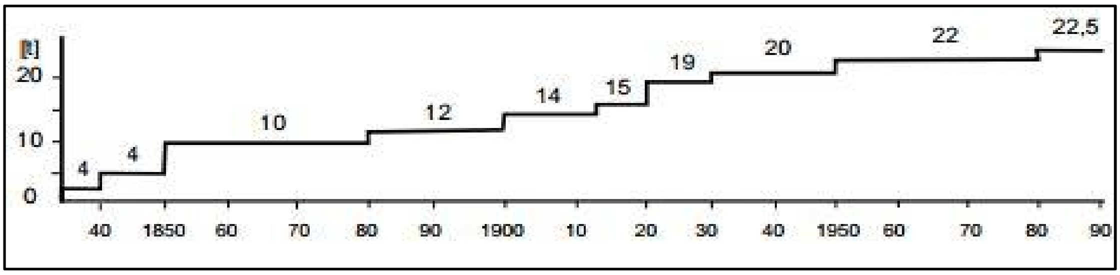
The individual parts of this system are linked by components exerting elastic and damping effects.

The elastic and damping elements between vehicle body and bogie, as well as between bogie andwheelset, are very well known and their behaviour can be very well expressed mathematically.

The track itself with its elasto-plastic properties cannot be expressed by an exact analysis becauseof the inhomogeneous behaviour of the ballast bed, the protective layer of the formation and the subsoil.Empiric parameters and connections found out by experiments are used for this purpose.

The strength of these forces is a function of the axle load, of changes in wheel loads when drivingon curves or in case of unequal loading, of braking and starting, and the rolling of ovalized unbalancedwheels on a defective track.

The track grid has to distribute these forces in such a way, that the maximum admissible values forballast compression below the sleeper and the admissible compressive strain on the soil will not beexceededFig. (4.3) shows the increase of wheelset loads and speeds in the course of railway history. It isremarkable how the wheelset loads for goods wagons have steadily risen to today’s value of 22.5tons – in the future it will be 25 tons. The speed of passenger trains has also increased: 250 to 300km/h on new tracks. Trains in general run at a higher speed nowadays, even on the rest of the railwaynetwork.



Development of weelset loads for passenger and goods trains

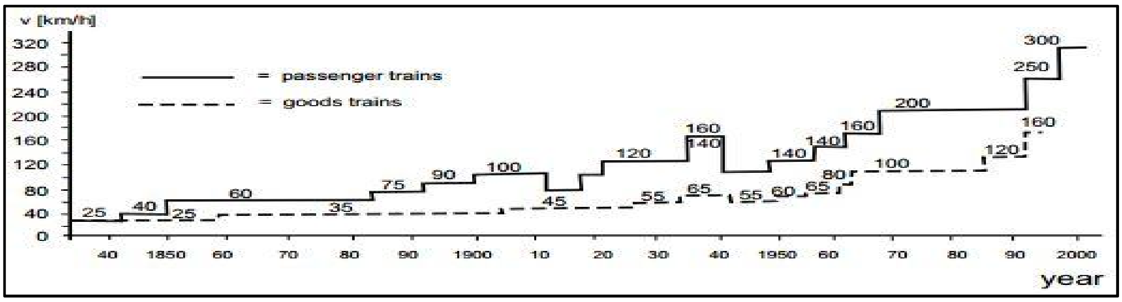


Fig (4.3) chronological development of wheelset loads and train speeds, for passenger and goods trains.

The theoretical comments and practical experience stated in the following are intended to explainhow to fulfil these higher requirements which, undoubtedly, will rise even further in the future.

Permanent way for high-density lines, which, according to modern knowledge, requires only a littlemaintenance, consists of the following elements:

• Heavy-profile rail UIC60,

• Hard-wearing rails in curves (head-hardened or high-alloy),

• Concrete sleepers of optimized quality for track and switches (soled sleepers, broad sleepers,frame sleepers, ladder sleeper track, etc.),

• Torsion-resistant and elastic rail fastenings (optimization of elasticity and damping is necessary),

• Permanently stable ballast bed and

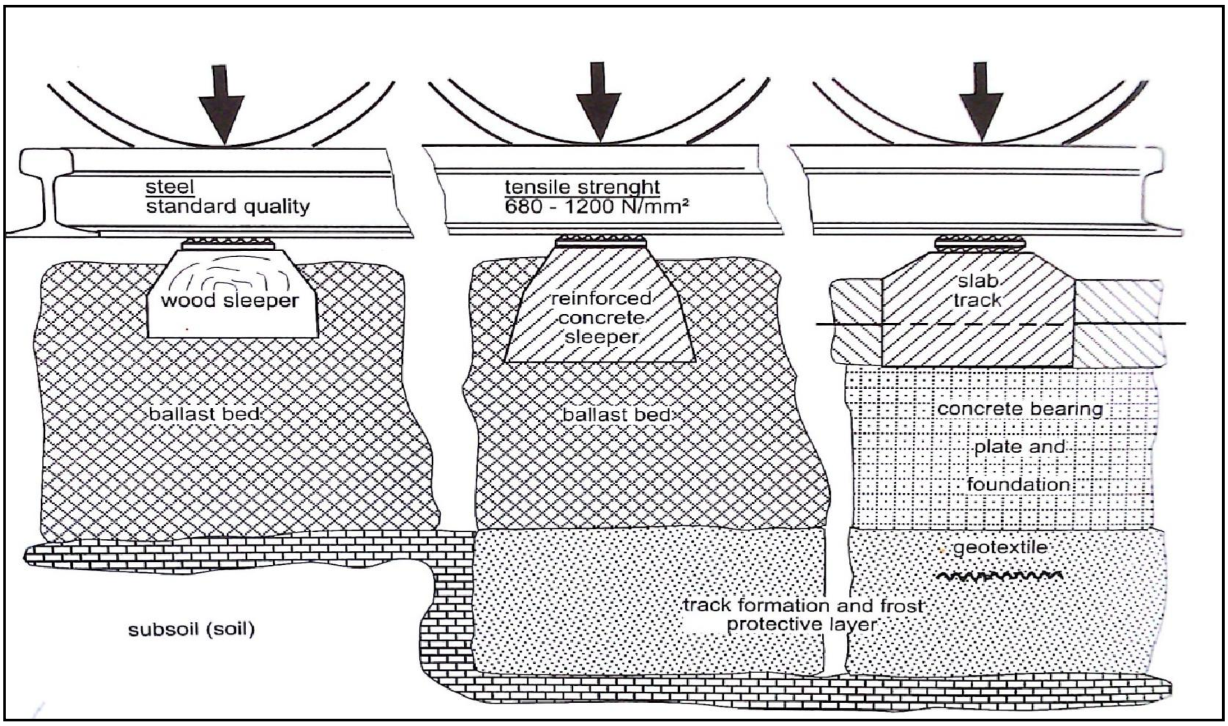
• Permanently stable, frost resistant track formation (by the insertion of protective layers and geotextiles).

2. The track structure

Fig.(4.4) gives an overview of the track structure. It shows that the entire system consists not only of ballast structures with sleepers "swimming" in them, and of the rails supported on them, but also of the track formation with its protective or improved structure, as well as of the subsoil itself.

Axle force	Track structure	
Vertical, static and dynamic	ballasted bed	slab track with read mixed
Horizontal-dynamic Vibration	transversal sleepers	concrete panels

	frame sleepers	with side-mixed concrete panels
	broad sleepers	with asphalt-substructure.



Fig(4.4) schematic structure of the main track forms

The following describes the individual components, as well as, in particular, their interaction. Only the high quality of each component and optimum adjustment of their properties to each other result in a durable low-maintenance track.

Conclusion on track maintenance

The introduction of 225 kNwheelset loads for goods trains driving at speeds of up to 120 km/h has considerably more influence on the track behaviour than the increase of maximum driving speeds of passenger trains up to 200 Km/h or 250 km/h, respectively, this is valid because the increased wheelset loads act with the fourth power and therefore exert the dominant influence.

Form these various considerations the general condition required is to make the mechanical properties of ballasted track as uniform as possible. The requirement of evenness and homogeneity has to be considered at least as important as the requirement of observation of certain minimum tensile properties or deformation resistance. It is necessary to bear in mind that changes in track geometry, apart from operational circumstances, also substantially depend on the quality of initial installation.

A well maintained track is the most economical solution for railway and vehicles.

The more rigid the track grid is the less will be the stress for the underlying layers. Rigidity of the individual layers should decrease from top to bottom to avoid incompatible concentrations of tension in softer layers. The rigidity of the ballast permanent way should be as uniform as possible; unequal rigidities lead to longitudinal unevenness.

Maintenance measures

The following illustration gives an overview of the individual track components and possible correction and measurement works that may be carried out for maintenance purposes.

3. The rails

a) Rail requirements

The rail is running surface, carrier and guiding element at the same time. It is subject to equal static and dynamic stress. In heavy haul traffic, axle loads up to 35 t are applied. Nowadays in regular high-speed traffic speeds of up to 300 km/h are reached. Depending on the topography rails are laid with radii as low as 300 m, therefore, they are subject to the very high lateral forces exerted by the wheel flange striking against the gauge corner of the outer rail. To be able to withstand these manifold and high forces, the rails must meet the following requirement:

• High resistance to wear,

• High resistance to compression,
• High resistance to fatigue,
• High yield strength, tensile strength and hardness,
• High resistance to brittle fracture,
• Good weldability,
• High degree of purity,
• Good surface quality,
• Evenness and observance of profile and
•low residual stress after manufacturing.

b) Rail defects

The so-called catalogue of rail defects with definitions of rail defects is edited by the UIC. Rail defects are cracks, fractures and damage to rails and rail connecting areas.

Defects are catalogued according to their position, appearance and cause, The UIC leaflet 712 "Rail defects" deals with this topic. A system of 4 figures is used for cataloguing:

1st figure: position of the defect in the rail
2nd figure: position of the defect in profile
3rd figure: direction and type or cause of the defect
4th figure: additional features

The current classification contains 55 different types of rail defects.

Table (4.1) indicates the most frequent and typical defects occurring at present in long welded tracks.

Rail defects and incipient cracks are detected by ultrasonic tests. As an ultrasonic wave is reflected by any surface, the damage can be recorded. For this purpose several oscillator crystals with water contact are mounted on the running face. Type and dimension of a fracture may be recognized from the reflected picture of a fracture or cavity.

Table (4.1) most frequent rail defects in long welded tracks.

Numbering	Short designation of the rail defect
111/211	Tacheovale
2222	Shelling
2223	Head Chechs
not contained in the UIC leaflet	Belgospis
227	Squats
301	Indentaions
125/225	Skid marks
2201	Corrugation
2202	Waves
411/421/431	Cross-cracks (welding)

4. The purpose of the rail fastenings

It is the purpose of the rail fastening to maintain the track gauge and to transmit forces acting on and in the rails to the sleepers (cross, longitudinal sleepers, concrete plates etc.).

As the fastening system must be damped in a vertical direction for movements upwards as well as downwards, the properties of the fastening have to be considered when the rail is lifted off under traffic load (lift-off wave). This is of particular importance for concrete sleepers, as the weight of the track grid is significantly higher (2-3 times) for concrete sleepers, as opposed to wooden sleepers. It is important that the fastening system offers sufficient resistance in a vertical direction. Each pair of rail fastening elements has to be able to take over the weight of the concrete sleeper and the respective rail section without excess deformation.

One of the main functions of the fastening system used on concrete sleepers is to electrically insulate the sleeper against the remaining track grid. This is necessary to minimize the loss of signals of the direct-current circuits which occur due to insufficient insulation of the rail against the subsoil.

Electric insulation has, of course, to be guaranteed, also during rainy weather and humid air conditions.

The effective forces

a) **Vertical forces**

Vertical forces acting across the track are partially absorbed by the pads inserted below the rails, and by their appertaining spring elements.

b) **Lateral forces (acting across the track)**

Dynamic forces acting across the track are partially absorbed by the compressed elastic rail pads, the remaining force may be transmitted directly to the sleepers via vertical elastic rail pads.

The lateral forces acting on the track may be very high. The rail fastenings have to transmit them to the sleepers via a firm connection. If this is not possible, the occurring forces will increase to such an extent, that the track will be crushed.

5. The sleepers

At the beginning of railway technology the preferred sleeper material was wood and it continued to be so for the next 50 years. Wood is susceptible to weathering and other external influences. This and the intense steel production in the last quarter of the 19th century finally led to the changeover to steel sleepers. These were used for more than 50 years in many parts of the world including Europe. However, increase axle loads and train speeds soon required heavier sleepers. The first concrete sleepers were introduced at the end of the 19th century. The French gardener Moinier designed the first reinforced concrete sleeper before the turn of the century and applied for a patent. In 1906 the first experiment with a conventionally reinforced concrete sleeper was made in Germany on the line Nurnberg-Bamnerg. During World

War II the concrete sleeper production was extended and in 1939 the production of steel sleepers was discontinued. In general, two different basic types of concrete sleepers where developed:

• The twin-block concrete sleeper where two prestressed concrete blocks are connected via a steel rod or a steel beam, and

• The monoblock concrete sleeper which consists of one prestressed concrete beam.

The introduction of heavy prestressed concrete sleepers was essential to enable the use of long welded tracks.

High Speed traffic with speeds of 200-350 km/h led to the development of different types of ballastless track. At the same time new systems of concrete sleepers, such as the broad sleeper, the frame sleeper and the ladder sleeper were developed and presented.

Nowadays classic steel sleepers are hardly used. The only exception is a special form of the steel sleeper, the so-called Y steel sleeper which is used under special conditions due to its small overall height and width.

A total number of approximately 3 billion sleepers are used all over the world. Only 20% of them are concrete sleepers. About 5% of the sleepers are replaced annually. The total production of concrete sleepers all over the world amounts to about 20 million.

Comparison between wooden and concrete sleepers

The advantages of concrete sleepers compared to wooden sleepers are the following:

•longer life cycle and service life,

•less expensive than hardwood sleepers,

•lower maintenance of the fastenings

•higher resistance to lateral displacement due to higher weight.

On the other hand concrete sleepers have the following disadvantages:

•susceptible to shock and impact.

•difficult handling due to greater weight, and

•maintenance of longitudinal level is somewhat more difficult because of the higher moment of inertia and the lower elasticity.

The purpose of sleeper

The purpose of the sleepers is:

•to establish and maintain track gauge,

•to distribute and transmit forces to the ballast bed, such as:

- •the perpendicular axle loads,
- •the horizontal centrifugal forces, and
- •the longitudinal forces within the rails.

•to hold the rails

- •in height (in the case of arching or settlement),
- •to the sides, against centrifugal and transversal forces *(H*forces),
- •in the longitudinal direction against rail creeping, brake, acceleration and temperature forces

•to secure the track

- •under construction,
- •in the case of a rail breakage, and

• after derailment (in such cases the measures prescribed by the railway authorities, such as temporary rail connection, have to be taken), and

•to dampen rail vibration and

•to reduce the influence of sound and impact waves on the environment.

These manifold tasks can be fulfilled by:

•transversal wooden, steel or reinforced concrete sleepers.

•longitudinal reinforced concrete sleepers, and

•reinforced concrete sleeper plates.

6. Ballast and ballast bed

The track grid is carried in the track bed on a "floating" support. This causes the track geometry to deteriorate under the influence of dynamic forces, however, it has the advantage that this deterioration may be remedied at low cost by fully automated permanent-way machines.

Ballast bed requirements

The ballast bed has to provide the following functions:

•to transmit the sleeper pressure to the subsoil as evenly as possible,

•to have high resistance to longitudinal and lateral sleeper displacement,

•to easily re-establish the track position after it has been changed (tamping and lining),

•to have good air and water permeability to maintain the bearing capacity of the subsoil,

•to ensure the track elasticity in order to minimize dynamic forces, and

•to enable corrective actions in the track (tamping and lining).

These requirements may be met in the following ways through:

• choice of the ballast bed thickness,

• choice of the ballast bed cross section,

•choice of the ballast quality, and

•quality of ballast consolidation.

The following factors may have an adverse influence on these properties:

•operational load: due to the wheels rolling over the track the sleeper is lifted off and rebounds on the ballast, the dynamic forces may overstress the ballast and lead to shelling, grain shifting and abrasion,

• maintenance work - wear by tamping or cleaning,

•material rising from the subsoil, it there are no formation protection layers or if they are defective, and

•ballast contamination by lost cargo (coal, ore sand, etc.), remainders of vegetation and other environmental influences (frost, fluctuation of temperature, moisture, etc.).

The ballast bed cross-section

The ballast– thickness this is the distance between the lower surface of the sleeper and the subsoil– should be as large as possible. The pressure distribution lines should intersect, otherwise the subsoil would be pressed up between the sleepers.

•the required ballast bed thickness depends on:
• the sleeper spacing,
•the steeper width, and
•the angle of friction of the ballast.

The thickness of the ballast bed should be at least 30 cm for axle loads of 220 kN, a sleeper spacing of60cm and a sleeper width of 28 cm. For high speed lines a thickness of 40 cm is advisable.

It isimportant to provide 45 to 50 cm of ballast at the sleeper ends. This guarantees a significant resistance to lateral displacement.

The ballast bed cross section should have the following target dimensions: ballast width at the sleeper ends:
• 0.4 m at v ≤160 km/h,
• 0.5 m at v >160 km/h,
• 0.45 m for B75 sleepers (sleeper length: 2.8 m), and
• ballasting up to the upper edge of the sleeper.

Inclination of the ballast shoulder:
• 1:1.5 designed,
• 1:1.25 implemented, and
• a cross fall of the formation of 1:20

Minimum ballast bed thickness below the lower sleeper edge (measured below the low rail)
• 0.3 m for existing lines, and
• 0.35 m or 0.40 m, respectively, for new lines.

This means that the thickness of the ballast bed increases adequately below the superelevated rail, furthermore, the formation has to be broadened correspondingly.

It is advisable to provide ballast of high compactness and tension. This can be reached by inserting and compacting the ballast in layers. The best way to insert the ballast in layers is with the help of a tamping machine, sleeper end consolidators and adynamic track stabiliser.

7. The track formation

General information on the bearing capacity of the track

High track quality is characterized by long-term track stability requiring low maintenance even at high train density, increasing running speeds and axle loads. The track has to be regarded as a whole, as a "system". Each individual element ofthe system, beginning from rail and sleeper, via plasto-elastic properties of the ballast bed down to the formation protection layer and the subsoil, is a component of inherent track quality. The main task of these components 15 to transmit and divert forces to the subsoil caused by trains.

The decisive element in this connect is the bearing capacity of the track. Insufficient bearing capacity of the earth formation is often caused by:

•lacking or poor drainage facilities,

•restricted water flow directly off the ballast shoulders or the cesses, orformation of hollows in contaminated ballast beds due to insufficient cleaning of the ballast bed shoulders as a result of too narrow an excavation width.

The following circumstances indicate such detects of the substructure:

•recurrent detects of track position.

•wet spots - due to subsoil material rising into the track ballast,

•frost damage causing heaves,

•soggy soil being pressed out to the side, and

•deformation of the earth formation by ballast pockets and troughs in which water collects.

Measurement of the bearing capacity of formation

a) Plate load bearing test

The typical parameter for the dimensioning of formation protective layers as well as for quality checks is the modulus of deformation E_{v2} derived from the plate load bearing test. A detailed description ofthis test is given in the chapter "The subsoil".

b) Dynamic plate load device

This method an impact load IS applied for a short term to the formation to be examined. This load formation to vibrate in damped oscillations. It the settlement and amplitude of the damped oscillation are high, the bearing capacity of the formation is low. More detailed information can be found in the chapter "The subsoil".

Drainage of the formation

It was found out early that water contained in the formation has an adverse effect on the track position. The aim was therefore to reduce the water content of the formation by the following measures:

•removal of growth on the surface.

•cleaning the ballast bed and the establishment of a cross fall.

•cleaning of side ditches and drainage facilities.

•arrangement of lateral drainage facilities with inspection and ventilation shafts.

8. The subsoil

If the tearing capacity IS insufficient (see chapter "The Track Formation" and chapter "Maintenance -Track Formation Upgrading Methods"), thealternations of load caused by the vehicle wheelsets will lead in the course of time to plastic deformation of the subsoil. Non-cohesive as well as cohesive soil is compacted due to the rearrangement of grains. In addition, the volume of the cohesive soil is reduced, as the pore water is pressed out of it, which leads to an enlargement of hollows in the ballast bed and the soil formation.

Types of soils and their parameters

Soil types may be roughly divided as follows:

Coarse- grained soil

Particles can be distinguished with the naked eye:

• boulders larger than 300 mm,

• round or square cobbles 75-300 mm,

• gravel <75 mm and >5 mm, and

• sand<5 mm and >0.06 mm

Coarse-grained soils are distinguished by density, grain size distribution (grading *curve)* and grain form. Sand and gravel, in dry conditions form heaps of large grains lying loosely next to each other. Hence they are called non-cohesive or loose, as opposed to silt and clay. The share of fine grains in coarse-grained soils IS less than 5%. Fine grains are grains smaller than 0.063 mm. Soils with mixed grain size are soils with a share of fine grains between 5 and 40%.

Fine-grained soil

Silt and day belong to the category of fine-grained soils. Fine-grained soils are described⁝ by their plasticity, structure, colour and smell. Plasticity is determined by the content of clay. The share of fine grains with a grain diameter below 0.063 mm in fine-grained soils IS over 40%.

Organic soils

Organic soils are e. g., bog and peat. The organic components of the soil may be of animal or vegetable origin. As they burn at high temperatures, they are characterizedby the loss due to burning V_d,which is the percentage loss in weight.

Soil analyses determine characteristic soil parameters, parameters of carrying capacity, sensitivity to settlement and frost.

Defects of the soil formation

Traffic loads and atmospheric influences (precipitation, erosion and wind) on various types of soils lead to various levels of excessive stress, thus causing deformation and damage to the soil formation. Irregular settlement under the sleepers occurs. The following defects can be observed:

• in uniform soils, such as uniform sand, the sand , as a consequence of soil vibration, creeps through the ballast bed up to the sleeper surface,

• in non-cohesive and slightly cohesive soils the soil formation and the subsoil are loosened bydynamic load, which can lead to the formation of cracks and crack zones,

•in silty soils in wet weather the soil formation quickly becomes undulated and muddy soil is pumped up to the ballast surface under the influence of load alternation, i.e., wet spots occur and the ballast sinks into the ground below the sleepers (ballast pockets). During rainy weather the track position becomes poor immediately and recovers stability relatively quickly when the weather is dry,

•in highly cohesive soils which are compacted under the influence of load alternation hollows are formed under the sleepers and the soil arches between the sleepers and towards the cess. During rainy weather mud is also pumped up to the ballast surface in the area of sleepers. These soil deformations take place very slowly in rainy as well as in dry weather.

• in cohesive (silty, clayey) soils frost heaves occur in winter and thawing damage with loosened zones particularly at the edge of the track formation in spring, and

• in uniform-grain sand damage can occur due to wind erosion, and in highly-cohesive soils shrinkage cracks in the dry season and swelling movements during rainy weather, particularly in hot climate areas.

This damage to the soil formation occurs only after a longer lifetime, if the traffic load is low, but at high traffic load may occur even after a short lifetime.

Reasons for damage to the soil formation

The reasons for damage to the soil formation are:

•poor subsoil, i.e., subsoil consisting of inadequate types of soils.

•high static and dynamic load on the subsoil,

• insufficient compaction of the soil formation and other loss in volume of the subsoil,

• insufficient drainage of the soil formation or the subsoil during rainy weather, flooded track, p high groundwater level,

• mud cracks in the soil are filled with precipitation water, the water is stored,

• use of rails the bearing capacity of which is insufficient for the prevailing operational conditions,

• the sleeper spacing is too wide and sleepers with too small a support surface and too high aweight are used, and

• defective condition or construction of structure.

Soil drainage

Water is never desirable in earthworks. Its negative influence has to be eliminated by drainage measures. It is important that water can flow away without hindrance. The service life of earth structuresdepends to a high degree on the quality of the drainage facilities.

If there is much water, the water content of the soil below the soil formation is increased and, in connection with soil vibrations, the shear strength of the soil is reduced. Silty soils begin to flow even after minor water absorption. This creeping is supported by local soil fracture occurring under load as a consequence of too large a difference between vertical and horizontal tensions. Shearing deformations occur in sliding areas of highly cohesive soils. Within a limited surface range the water content of the soil becomes so high, due to subsequent compaction because of load alternations and due to additional water absorption, that it finally exceeds its liquid limit. The high interstitial pressure which exists when cohesive soils are fully saturated by water prevents an increase of the frictional resistance, i.e., the shear strength at load, which favours deformation of the soil formation.

a) Water in the soil

The following forms of water in the soil can be distinguished.

• surface water from rainfall,

• percolating water: that portion of surface water which seeps into the ground and moves downwards according to the law of gravity to fill the cavities in the ground,

• groundwater: is banked-up percolating water. When the percolating water reaches an impermeable soil layer, it fills the soil pores above this layer entirely,

•capillary water which is drawn up into the cavities above the groundwater level, even against the law of gravity, due to the surface tensions of the soil bodies (capillary effect), and

•contact moisture: water adhering to grains, such as adsorption water which moistens the surface of the grains, but does not cover them.

b) The influence of water on the soil

The finer the soil particles, the more water they are able to absorb. The surface of particles and consequently also the surface tension rapidly increases, when particles get smaller. This is why fine-grained soils bind not only capillary water, but also great amounts of contact moisture and adsorption water. Once fine-grained soils have absorbed water, it is practically impossible to set this water free by structural measures. Pressure procedures (such as trains passing) increase the interstitial pressure, which may lead to a sudden loss in bearing capacity (a similar effect is well known to everybody who stands on the shoreline at the edge of the sea where waves are coming and going due to the tidal effect - here you can literally feel the loss in bearing capacity under your feet).

In fine-grained subsoil the surface water soaks the ground. The ballast bed then shows deformations under operational load. These deformations frequently lead to the formation of very deep ballast pockets.

9. The ballasted track

Initial quality and rate of deterioration are the most important quality parameters of ballasted track.

The key to economically effective operation of the track is knowledge of the influencing parameters and their effect, as well as the extent to which maintenance measures can influence them.

One of the most important influencing parameters is the interaction between wheel and rail and the static and dynamic forces arising from this interaction.
A track is considered to be of "optimum" quality, when the total costs of the chosen track design, together with those of maintenance throughout the service life of the track used as designed, are at a minimum level.
To be able to objectively assess the condition of a track structure, it should be able to be described by significant measurable parameters. The mean defect in longitudinal level over a certain track section has proven particularly suitable to describe track quality.
Other usually applied track quality figures consist of weighed defects in longitudinal level, alignment,superelevation and other geometrical parameters.

Properties of the ballasted tack

Pitapostulates the existence of an optimum vertical stiffness. The results of his experiments show a great variation of stiffness below the sleeper, in a transversal as well as in a vertical direction. From this variation of stiffness he explains the greatly differing courses of settlement of certain track sections by means of a *FEM*model of the ballast bed, where he assumes a random scattering of the modulus of elasticity according to statistical experimental data.
Local variation in stiffness is the starting point of irregular ballast settlement also according to other scientific papers. The stiffness of the track itself hardly changes under operational load. The large range of ballast settlements and sleeper cavities are attributed to
•the accidental arrangement of ballast grains, and
•the scattering of initial compaction
The angularity of the ballast is very important for settlements. Ballast contamination leads to clogging of the cavities in the lower layers, which increases the ballast compaction and reduces its elasticity the ballast pressure grows, resulting in accelerated ballast destruction and, thus, also an increased

rate of deterioration. The contamination impedes the running off of water - subsequently damage due to frost threatens.

Satohas shown that the frequency distribution of the rate of deterioration of a line section is subject to exponential distribution.

The value of ballast pressure under the running wheel is often regarded as the decisive factor for the development of defects in track position. A mathematical connection between the ballast pressure and the frequency of track maintenance works is often formulated proceeding from the 2^{nd} or 4^{th} exponential law. The influence of the ballast pressure on the quality of the track position is overestimated according to a recent study. The increased occurrence of defects in track position in discrete centres of irregularities rather indicates that the dynamic effects of the interaction between wheel and rail are of great importance.

A study has shown a surprisingly low number of points of contact between the sleeper base and the ballast bed. The contact surface of wooden sleepers is between 4-10% and of prestressed concrete sleepers between 1-9%. The contact surface for new track amounting to 0.5-3% is amazingly low. The contact surfaces are increased by about 40% by tamping and stabilizing by the Dynamic Track Stabiliser. Apart from the use of the Stabiliserit is considered advisable to optimise the grain size distribution (admixture of 15% of permanent-way ballast of grain size 2) and the ballast grain shape and to use soled sleepers to increase the contact surfaces and, thus, to improve the load transmission behaviour. The higher the number of contact surfaces, the better the load transmission behaviour and the slower the fast initial deterioration under the first train loads after maintenance.

4.3 Up-graded of some SRC main line routes for higher speeds.

4.3.1 Introduction

There is always vague jargon about the construction of STANDARD GAUGE (SG) lines in SRC as means of up-grading their speed & capacity. There is no need to that because the NARROW GAUGE of SRC has not exhausted its ability to fulfil the duties expected for a hundred year to come if not more. To go (SG) you will go very far, new buildings, new stations, new workshops, new structure & loading gauges, new plateforms, new … new bilions& billions. FOR WHAT?!

Many railway authorities have successfully developed heavy rail routes for NARROW and STANDARD gauges but there have been some costly mistakes during development.

For this reason it is essential that experienced engineers and econometrists should be employed at an early stage to study all relevant factors for each scheme.[12]

The routes with heavy axle loads lines can be provided by strengthening the existing lines. [12]

Tables (4.2) & (4.3) compare trains characteristics and give their types [12

Table (4.2) comparison of train characteristics [12]

Train characteristic	Convertional Railway	Heavy Haul Railway unit trains
Train total mass	Less than 3000t	Greater than 5000t
Axle load	Less than 18t	Greater than 20t
Train length	Less than 800m	Greater than 1000m
Number of wagons	Less than 40	Greater than 100
Braking system	Vacuum& air	Greater than Air

Table(4.3) Track types with recommended sleepers [12]

track	Axle load	Rail joint type	Type of sleeper

Light	10 – 15	Fish plated or LWR	Steel ,wooden 180 kg concrete.
Medium	15 – 20	Fish plated or LWR	Wooden , 220 kg concrete.
Heavy	20 − 30	CWR	280 kg concrete.

4.3.2 Main features of standards of SRC main lines (75/90 lb/Yd.rails):

- Single line
- Narrow Gauge 1067mm (3′ 6")
- flat Bottom Rails

- Jointed Track
- Wooden &steel sleeper
- suspended Rails Joints
- Square Rail Joints
- Sleepers Density (N+2)
- Built-up Bolted Common Crossings
- Resilient Fastening& Rigid Fastening
- MaxGradient 1%
- MaxAxle load 16.5 tons
- MaxDegree of Curvature 4° 30" ($MinR$ 388)
- Max Superelevation (cant) 76.602mm (3″)
- Max Speed 60 Km/hr
- Cant of Bearing Plates 1in 20

❖ Examples in calculation of cant (superelevation)[12]

Example (1) as an exercise (master example):

- To find e (cant) of existing main lines of SRC which is already known to be 3" (76.602 mm)
- Equilibrium speed ≡ the speed in which a movimg vehicle shall be in equilibrium.
- We calculate the cant (e) in a curve of 3° with 60 km/h speed in a gauge of $3'6"$ (1067 mm).

  $\frac{e}{G} = \frac{F}{w}$,

  G≡ gauge,

  F ≡ Centrefugal force,

  w ≡ weight of vehicle

  $\frac{wv^2}{gR} = F$,

  v ≡ the speed in km/h,

  g ≡ gravity 32.2 ft/sec^2.

$$\frac{e}{G} = \frac{wv^2}{gRw} \Longrightarrow e = \frac{GV^2}{gR},$$

The American and colonial practice is to describe the curve by the angle subtended at the centre by a chord of 100 feet.

This latter system will be made clear by considering the whole circumference of a circle to be made up of 360 sections, each of 100 feet. Each of these sections will thus subtend an angle of 1° at the centre.[13]

The length of the circumference, 2R, will thus be for practical purposes 36,000 feet and the radius will be:

$$R = \frac{36{,}000}{2\pi} = \frac{36{,}000}{6.28}$$

$$= 5{,}730 \text{ feet.}$$

Thus a 1°curve has a radius of 5,730 feet.

R ≡the curve radius in ft or in.

- $R = \frac{5730'}{d^\circ} = \frac{5730'}{3^\circ} = 1910'$
- $G = 3'6'' = 3.5'$
- $v = \frac{60 \times 3.28 \times 1000}{60 \times 60} = \frac{104'}{3}$
- $g = 32.2'$

$$e = \frac{GV^2}{gR} = \left(3.5 \times \left(\frac{104}{3}\right)^2 \times \frac{1}{32.2 \times 1910}\right) 12 = 2''$$

2"Cant for curve of3°.

Cant for 1° $= \frac{2''}{3^\circ}$

The cant for SRC main line curve of 3° 30" (4.5°) is:

$$\frac{2}{3} \times \frac{9}{2} = 3''$$

Which is max cant in SRC

The max radius & min radius:

$\frac{5730'}{3.28} = 1746\ m$,$\frac{1764}{4.5°} = 388\ R_{min}$. (Main line)

$\frac{1746\ m}{12.8°} = 137\ R_{max}$. (Sidings)

Example (2) we calculate the cant with 120 km/h speed & see if it is within the allowable cant according to international standards.

$$\frac{e}{G} = \frac{F}{w}$$

$$\frac{wv^2}{gR} = F$$

$\frac{e}{G} = \frac{wv^2}{gRw} \Longrightarrow e = \frac{GV^2}{gR},$

To find e (cant) in curve 3° with 120 km/h speed in a 3′ 6" gauge= 3.5′

$e = \frac{GV^2}{gR},$

$R = \frac{5730'}{d°} = \frac{5730}{3°} = 1910',$

$$g = 32.2'$$

$v = \frac{120 \times 3.28 \times 1000}{60 \times 60} = \frac{328}{3} = 109.333\ m/s.$

$$e = \left(3.5 \times 109.333^2 \times \frac{1}{32.2 \times 910}\right) 12 = 6.83"$$

6.83" cant for 3°

Cant for 1° $= \frac{6.83"}{3°} = 2.28"$

The cant for a 4° 30" curve is

$$e = 2.28 \times 4.5 = 10.245"$$

(4.5°) $e = \frac{2.28 \times 9}{2} = 10.245"$

$(2°)$ e for $2°$ curve = $2.28×2= 4.56''$

$(2.5°)$ e for $2.5°$ curve $= 2.28 \times 2.5 = 5.7''$

For a speed of 120 km/h the maximum degree of curventive $4° \, 30'$ (for the 60 km/h speed) needs to be changed to a $2° - 2.5°$ curve to have a cant within the allowable limits.

Curvature & cant

In order to provide an acceptable level of passenger comfort, it is necessary to regulate the speed of trains in relation to radius of curve, cant &length of transition. On the basis of many years of experience, the following limiting values have been adopted by British Railways.

Maximum cant on curved truck	150mm.
Maximum deficiency of cant on plain line (c .w. R)	110mm.
Max .rate of change of cant or of cant deficiency	45mm/sec.
Desirable rate of change of cant or of cant deficiency	36mm/sec.

Using these values, it can be seen that the maximum and desirable gradients of cant &cant deficiency for 200 kph running are 1 in 1200 and 1 in 1600 respectively. It can also be calculated that the minimum radius of curve at this speed, using maximum cant deficiency , is 1800 meters while that at 175 k.p.h is 1400 meters [12].

Two annexes with this section (4.3) follow as references for understanding what is needed to up- grade the speed in an existing railway line.

Annexe (1)

Centrifugal force and overturning [14]

When a vehicle travels around a curve, centrifugal force acts on its centre of gravity and forces it outwards.

Centrifugal force results in an overturning moment. This is opposed by a stability moment. Where the vehicle speed exceeds the balancing speed for the curve, the net result of these moments is to transfer a portion of the wheel load from the inner to the outer rail. The effect would then be to apply a greater load on the outer rail than on the inner.

Additional flange force caused by excess, unbalanced centrifugal force would normally have little influence on the total flange force as it is usually considerably less than that portion of the flange thrust due to curving.

Unbalanced centrifugal forces, however, can contribute significantly to vehicle rollover tendency, particularly when the speeds exceed the design speed of the curve and/or there is a track defect that results in a sudden increase in superelevation deficiency.

Thus, for any given radius, by varying either the speed or the amount of superelevation (within limits) there is very little scope for reducing flange thrust.

Centrifugal force is given by standard physics as follows:

$$F = \frac{WV^2}{gr}$$

Where F = Centrifugal force in tonnes

W = Mass of vehicle in tonnes

V = Velocity in m/sec

r = Radius of curve in meters

g = Acceleration due to gravity, or 9.8m per second

S = Superelevation in millimeters

G = Gauge of the track, or centre to centre of rails adopted as 1500mm

The level of unbalance is represented by the cant deficiency. Standard requirements for design of curves are given in ESC 210.

Effect of centrifugal force and flange thrust

When vehicle speed exceeds the balance speed for the existing superelevation on a curve, the resulting centrifugal force has two separate effects,

a) A flange force acting against the outer rail, and

b) An increased load on the wheel running on the outer rail due to load transfer from the inner wheel.

As the speed increases, so does the centrifugal force, and both flange force and wheel load on the outer rail increase together. Actually, centrifugal force acts on the centre ofgravity of the vehicle body. The body rolls on its suspension, transferring additional load onto the wheels on the outer rail. Wheel climbing is less likely to occur due to increase in centrifugal force provided that the path followed by the wheel is smooth and not interrupted by a track defect.

Flange force, which promotes wheel climb, is opposed on curves at speed by the greater load imposed by centrifugal force, resulting in the wheel being held to the rail.

The probability of derailment by wheel climb, as measured by the ratio between the horizontal flange force and the vertical wheel load, is remote, however great the flange force, unless some other factor enters into consideration which tends to reduce the wheel load. For instance, a low joint on the outer rail may momentarily reduce the vertical wheel load on the guiding wheel while leaving the flange thrust almost unaffected and to such an extent that the wheel will mount the rail. Lurching may do likewise, and all such contributing factors may operate simultaneously to cause a derailment.

If the superelevation is much greater than that required for any given speed (i.e. superelevation excess), wheel load is transferred to the inner rail where it is least needed. In this case, vehicles travelling at slow speed, or speeds less than the curve balancing speed, may tend to climb the high rail because of the reduction in wheel load on the outer rail. This particularly applies to vehicles with a high centre of gravity.

Danger of overturning

Overturning can only take place when centrifugal force, which acts horizontally, and the effect of the mass of the vehicle, which acts vertically, combine to produce a resultant force that passes through or outside the point of contact between the wheel tread and the head of the outer rail. The vehicle is then in unstable equilibrium, as the load on the inner rail approaches zero, and any additional lateral force on the vehicle will result in overturning. The speed at which this can occur is called the "Critical speed for overturning."

Derailment by overturning on the high rail is extremely rare. A vehicle travelling around a curve at high speed is far more likely to derail owing to a high L/V ratio rather than overturning.

Overturning can occur on the low rail where a high centre of gravity vehicle is travelling at low speed and is disturbed by cyclic irregularities in superelevation. This causes the vehicle to roll towards the inside of the curve, bringing the resultant of the weight component of the vehicle and the inertial forces through or outside the low rail.

Overturning limits can be calculated from basic geometry by taking moments about the rail over which overturning occurs (normally the high rail).

F x COG = W x horizontal distance of COG from centre of high rail

Where F = Centrifugal force in tones

W = Mass of vehicle in tones

COG = centre of gravity

Annexe (2)

The Impact of superelevation (or cant Deficiency and why it’s important) [15]

Here is introduced the concept of unbalance or underbalance in operation through horizontal curves. In passenger rail terminology, this same concept is generally referred to as “cant deficiency.” The term is drawn from British and European practice where superelevation is referred to as “cant” and the term “cant deficiency” describes the circumstance where a vehicle operates through a curve with insufficient cant to achieve equilibrium.

Superelevation (banking or track cant) is a necessary ingredient for safe and comfortable curve negotiation. Superelevation is used to counteract the effects of centripetal acceleration (centrifugal force) on the vehicle and the occupants. The amount the outer rail is elevated is determined by the sharpness of the curve and the speed the vehicles operate through it.

Some definitions are in order:

- Balance Speed: The speed at which the combination of curvature and superelevation exactly balance the centripetal acceleration and the resultantforce vector is normal to the track plane.

- Cant Deficiency: Also known as Underbalance. This is the amount of superelevation, or cant that is missing from the track and would be needed toproduce a balance condition for the speed operated. Underbalance is generated by operating through a curve at speeds faster than the balance speed.

- Overbalance: This is the amount of excess superelevation, or cant that is in thetrack in order to produce a balance condition for the speed operated.Overbalance is generated by operating through a curve at speeds slower than balance speed. It is also generated by stopping in a superelevated curve.

Why are these important? Passenger comfort for one, maintenance and deterioration of the track structure for the other. In track that is used by both passenger trains operating at speeds of approximately 70 mph or greater and heavy axle freight trains, there is a compromise that must be reached concerning the elevation of curves. Higher superelevation is desired by the passenger operator for higher speeds through the curves. Lowersuperelevation is desired by the freight operator to protect againstoverturning of slow or stopped freight cars with high centers of gravity. There is also the effect of the excessive burden placed on the low rail by the heavy wheel loads of freight cars operating in the overbalance condition. This burden manifests itself by causing accelerated deterioration of the low rail surface geometry, head wear, or crushed head of the low rail, or increased superelevation. All of these have a snowball effect on the maintenance requirements of the curve.

Freight equipment can operate effectively at elevations up to 6 inches, although operators typically want elevations less than 4 inches. Passenger equipment can accommodate superelevations up to 8 or 9 inches, although a maximum of 6 or 7 inches is desired in passenger service as a comfort limit for a passenger’s ability to walk or stand on a train stopped in curve. Cant deficiency is used to increase the effectivesuperelevation by taking advantage of the passenger equipment's ability to negotiate a curve at speeds much greater than balance speed.

Therefore, design consideration can be made to accommodate freight operation by designing a curve with elevations between 4 inches and6 inches. Then, increasing the cant deficiency can create additional speed for the passenger train. Each inch of cant deficiency is equivalent to an additional inch of superelevation. For example, operating at 3 inches of cant deficiency on a 6 inch elevated curve is the equivalent of operating on 9 inches of superelevation.

Car body tilt systems have been used to gain even more speed in curves. Passenger cars can typically accommodate much higher cant deficiency safely than that which is comfortable for the passengers. To take advantage of this characteristic, the car body can be tilted to create a near balance condition in the interior of the car while externally it is developing quite a lot of centripetal acceleration. For example, for a tilting car operating at 9-inch cant deficiency on a 6-inch superelevated curve (i.e., the equivalent of 15 inches of superelevation), if the car tilts at 8 degrees, the net cant deficiency experienced by passengers in the car interior is approximately 2 inches. This a powerful method to increase speed on an existing line where major changes inalignment or curvature are not feasible.

4.4 Up-grading the same lines to 20 tones axle load

4.4.1 Introduction

This is a process of strengthening every component of the track. It involves introduction of new materials if the existing ones strength is not suitable, the addition of ballast to the appropriate depth, the treatment of the subsoil & the formation, strengthening of bridges, choice of the right materials, the drainage systems, much consideration for the maintenance works (manual, semi – mechanised or mechanised) etc. see fig.(4.5a) for the pressure distribution of the wheel force on the track components.

The target should not be engineering excellancy but the achievement of the job which will do its specified duties with minimum execution cost.

For types of tracks &their characteristics see tables (4.2) &(4.3).

Here under are annexes which cover the construction, histories, European& American view points on sleepers & their uses & the railway bridges.

Annexe (1)

Track structure[16]

Historical Background

Most of the mainline tracks in North America are now more than 100 years old and still remain on the original roadbed. It is important to consider the impact of initial construction and subsequent maintenance on the performance of current track structures.

Originally, the mainline tracks were constructed on a very compressed schedule using immediately adjacent soil sources, with the greatest emphasis placed on production rather than track quality. Furthermore, the initial construction was meant for trains that were much lighter and traveled much slower than today's vehicles.

The historic construction methods used 150 years ago would not meet today's standards. For instance, it was common to lay a skeleton track (track structure laid immediately on top of the subgrade with the tie cribs and shoulders being devoid of ballast) directly on either the original ground surface or on a minimum amount of uncompacted fill. Ballast, then, was placed as required to sustain traffic. In the mountains and similar steep terrain, side hill cuts and fills were common, with a portion of the subgrade supported on rock foundations and the remainder on loosely placed fill.

Soft track, groundwater discharge, bearing capacity issues, slope failures and sinkholes are all commonplace problems. Remediation of these problems must address not only what is apparent in the track structure today, but also the legacy of construction and maintenance over the past 100 years. For instance, it is not unusual for track that functioned very well for more than 50 years to suddenly develop severe geotechnical problems.

In solving problems today, the experiences and effects of the last 100 to 150 years ofrailway practice must be considered. Not only are the railways dealing with ever-increasing loads and ever-increasing traffic, but also a maintenance effort focused on rails and ties. Ballast, being less visible, receives less attention, and the subgrade, less still except when problems develop. Nonetheless, knowing the history of a section of track is an important component of effective track maintenance.

Components and Functions

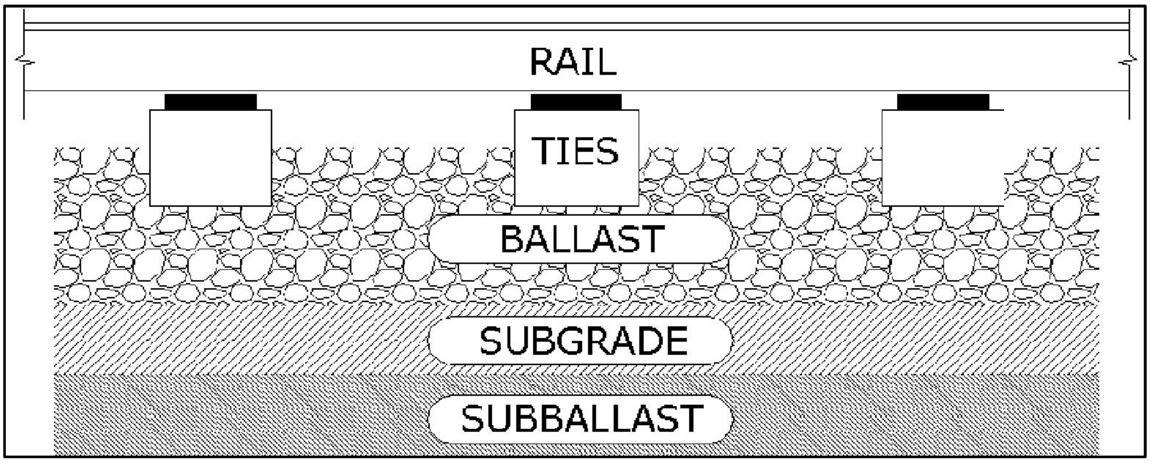


Fig. (4.5b) The Track Structure

The track structure is made up of subgrade, sub-ballast, ballast, ties and rail as illustrated in Fig. (4.5) each of these contributes to the primary function of the track structure, which is to conduct the applied loads from train traffic across the subgrade safely. The magnitudes of typical stresses under a 50,000 lb axle load are shown in Fig. (4.6) these stresses are applied repeatedly, and each repetition causes a small amount of deformation in the subgrade. In theory, the track structure should be designed and constructed to limit rail deflections to values which do not produce excessive rail wear or rates of rail failure. In reality, cumulative deformation of the subgrade causes distortion of the subgrade, leading to formation of .ballast pockets" (Fig. (4.6)) or outright shear failure.

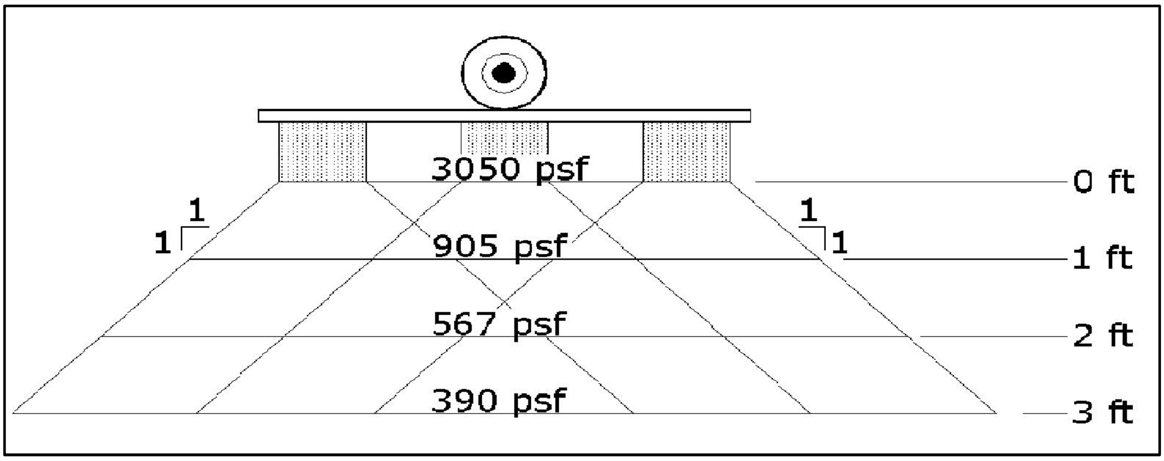


Fig. (4.6) Stresses Imposed by Train Axle Load

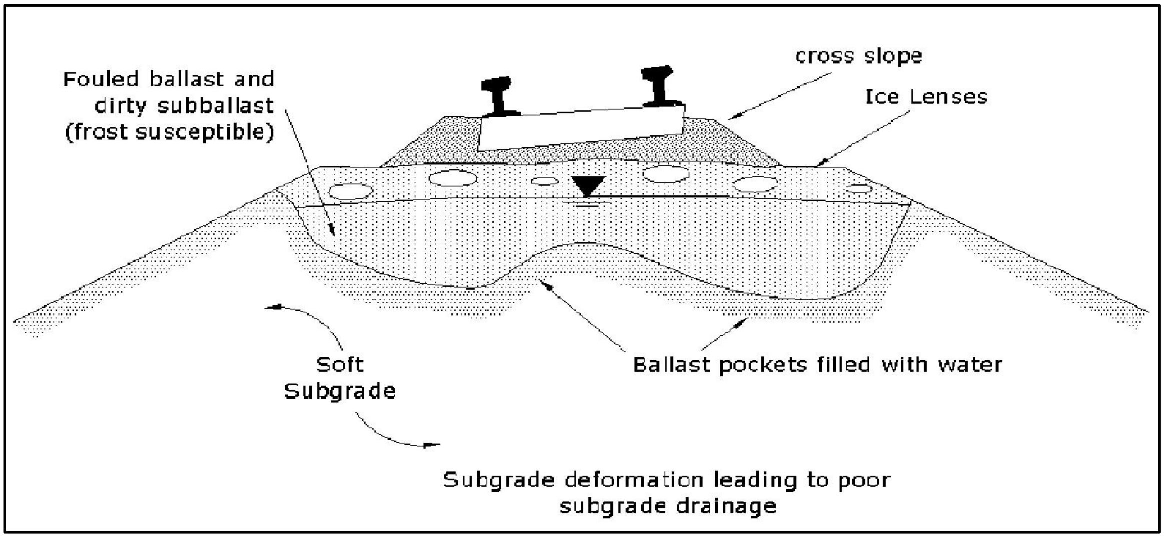


Fig. (4.7) Ballast Pockets in Subgrade

Subgrade

The purpose of the subgrade is to support the track structure with limiting deflections .Every subgrade will undergo some deflection (strain) as loads (stress) are applied. The total displacement experienced by the subgrade will be transmitted to other components in the track structure. The stiffer the subgrade (i.e., the higher the modulus of elasticity), the lower the deflection values will be. It is important that adequate subgrade strength and stiffness be available on a year-round basis, particularly during spring thaw and following heavy precipitation events.

The strength, stiffness and total deflection of the subgrade can be improved by:

- Carefully selecting materials that are naturally strong (sand, gravel, boulders) with ahigh angle of internal friction.
- Limiting access to water to avoid buildup of porewater pressure and subsequent reduction of strength.
- Improving the soil properties, using techniques such as compaction, in situdensification, grouting and preloading.
- Maintain good drainage.
- Maintain stable subgrade geometry.

Sub-ballast

The purpose of sub-ballast is to form a transition zone between the ballast and subgrade to avoid migration of soil into the ballast, and to reduce the stresses applied to the subgrade. In theory, the gradation of the sub-ballast should form a filter zone that prevents migration of fine particles from the subgrade into the ballast. In practice, insufficient attention has been placed to sub-ballast gradation historically, and much of the sub-ballast does not adequately perform that function. This notwithstanding, the number of occurrences of subgrade contamination of ballast are relatively few.

How Track Fails

In a nutshell, track fails when differential rail deflections become excessive. This differential deflection may be expressed in differential elevation between tracks, punching of ties, elastic or plastic deformation of the subgrade, or degradation of ballast.

When the bearing capacity of the subgrade is exceeded, the subgrade will deform plastically, resulting in a small amount of permanent deformation under each wheelload. A progressive deterioration of the track begins, as illustrated in Fig.(4.8 &9). It starts with minor deflections and may progress to a fully visible surface heave, where subgrade material is pushed above the elevation of the rail and ties. Under those conditions, ballast drainage is impeded, resulting in further softening and degradation of the subgrade to a point where large, saturated pockets of ballast are trapped in the subgrade.

Frost heave and further degradation commonly follow, leading eventually to a severe loss of utility of the track structure

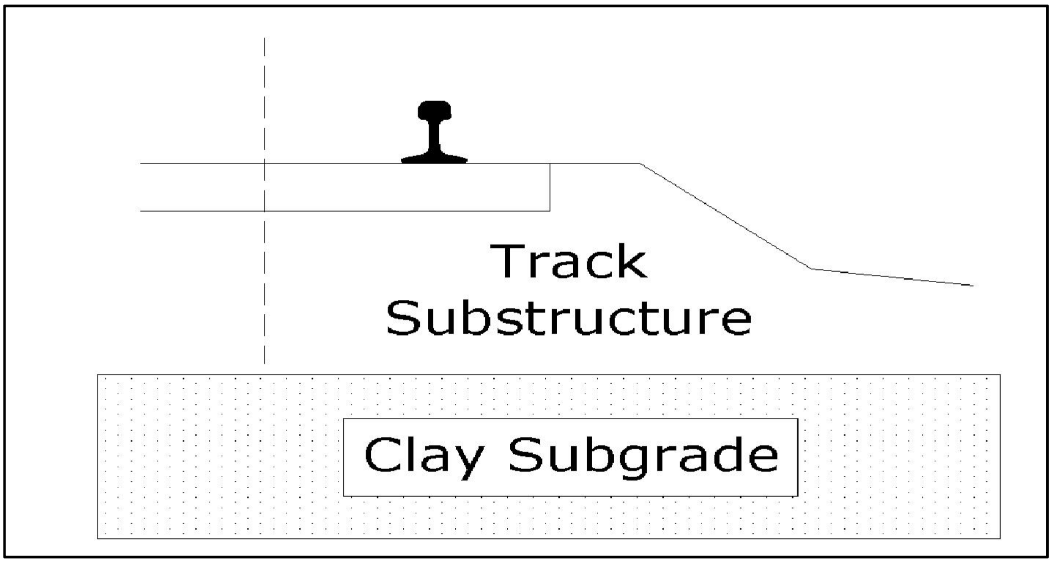


Fig. (4.8) Stable Site

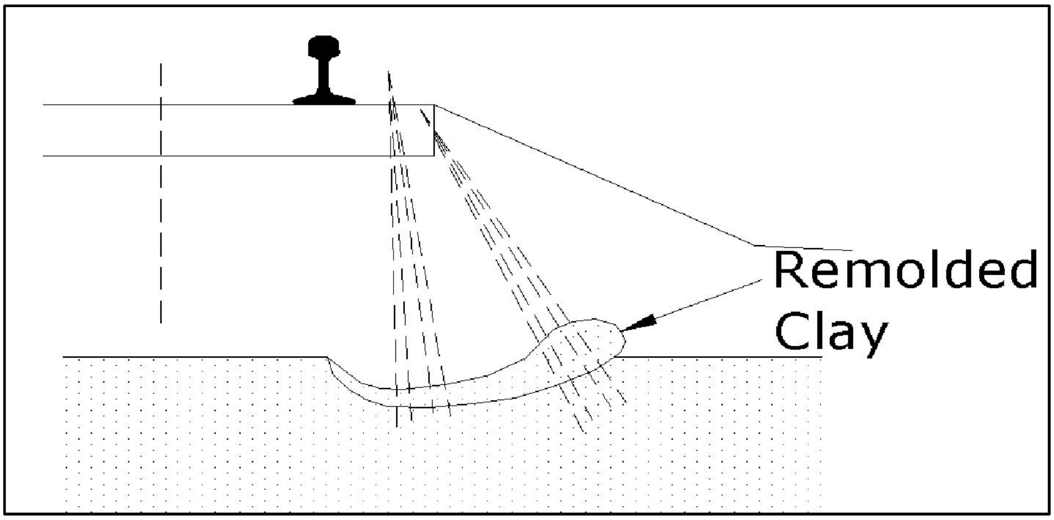


Fig. (4.9) Onset of Instability

Settlement

Basic Theory

Settlement results when the soil changes volume, when load is added because pore fluid is squeezed from the pores. The characteristics of settlement behavior include:

- Very little settlement occurs during initial placement of fill, since virtually novolume change takes place until the preconsolidation pressure is reached. Thus, the initial settlement is mostly elastic in nature.
- Beyond the preconsolidation pressure, settlement increases with the increasingload. This is called the zone of primary consolidation, which takes place until all ofthe excess porewater pressure in the soil is dissipated. This may take from severalminutes in the case of sand or gravel, to tens of years in the case of lowpermeability clay. For this reason, sand and gravel are preferred, both asfoundations and as fill construction materials.
- Even after all porewater pressure is dissipated, settlement may go on in some soils.This is called secondary consolidation, and is particularly common in organic soilssuch as peat or organic floodplain deposits. This is the reason that settlement iscommon where these materials from either the fills or the foundation for fills.Secondary consolidation may go on for several decades or more.

While total settlement is important, it is the differential settlement that causes tracks to be rough and some parts of fills or structures to settle more than other parts. As a rule of thumb, differential settlement within a fill typically is about 50 percent of the total settlement. This is important because it is the differential settlement that causes the need for resurfacing.

Influence of Construction Methods

Construction methods can have a dramatic effect on the total amount of settlement experienced. Settlement occurs both in the fill and in the foundation. Uncompactedfills can experience settlement equal to more than 10 percent of the fill height, while properly compacted and conditioned fills may experience no settlement. Further, moisture conditioning of the fill plays a very important role in controlling future settlement. For instance, any cohesionless fill compacted dry of optimum is likely to experience severe settlement upon first wetting. Settlement in excess of 10 percent of fill height has been recorded where fills were compacted dry and subsequently were saturated by runoff or ponding water. A combination of heavy watering and compaction using a vibratory roller produces the best results, reducing settlement to manageable values.

Influence of Soil Type

Settlement can be experienced with all types of materials. Rapid settlement of foundations will be experienced in sand and gravel, while settlement of clay foundations may take several decades to complete. As indicated earlier, secondary consolidation of peat or organic soil foundations can go on for many decades, requiring ongoing maintenance.

The type of material used in the construction of fills also influences settlement behavior. Sand and gravel are preferred, because once wet and compacted, they will produce a stable fill with low settlement characteristics. Silt is difficult to use as construction material, because it is extremely moisture sensitive and very difficult to compact. Clay must always be compacted, since if placed loose, long-term, chronic consolidation settlement can be expected. Highly organic soils should never be used for construction of fills for railway subgrades, because of their low bearing capacity and high resilience, and because of the characteristic long-term secondary consolidation that can be expected. In some cases, low level organic soils may be acceptable for use below the subgrade level.

Settlement is pure vertical movement, which may be due to expulsion of pore fluidfrom the soil. There is no shear movement associated with settlement. On the other hand, when the shear strength of foundation soils is exceeded, shear displacement commonly takes place in both horizontal and vertical directions. While bearing capacity failures and other forms of slope instability may produce characteristic settlement profiles that require lifting, strictly speaking, this is not a settlement problem.

Settlement can usually be improved by improving soil characteristics by compaction or jet grouting in the case of coarse grained soils, or compaction or replacement of clay soils. Such treatment will normally not provide any relief if the shear strength of the foundation soils is being exceeded.

Summary

In summary, the approach to natural hazard management includes:

1. Understand the contributing factors:

- Terrain components:
  - Landforms
  - Natural materials
  - Groundwater regime
  - Earthquakes
- Climatic influences:
  - Precipitation
  - Frost, thawing
  - Runoff
  - Erosion
- Vegetation influences:
  - Vegetation impacts
  - Vegetation management
  - Deforestation
- Influence of humans on aspects such as:
  - Drainage
  - Development
  - Maintenance
  - Engineering
  - Work on railway property by others

2. Understand the mechanisms:

•Principal of effective stress:

- Role of pore water pressure
- The difference in behavior between cohesive and cohesionless soils

•How the track structure works:

- Dependent on subgrade
- Principal of limiting stresses on subgrade
- Importance of ballast
- Importance of pore water pressure

•Landslides

•Bearing capacity

•Drainage

•Seepage

•Freezing and thawing

•Settlement

3. Identify hazards:

• By inspection

• Concentrate on high risk areas

• Concentrate on high risk periods for runoff and breakup:

• Extreme precipitation events

4. Describe hazards:

• Landforms

• Materials

• Groundwater conditions

• Geometry

• Mechanisms (type of hazard)

• Effects

• Actions required

5. Take appropriate action:

• Immediate action to mitigate hazard:

- Urgent
- Priority
- Routine
- Request assistance
- Monitor

6. Provide documentation:
   - Notes and diaries
   - Photographs
   - Records of observations

Annexe (2)

Sleepers (ties) [17]

There are five main types of support for rail, namely:

Concrete slab

Concrete sleeper

Steel sleeper

Softwood sleeper

Hardwood sleeper

All have their advantages and disadvantages from a technical point of view.

In broad terms the factors which have to be borne in mind in economic selection are not necessarily those which will give technical excellence. For instance indigenous materials are likely to be cheaper than imported sleepers and indeed different governments may wish to use such materials because of the hard currency or strategic situation. On the other hand there is not much point in using softwood if there are local conditions such as termites/fungi which would destroy them within a very short period. It is a matter of weighing up all the factors of first cost against availability against life against maintainability. Even then a definite decision cannot be made in isolation from the rest of the track components and several alternatives should be evaluated to ensure a minimum overall cost solution.

I think that environmental as well as economic pressures will force most railways away from timber sleepers in the foreseeable future. Concrete and steel sleepers, properly designed, can do everything that timber can do. Concrete has the problem of weight which makes it expensive to export/import although if the projects are big enough local production can overcome some of these logistic problems. Personally I think that if the steel firms make a sustained attack on the sleeper market with a product in the right price range they will eventually take the majority of the business in most countries. Steel sleepers are light compared with concrete, are less bulky and therefore are easier to transport and handle and have a high residual value.

Ties are typically made of one of four materials:

• Timber
• Concrete
• Steel
• Alternative materials

The purpose of the tie is to cushion and transmit the load of the train to the ballast section as well as to maintain gage. Wood and even steel ties provide resiliency and absorption of some impact through the tie itself. Concrete ties require pads between the rail bases and tie to provide a cushioning effect.

Timber Ties

It is recommended that all timber ties be pressure-treated with preservatives to protect from insect and fungal attack.3 Hardwood ties are the predominate favorites for track and switch ties. Bridge ties are often sawn from the softwood species. Hardwood ties are designated as either track or switch ties. Factors of first importance in the design and use of ties include durability and resistance to crushing and abrasion. These depend, in turn, upon the type of wood, adequate seasoning, treatment with chemical preservatives, and protection against mechanical damage. Hardwood ties provide longer life and are less susceptible to mechanical damage.

Track Ties

Timber track ties are graded with nominal dimensions of 7" x 9" x 8’-6" or 9’-0" or smaller ties which are 6" x 8" x 8’-0". (See Fig. (4.10)) The 6" x 8" x 8’-0" are typically utilized for sidings, industry tracks and very light density trackage. An industrial grade of both ties is also available. These ties have more wane, bark, splits or other surface related defects than recommended under the timber grading rules. Both AREMA and the Railway Tie Association (RTA) publish specifications and standards relating to the grading of timber and the definitions for the above timber physical characteristics. The cost savings may make industrial grade ties attractive for some plant trackage exposed to infrequent and light tonnage. It is generally acknowledged that the quality of hardwood tie available today does not meet yesteryear’s standards. Thus, the additional cost of providing gang plates, S-irons or C-irons for the tie ends may be a worthwhile investment in extending tie life from end splitting failures. Track ties may be ordered adzed and predrilled for the appropriate rail section to be used if desired. Secondhand ties, reclaimed from line abandonments, may also be available. There is wide debate regarding the suitability and cost effectiveness of using recovered ties. Deterioration of that part of the tie previously buried in the ballast occurs rapidly once the tie is exposed to the air. If second-hand ties are used, do not turn the tie over, thus providing a fresh surface for the top of the tie. These ties will deteriorate very quickly. Better to plug the tie, adze the surface if necessary and insert the tie as it was originally orientated. Occasionally, softwood ties may be specified for a track tie. Their use is limited to temporary track situations such as shoe-fly's, etc., or where tonnage is very light or hardwood species are prohibitive in cost.

For quality maintenance, ties should be not less than 8 ft. 6 in. in length. For moderately heavy or heavy-traffic conditions, especially on curves of 6 degrees or more, the 9-ft. tie is preferred, 7 in. by 9 in. in cross-section, because of the greater stability from the larger support and friction area. It also assists in restraining continuous welded rail.

For lines of moderate to medium tonnage, a tie spacing equivalent to 22 ties per 39-ft. rail (21-1/4 in.) is sufficient. Heavy tonnage lines or lines with sharp curves will find 24 ties per rail panel (19-1/2-in.) to have advantages in holding gauge and reducing bending moment stresses in the rail.

Fig (4.10) Hardwood Track Ties – Photo by J.E.Riely

Switch Ties

Switch ties Fig. (4.11) are commonly hardwood species, usually provided in either 6" or 12" increments beginning at 9'-0" up to 23'-0" in length. Nominal cross-section dimensions are 7" x 9", although larger ties are specified by some railways. The primary use for switch ties is relegated to turnouts (thus their name). However, they are also used in bridge approaches, crossovers, at hot box detectors and as transition ties. Some railways use switch ties in heavily traveled road crossings and at insulated rail joints. Switch ties ranging in length from 9'-0" to 12'-0" can also be used as "swamp" ties. The extra length provides additional support for the track in swampy or poor-drained areas. Some railways have utilized Azobe switch ties (an extremely dense African wood) for high-speed turnouts. The benefits associated with reduced plate cutting and fastener retention may be offset by the high import costs of this timber.

Fig(4.11) Switch Timber

Softwood Ties

Softwood timber figure(4.12) is more rot resistant than hardwoods, but does not offer the resistance of a hardwood tie to tie plate cutting, gauge spreading and spike hole enlargement (spike killing). Softwood ties also are not as effective in transmitting the loads to the ballast section as the hardwood tie.

Softwood and hardwood ties must not be mixed on the main track except when changing from one category to another.

Softwood ties are typically used in open deck bridges.

Fig. (4.12) Softwood Timber - Photo by J.E.Riley

Concrete Ties

Concrete ties Fig. (4.13) are rapidly gaining acceptance for heavy haul mainline use, (both track and turnouts), as well as for curvature greater than 2°. They can be supplied as crossties (i.e. track ties) or as switch ties. They are made of pre-stressed concrete containing reinforcing steel wires. The concrete crosstie weighs about 600 lbs. vs. the 200 lb. timber track tie. The concrete tie utilizes a specialized pad between the base of the rail and the plate to cushion and absorb the load, as well as to better fasten the rail to the tie. Failure to use this pad will cause the impact load to be transmitted directly to the ballast section, which may cause rail and track surface defects to develop quickly. An insulator is installed between the edge of the rail base and the shoulder of the plate to isolate the tie (electrically). An insulator clip is also placed between the contact point of the elastic fastener used to secure the rail to the tie and the contact point on the base of the rail.

Figure(4.13) Concrete Ties . Photo by KevinKeefe

Steel Ties

Steel ties figure(4.14) are often relegated to specialized plant locations or areas not favorable to the use of either timber or concrete, such as tunnels with limited headway clearance. They have also been utilized in heavy curvature prone to gage widening. However, they have not gained wide acceptance due to problems associated with shunting of signal current flow to ground. Some lighter models have also experienced problems with fatigue cracking.

Fig. (4.14) Steel Ties

Alternative Material Ties

Significant research has been done on a number of alternative materials used for ties. These include ties with constituent components including ground up rubber tires, glued reconstituted ties and plastic milk cartons. Appropriate polymers are added to these materials to produce a tie meeting the required criteria. To date, there have been only test demonstrations of these materials or installations in light tonnage transit properties. It remains to be seen whether any of these materials will provide a viable alternative to the present forms of ties that have gained popularity in use. Fig. (4.15)

Fig. (4.15) Alternative Type Material Tie

Annexe (3)

History of railway bridge engineering [18]

WilliamWorthington of the Smithsonian Institution presented a historical survey of American railway bridges at the 1991 meeting of the AmericanRailwayBridge and Building Association. He covered 19th and early 20th century developments thoroughly and this section summarizes part of his presentation.

Displayed in the National Museum of American History next to the John Bull, Stephenson's steam locomotive, which ran in New Jersey in 1831, is the nation's first cast and wrought iron RailroadBridge, built in 1845 by RichardOsborne for the Philadelphia and Reading RR. This bridge was in use until 1901.

Of course, stone was the preferred bridge material when promptly and economically available. However, stone construction was slow and expensive. Fortunately, the continent was covered with forests and wood was the best solution where available, despite its structural limitations and fire hazard. Many stone bridges were constructed and a large number of these 19th century masonry bridges are still in use.

Using wood, American railway bridge designers soon played a lead role in bridge truss design. Almost one wooden bridge design patent was issued each year in the first halfof the 19th century. Among those were the Pratt and Howe truss designs, which could be used with both wood and metal structures. Although their life was limited, wood bridges made it possible to extend a line quickly and cheaply.

A key wooden structure, making it possible to keep railway grades low, was the timber trestle. It could be constructed quickly and would have a life of at least 15 years.Numerous trestles were ultimately converted to fills by hauling material to the site cheaply by train. Worthington believes that the 1892TwoMedicineBridge on the St. Paul, Minneapolis and Manitoba Railway is perhaps the ultimate example of the 19thcentury wooden bridge builders' art. It was 750 feet long and 210 feet high. Of particular interest is that by that time steel was the material of choice. But location, cost and time constraints dictated a wooden trestle at this location.

The distinctive Bollman truss, incorporating elements of truss and suspension bridgedesign, was used in the 1850-70 period to replace many of the first generation wooden bridges, particularly on the B & ORR.

As in other parts of the world, there were failures. One notable 1887 accident, costing23 lives, occurred on the Boston and Providence RR. Before the failure, loose nuts had occasionally been found below the bridge. The failure was traced to the fracture of two hangers suspending the track structure from the top chord. They were poorly designed and of inadequate strength and the fracture had existed for a considerable time. Theodore Cooper, father of the bridge loading analytic system still in use, characterized this as

"An abortion in design and construction in which no engineer had any part."

Perhaps the most unusual American railway bridge of that century was the Niagara Gorge suspension bridge designed and built in the 1860's by John Roebling, designer of the Brooklyn Bridge. The only feasible construction technique available was the suspension type, which with stiffening could accommodate the light railway fleet of the day. Rail traffic used the upper deck and vehicular movements were on the lower level.

Despite limiting rail traffic to 5 mph the deck truss flexed somewhat. In 1869, Mark Twain observed that when crossing it you:

"Divide your misery between the chances of smashing down 200 feet into the river below and the chance of having the railroad train overhead smashing down on you. Either possibility is discomforting taken by itself, but mixed together they amount to positive unhappiness."

One of the most significant steps taken in bridge construction after the Civil War was the application of the ancient method of cantilever construction.

During the 1870-90 period, steel manufacturing developments created a market for this material and the steady supply of reasonably priced products in many shapes permitted construction of all-steel bridges. As bridges became stronger, more powerful and heavier, locomotives required even stronger bridges. For example, on the B&ORR, the heaviest engine in 1865 weighed 91,000 pounds, while in 1890 it had increased to 13,300 pounds and another 25 years later in 1915, it reached 463,000 pounds.

To accommodate these increased weights, speeds on older bridges had to be limited to an unacceptable 15 mph and bridge replacements were necessary. The first all-steel bridge in Glasgow, MO was replaced in 1901 after only 22 years of service.

Twentieth century bridge design exhibited a sturdy sameness. Smaller bridges were likely assembled of Pratt or Warren designs. Some longer and higher bridges were built, culminating in the high Huey P. Long Bridge over the Mississippi River at New Orleans.

New Technology – Bridge Developments in the Last Twenty Years

Innovation and technology development over the past twenty years has focused on the challenges related to the maintenance of existing railway bridges; as well as the design and construction of new bridges required to improve railway infrastructure safety and reliability.

Existing Railway Bridges: Inspection and Assessment

Like many railway-engineering personnel, the railway bridge infrastructure is aging. The existing bridge infrastructure is also being subjected to heavier axle loads and increased traffic volumes. The planning and design work associated with the assessment and maintenance of existing railway bridges is an engineering challenge requiring an understanding of the modern railway live load regime and the behavior of railway bridge structures.

Inspection of railway bridges has improved thorough the use of on-track bridge inspection vehicles and various non-destructive testing techniques that allow the engineer or inspector to obtain a thorough understanding of existing bridge conditions in a safe manner. There have also been many bridge access safety improvements in recent years such as the provision of fall protection lifelines and walkways on bridges.

Modern railway live loads are of large magnitude and frequency. While heavy locomotive weights have not increased substantially since the 1920.s, car weights have increased considerably. Modern car axle loads are of the same magnitude as locomotive axle loads. This means that existing railway bridges are subjected to many more applications of heavy axle loads than envisaged at the time they were originally designed and constructed.

The resulting increased stress ranges and greater number of cycles of load precipitates fatigue damage accumulation in some bridge components. Recent developments associated with structural analysis, stress-life fatigue behavior and crack behavior, enable the railway bridge engineer to assess the safe fatigue life of railway bridges. The railway bridge engineering community has been instrumental in developing improvements in the stress-life testing of components with characteristics typical of existing steel railway bridges. Railway bridge engineers have also been leaders in the use of linear elastic fracture mechanics in conjunction with acoustic emission monitoring and other non-destructive techniques for fatigue life assessments.

Modern computer programs have brought a host of analytical tools, such as three dimensional structural analysis, to the engineer's desktop. Modern non-destructive testing techniques, such as strain measurement and ultrasonic testing, can be used with advanced structural analysis to gain a better understanding of structural behavior of components and details. Furthermore, recent developments in railway bridge strength rating methods have allowed for strength rating calculations based on load regimes on bridges over an indefinite period of time and at infrequent intervals.

Innovative techniques and materials strengthened with fiber reinforced composite materials and cable post tensioning have been used to strengthen railway bridges.

Bridge engineers have been able to develop bridge replacements and/or rehabilitations on a project and program basis through the use of computer based bridge management inventory and condition rating systems. These developments have enabled railway bridge engineers to propose appropriate and cost-effective rehabilitation and strengthening of existing structures to maximize the life of the structure.

New Railway Bridges: Materials, Design, Fabrication and Construction

Replacement of railway bridges becomes necessary when economical rehabilitation and strengthening are not feasible. To construct safe, cost-effective and maintainable railway bridges, engineers have adopted recent technology developments in the areas of analysis, materials, design and fabrication.

The computer is playing an important role in the analysis of structures. However, while sophisticated computer analysis is available and used by modern bridge engineers, it is not a substitute for an understanding of structural behavior. Many experienced bridge engineers may not know their way around a computer keyboard, but have an intuitive understanding of structural behavior that electronics technology cannot completely replicate. However, for experienced bridge engineers, an improved understanding of the load and force distribution is available through advanced computer structural analysis.

Over the past 20 years railway car axle loads have increased by more than 30%. Investigation into the dynamic stresses imposed on railway bridges and the stress-life behavior of bridge components have permitted improved engineering designs. Longitudinal traction loads due to new AC high adhesion locomotives have also been identified and included in modern railway bridge designs. Improved understanding of serviceability issues such as fatigue, deflection, vibration and concrete crack control under heavy axle load regimes has been facilitated by recent railway bridge engineering research efforts.

Design methods such as limit states methods for concrete structures have improved the reliability of concrete bridges. Modern seismic deign methods based on performance limit states have been developed in recent years. Composite steel-concrete structure design has been developed for railway bridges to produce economical, easily constructed and maintainable ballasted deck structures.

Material improvements have been considerable over the past two decades. Alloyed weathering steels that are resistant to atmospheric corrosion with good fracture toughness and high strength (yield strength up to 70 ksi and 100 ksi) have been used in the design and construction of new railway bridges. High strength concrete has made possible the efficient design of heavily loaded railway bridges with improved durability. Prestressed concrete has also been utilized for short span construction. Economical concrete box girder and slab bridges using precast prestressed and cast-in-place post tensioned technology have been used for ballasted railway bridge construction. Precast segmental construction has provided for cost effective substructure and superstructure replacement with minimum interruption to traffic. Technology development for the economic replacement of existing railway timber bridges has involved developments such as prestressed concrete rehabilitation and replacement components.

Welding technology improvements have enabled the economical construction of steel bridges with improved fatigue characteristics. Computerized shop fabrication has improved fabrication accuracy and efficiency.

In recent years, CWR has been installed on both open and ballasted deck bridges due to recent work on the understanding of effects of bridge movements due to thermal expansion, particularly on open deck type bridges.

Protective coatings and paint materials and methods improved considerably over the past 20 years. Zinc rich paints, epoxy and polyurethane paint systems for shop painting and overcoating have been developed.

It is expected that technology improvement in the area of railway bridge engineering will develop at an increased pace due to the need to maintain, rehabilitate and reconstruct an aging railway bridge infrastructure.

4.5 Stability & safety

4.5.1 Introduction

The stability & safety are very important issues in every railway. Most of the research and development is done in these areas. Big funds are directed to these fields because the results are very rewarding to both designers / manufacturers & the users.

The studies which are the back bone to the stability & safety are train track dynamics & derailments.

Appendix Z by the end of the report contains:

a) A short description of the principles of wheel/rail interface
b) Vehicle &train dynamics.
c) Derailments.
d) A power point presentation of train track dynamics.

CHAPTER FIVE

CONCOLUSIONS & RECOMMONDATIONS

5.1 Introduction

1) The locomotive bogie is a very important unit which has direct impact on the performance, reliability, maintainability, stability, safety & operation of the locomotive itself and the track. The cost of the bogie with its traction motors may amount to 35% of the price of the locomotive.

Unfortunately SRC specifications for locomotive bogie are of general description which do not help in the stage of evaluating the offered locomotives.

In the case of wagons specification it is even worth giving the details of a bogie while specifying another design.

2) SRC management, since 1990, executed locomotive purchases out of the specified SRC rules & conduct for purchasing. This has lead to deviations from specifications. Two deviations have been specified in this project:

a) The deviation from the medium speed traction diesel engine to the high speed diesel engine.

b) The deviation from the axle hung nose suspended system for traction motors to the rolling bearing motor suspension units which is of better design but it needs to be proven for a long time to be accepted.

The deviations in the motor suspension system (by purchasing an <u>UNPROVEN</u> product lead to a phenomenal fault in the wheelsets of the CRS

SDD_1 Chinese locomotives which needs to be thoroughly investigated & solved.

As to the deviation in the diesel engine rotational speed we shall recommend practical test runs for locomotives with medium speed &others with high speed according to the required performance specified in the locomotive performance and compare their fuel consumption and their time keeping.

3) The NARROW GAUGE of SRC main line routes has NOT EXHAUSTED its TOP capacity targets up to now.

The same narrow gauge of 3′6″(1067 mm) is in Japan & South Africa up to now with 160 km/hr speed & 20 tons axle load.

Our project produced results for the up-grading of the SRC existing main line routes to a higher speed of 120 km/hr and higher axle load of 20 tons.

The capacity of SRC by these up-gratings is expected to serve Sudan freight transportation needs for hundred year or more to come.

There is always a vague jargon that SRC needs to up-grade its capacity of transportation by constructing new railway lines with STANDARD GAUGE (1435 mm) and DOUBLE TRACK.

If such thing happened!! This will be a disaster to the people of the Sudan.

5.2 Conclusions

1. SRC freight wagons fleet has 84% conventional bogies which are very inferior in design, cause a lot of problems with high rates of derailments. 16% of the wagons’ fleet is of developed bogies. To highlight the advantages & disadvantages we present this study on comparison of bogies performance.

It has long been recognized that the conventional three-piece freight wagon bogie presents certain inadequacies which are particularly noticed under the more intense or higher speed service that railways are attempting to achieve these days. This inadequacy results in high wear rates on bogie and car body components particularly causing excessive wheel tread and flange wear as well inherent faults in the three-piece bogie design namely lateral instability on tangent tracks and high lateral forces on curved track.

A wheelset is centered on the rails by steering moment. This moment is generated by longitudinal forces developed when the conical wheel treads are laterally offset, causing a differential in the rolling radii of two wheels on one axle, the amount of which depends on the lateral offset. Since the throat location has a larger rolling radius than the outer portion of the wheel, the wheel riding up the rail will travel further per revolution than the opposite wheel on the same axle. Facing the direction of the travel, the wheelset will tend to turn counterclockwise if it's the right hand wheel which is riding up to the rail. At higher speeds, the wheelset will tend to oversteer and reverse reaction takes place. This is called hunting. Higher frictional resistance between bogie and wagon body or lower speed will permit the wheelset to return to a more stable tracking condition. This, of course, inhibits any attempt to operate at increased speeds.

On the conventional bogie, on tangent track, lateral constraint prevents yaw of the wheelsets and prevents them from steering. Steering to maintain the centered position of the wheelset on the rails has to be achieved by turning the whole bogie. Even if rotational dampening of the bogie is low in comparison with the high steering moment, mass inertia will cause the bogie to oversteer, once bogie rotation is started, displacing the bogie to the opposite side and reversing the forces. The typical hunting cycle is thus initiated. As each axle in the bogie is going in the same direction, flange contact causes the bogie to parallelogram, thus increasing the angle of attack of the wheel against the rail. The wheels will climb the rail on the throat radius at the limit of each lateral excursion, increasing the rolling radius differential, causing severe instability. The action is cyclical and develops into energy impact level is sufficient to overcome the inertial and frictional dampening components of bogie mass, centre and side bearing friction.

Hunting can be extremely severe when the car body lateral motion becomes synchronous with the bogie motion.

Lateral displacement of the wheelsets generates steering forces and steering moment in a steering bogie just as in so-called conventional bogie. But the bogie equipped with steering ability behaves in an entirely different manner. The rubber shear pads at the journals allow the longitudinal steering forces to constantly steer the wheelsets against the controlling force of the shear load of the rubber pads. The wheelsets, therefore, do not have to overcome the rotational friction and inertia of the whole bogie to make small steering corrections. In other words, the shear pads apply resilient control over the wheelset longitudinal movement relative to the side frames, even at maximum operating speeds, thus, elastically balancing the steering forces.

In a conventional bogie, each wheelset tends to act in the same way at any given point, thus encouraging the parallelogram defect. In the steering bogie, the steering arms are collected by cross braces or coupled interconnections, so that each wheelset thus provides the other with a counter-active stabilizing effect. The leading wheelset initiates the steering action and induces opposed steering in the trailing wheelset, but is inhibited from overreacting by the inherent resistance of the trailing wheelset. The trailing wheelset develops a steering moment opposite to leading wheelset, opposing forces which facilitate steering on curves, but which provides counteracting forces to prevent overcoming the bogie-rotational resistance on straight track.

On curved track, the wheelsets steered in the same manner as on a tangent track. The wheelset whoever, will find a position which is slightly offset laterally towards the outer rail because the outer wheel has to travel further and must roll on larger radius to do so. The same steering principle, of course, applies to a bogie with two separate wheelsets, but, of course, there is a restraint placed on individual wheelset steering by the bogie side frame.

To negotiate a curve, a bogie must be rotated about a centre plate to the appropriate angle relative to the wagon center line. if the bogie resistance to rotation due to inertia and/or friction in greater than the total longitudinal steering force developed at the wheel-rail interfaces, no rotation will occur until the leading outer wheel flange contacts the outer rail of the curve. The result of flange contact is always detrimental, but depends largely on the type and worn condition of the bogie involved.

In conventional bogies, there are degrees of freedom and conversely constraints that prevent the steering forces from developing on the one hand and developing to an excess on the other.

The principal constraints are:

(1) A low lateral constraint leading to low-resistance to relative lateral displacement of the wheelsets or parallel gramming.

(2) High longitudinal yaw constraints between the side frames and the bearing adapters which inhibits the steering action of the wheelsets. The application of rubber shear pads to the connection between the wheelsets and the bogie frame provides the vitally necessary elements of lateral constraint and the degree of yaw freedom in the correct magnitude and proportion. The longitudinal steering forces can, thusly, produce controlled of the bogie itself.

When the conventional bogie assumes a parallelogram position, maximizing the angle of attack of outside leading wheel, the low lateral constraint of the wheel sets causes the conventional bogie to often block in this configuration by the forced-couple produced by the heavy flange contact drag and the draft force acting at the center plate. The complete curve is often negotiated in this condition, with resulting heavy wear rates on both wheels and rails. This condition is aggravated further as the bogie components wear, thus allowing parallel gramming to become more severe and the angle of attack to increase the steering bogie, of course, upon entering a curve has the same tendency to become laterally offset and develop the longitudinal steering forces. By virtue of the longitudinal freedom of the wheelsetsto assume a radial position under the control of the elastomeric or rubber shear pads and the steering arms, steering is initiated almost immediately. This prevents the initial flange contact or, when entering sharp curves, reduces the severity of the contact on the outer rail and allows the steering forces to build up gradually until the bogie rotational resistance about the centre plate is overcome and the bogie turns to the required degree. The interconnected steering arms ensure that both wheelsets are approximate radial compliance, thus utilizing all four shear pads to dampen and balance the steering and inertial forces. Even under flanging contrail sidings, the steering bogie is a vastly considerable improvement over the conventional bogie, since the steering arms maintain the bogie in its maximum radial configuration with a relatively low angle of attack. This is obviously far superior to the parallelogram the high angle of attack configuration the conventional bogie assumes even in minimum curves.

The technical advantages of the self-steering bogie can be of no real value to railways unless there is an economic justification. This economic justification, proven from tests in North America show the bogies using the GREGG Barber Scheffel principle have actually operated for distances up to 150 000 *kilometers* and more with negligible wear of wheel treads and flanges.

Wheel wear is of course the counterpart of rail wear. While the wheel wear saving of the self-steering bogie is substantial, the major advantage is in the reduction in wear of the rail itself. Where the wheel wear advantage can be considered as a factor of 1, the rail wear advantage is some 9 times that of the saving of wheel wear.

Furthermore, fuel savings through reduced rolling resistance on curves, is at least twice that of the wheel wear saving. Relative cost will, of course, vary from one railway to the next and depends on relative fuel costs as well. Further savings, which are difficult to quantify, will arise from reduced wear of bogie components, to say nothing of lading damage reduction, improved track alignment and gage maintenance costs and obvious advantage of increased wagon utilization from operating both empty and loaded trains at higher speeds.

The load-sensitive variable dampening of the GREGGBarber concept provides that amount of resonance dampening that automatically adjusted relative to the varying loads carried. Obviously the energy required to dampen a laden vehicle is greater than that for an empty wagon. More energy, that otherwise is necessary, is required to quickly dampen a deep oscillation that might cause springs to go solid with resultant track, lading, wagon and spring damage. Thus the essential requirement of flexibility to automatically meet the varying loads and operating conditions is provided. Recent tests which provided data for computer analysis of bounce simulation indicated that, for a 100-*Ton* capacity bogie, maximum dynamic centre plate load was only 57% of the dynamic load of a competitive constant-friction device. At the same speed, the maximum rail force of GREGGBarber bogie was only 69% of that of the competitive constant-friction device. To provide an acceptable ride quality over a load range from light to loaded wagon, it is necessary to have the total column force directly proportional to the square root of the sprung weight. For maximum effectiveness, the GREGGBarber resonance-dampening system dissipates energy by friction using these directly proportional column forces.

The advantages of the GREGG Barber self-steering bogie have been further enhanced by the addition of Hollube Wear Eliminator centre and side bearing wear plates, which wear plates have been proven to last under actual operating conditions for up to 497 000 *miles*with no noticeable wear of either the body or bolster centre bearing.

2. The rate of derailments are very high in SRC. The investigations for the causes of derailments and the procedures followed are not fruitful and has to be changed because knowing the causes of derailments will save a lot of money by promotion of maintenance, raising the standards of performance, reliability & safety.

An independent body (from outside SRC) has to be adopted to do the very important job of finding the actual cause of derailment &raise a report to the railway with suggestions to improve the maintenance of the vehicles & track.

A summary of derailment causes is presented here under.

1) Derailments [19]

This section provides a basic explanation of ‘key’ derailment causes that have the most influence on derailments, either by number or cost. More detailed research into these causes will be completed in subsequent D-Rail work packages. More information about causes can be found directly in the investigation reports available on the D-rail website.

a- Train Handling

Train handling derailments are generally associated with excessive longitudinal forces such as developed by run in (high compressive forces) or run out (high tensile forces). These in turn can generate high Y/Q forces which result in a wheel climbing the rail and initiating the derailment.

Improper braking can likewise result in high longitudinal forces introduced into the track. Contributing factors include (but are not limited to) high levels of longitudinal loading that can cause the couplers to break or pullapart (high tensile forces) or which can cause high Y/Q values (high compressive train forces, particularly in a curve where they generate high lateral Y forces), improper braking , etc.

b- Infrastructure specific derailment causes

Rail failures

Rail failure derailments are generally associated with the fracture of the rail under a vehicle, usually under conditions of high dynamic loading. These types of rail failures are associated with internal defects that grow and reduce the strength of the rail section, making them more susceptible to fracture. Contributing factors include (but are not limited to) high contact stresses, high bending stresses, inadequate rail support, loss of rail section (usually due to excessive wear), high thermal tensile stresses, etc.

Switch failures.

Switch failure derailments include failure of a key switch component, usually under load, wheel climb in or through the switch, “picking a switch point” (wheel flange entering between a closed switch point and stock rail), loose or missing switch parts of switch mechanisms, improperly located guards, etc. Contributing factors include (but are not limited to) high dynamic impact loads, excessive vehicle-track dynamic movement, high Y/Q (or simply high Y or low Q), loose or missing bolts, rods or other parts of the switch mechanism, points that are not fully closed, improper guard rail geometry, etc.

Improper Track Geometry (excessive width, excessive twist, etc).

Track geometry related derailments are generally dynamic vehicle/track interaction related where the defects in the geometry (either individual or in a harmonic series) generate excessive movement in a vehicle (which can be improperly loaded or improperly performing). These are usually associated with high Y/Q values (or simply high Y or low Q), but can be also related to large deformation of the track under high Y or Q loadings. Contributing factors include defects in the track geometry (either individual or in some type of harmonic series), inadequate fastener strength, improper vehicle loading, poorly performing vehicle, bogies or suspension, improper speed, etc.

Other track structure failure

Other track structure failures include failure of the rail fastener, sleeper, or substructure. These are usually associated with excessive dynamic loading. Contributing factors include high vertical (Q) and lateral (Y) loads, improper inspection and/or follow up maintenance, or in the case of track substructure improper drainage, excessive moisture.

c- Freight vehicle specific derailment causes

Wheel failures

Wheel related derailments are generally associated with the fracture of the wheel under a vehicle, usually under conditions of high dynamic loading. High temperature/overheating related failures are associated with the change in metallurgical properties of the wheel and the formation of thermally induced cracks. Contributing factors include (but are not limited to) high levels of loading (Y and Q) and associated high wheel stresses and stress related failure mechanisms (e.g. shelling), thermal overheating, excessive wear (thin flanges or worn tread), etc.

Axle Failures

Axle related derailments are generally associated with the fracture of the axle under a vehicle, usually under conditions of high dynamic loading. High temperature/overheating related failures are a major category of axle failures with the wheel bearings overheating and causing a "burn off" of the axle. Contributing factors include (but are not limited to) high levels of loading (Y and Q) such as due to poor suspensions or problems with the bogie frame, overheating of bearings, etc.

Spring and suspension failure

Spring and suspension failures are generally associated with improper loading or failure of the spring and suspension elements under load. They can include broken springs, failed elastomeric elements, etc. Contributing factors include high levels of dynamic loading (Y and Q), improper loading, improper inspections and/or maintenance, etc.

Journal rupture due to bearing damage

Bearing damage have many root causes such as overloading by dynamic effect due to wheel tread damage, current linkage, broken cage, metallurgical properties or mounting fault (choc, loss of fit…). Some derailments are due to wheel rupture (solid or tyre). In Europe tyre wheels are being removed from service over the coming years.

Type of wagon most prone to derailments

It seems that the types of tank wagons are mainly involved in derailment have two characteristics which differ from other wagons i.e. the centre of gravity height and great torsional stiffness.

d- Operation specific derailment causes

Wagon loading

Wagon loading derailments are generally associated with either improperly loaded wagons that generate undesirable dynamic behaviour or which generate excessive loading which can result in failure of a wagon structural component or a track component. Improper loading includes uneven loading (end to end or side to side) or simply excessive loading or overloading. Improper loading which results in high centre of gravity also results in very adverse dynamic behaviour that can result in a derailment. Contributing factors include (but are not limited to) high levels of dynamic loading (Y and

Q), to include unloading on one side (with consequently high Y/Q), excessive rocking of the wagon (increased susceptibility to track geometry faults such as excessive twist), high centre of gravity, improper welds or other structural components of the wagon which can fail due to overloading, etc.

Improperly fastener loads

Improperly fastened load derailments are usually associated with loads that are not adequately tied down, or fastened, or portions of the load that become loose and drag along the track, fouling switches or other track components and resulting in a derailment.

Human factor

The underlying cause of most derailments (even after what is identified as vehicle and infrastructure) is human factor.

GB human factor analysis

RSSB reviewed the 17 formal inquiry reports that DNV studied for their work submitted to the ERA, with the aim of establishing if there were any more underlying, contributory and causal factors relating to Human Factors and Operations. This review and a review of the top 10 analysis results from the UK Safety Management Information System (SMIS), highlighted underlying and contributory causes related to errors in procedures, planning, maintenance/inspection (train & track) and communication of information. These are the Operational and Human Factors issues that cause the incidents most frequently. The analysis highlights that, whilst technological solutions act as an aide in identifying when a derailment is more likely to happen, it does not address the underlying causes and only highlights the mitigating consequences. Whilst there is merit in developing new technological systems, it is also necessary to understand and address the root causes and the human failures associated with derailments.

e- Combined causes derailment data

Derailments have major and contributory causes that lead to accident. Again, many of these causes have underlying factors. In order to better understand these underlying factors it would have been necessary to review the individual investigation reports – many of which are confidential. To overcome this, it was considered appropriate to identify multi-cause accidents, from data sources available to the project and for which sufficient data was available.It was clear that gathering more detailed data was not going to be easy either due to the large amount of derailments or simply due to a lack of adequate data in some instances. However, access was eventually provided to some investigation reports and these will be used in other WPs to define root causes of derailments. The information from many derailments are presented in a single database, available on the project website and available to project members, but it is suggested that when looking at combined cause derailments a direct study of the investigation reports is made. Where possible original reports are uploaded resulting in a selection of reports available on the D-rail website from:

• ERA, GB and Austria, which are public

• USA, Russia which are not public, and may only be used by consortium partners. Publication of any of these results must be approved by with the report owner.

Some single cause derailments have a clearly identified cause, e.g. broken wheel or journal rupture due to bearing damage, etc. For combined cause derailments it is not always easy to clearly identify their causes. Many parameters could be identified, all within tolerance levels (sometimes at the limit) but it is the combination that leads to derailment. The probability for a vehicle to derail is not 0 if all the parameters are at the limit.

While most railroads and national agencies tend to report derailments as due to a single cause or to a primary and additional secondary causes, in actuality many derailments are combined-cause derailments, where the combination of several contributing factors are necessary for the derailment to occur. Thus, if one of these contributing causes was not present, the derailment would not have occurred. Thus, while sometimes reported as a secondary cause, or in many cases not included as a cause, these contributing causes are critical to the occurrence of the derailment and to its prevention.

Very typical of this class of combined cause derailments are those associated with track geometry defects. In many cases, key additional contributing factors to these types of defects are speed, often within a "critical speed range", non-uniform loading which can include under loading of one side or end and overloading of the other side or end, poorly performing bogies, and excessive wheel or rail wear, particularly when they form a shallow angle that makes it easier for a wheel to climb the rail in a curve. Likewise derailments in switches are very often combined cause derailments, with both the condition of the switch and the condition of the vehicle contributing to the derailment.

f- Shunting yards and sidings derailment data

The data available on marshalling yards is mainly from France (SNCF) and the UK (RSSB). SNCF and RSSB have made available brief analysis of this type of derailment occurring in their countries.

f.1- GB derailment data for yards, depots and sidings

Chapter 3.4 described the arrangements for the collection of safety data in the GB rail industry by means of the Safety Management Information System(SMIS). In addition to the mandatory requirement to record all mainline derailments, a number of derailment events which take place in yards, depots or sidings are also recorded in SMIS. However, as this is not a mandatory requirement, the data can only be considered indicative of a probable larger number of derailment events which take place in yards, depots and sidings but are not recorded. It is considered likely that the yards, depots and sidings events recorded in SMIS are the more significant ones, but this distinction is not formally defined and is dependent on the voluntary use and interpretation by various train operating organisations. It is thought that the majority of derailments in yards, depots and sidings occur at low speed and cause little damage, or have low potential consequence.

The Table (5.1) indicates the top 5 causes of derailment in GB yards, depots and sidings, based on the limited data available for the 6 year data period. This is based on the D-Rail categorisations of derailment causes. Where an event has more than one recorded cause a proportion has been used to split the record accordingly.

Table (5.1) GB top 5 causes of derailments in yards, depots and sidings, ranked by frequency

Case Area D-Rail Cause	Category	Number of derailments in 6 year period (Jan 05-Dec 10)
Operations	Human factors	25.1
Infratructure	c. Switch component structural failure	5.5
Infratructure	e. Exessive track width	4.0
Operations	c. Insufficient fastening of load	3.0
Infratructure	a. Exessive track twist	2.5
	Other causes	25.9
	Total	66.0

As in the data for mainline derailments, Human Factors is the dominant cause of derailment in yards, depots and sidings. The top 5 causes represent over 60% of the yards, depots and sidings derailments. All of the top 5 yards, depots and sidings derailment causes appear in the top 5 mainline derailment causes (ranked by frequency) apart from "c.

Insufficient fastening of load". This would suggest that the profile of derailment causes in yards, depots and sidings is broadly compatible with the profile suggested by mainline derailment data.

f.2- French data for shunting yards

SNCF had reported 201 marshalling yard derailments for our investigation period 2005-2010. They looked at causes of these derailments and concluded that:

• 92 % of marshalling yard derailments are due to operational error (human factor) or condition (backing movement);

• 5.3% are related to Infrastructure problem, mostly points and switches;

• And 2.7 % are related to rolling stock problem.

f.3- Conclusion about shunting yards, sidings and depots derailment data

These types of derailments are often not reported as many counties do not have alegal requirement to report them. Although they occur frequently, e.g. France had 201 derailments in 6 years, costs per derailment are low, so for France is estimated at only 1% of the cost of main line derailment. The main cause for these derailments is considered operational and mainly a result of human error. It is not recommended that the D-Rail project focuses any further, in its research, on these types of derailments.

2) Measuring and monitoring techniques

Current monitoring technologies and techniques in relation to both freight vehicles and infrastructure (wayside) are included in the database, so each partner country is able to determine which causal effects are currently being monitored. Comparisons for each cause frequency and the monitoring technique for it, by country is available and to assist in establishing the effectiveness of these technologies in relation to freight operation and safety.

From available data we can conclude that most countries use monitoring techniques as: recording cars, track and vehicle-based, maintenance yard inspection methods and checks, and also visual inspection.

Causes and how they are monitored are presented in following Table (5.2):

Table (5.2) Derailment causes and technologies, according to D-rail database.

Cause	Technology	How it is used
Hot axle box	Yes	Widely used
Track geometry	Yes	Widely used
Rail failures	Yes	Widely used
Improrer loading of wagon	Yes	Widely used
Brakes	NO	Widely used – Mainteance checks
Switch	Yes	Not much used
High speed	Yes	Not much used
Spring & suspension failure	No	Not much used – visual check
Wheel break	No	Widely used –

		visual check
Axles	Yes	Widely used – Maintenance check

This is a preliminary analysis from supplied data. A more detailed review of monitoring technologies will be completed in WP4.

3) Conclusions

A result of WP1 has been the creation of a database containing freight train derailment information from USA, Russia, GB, Austria, France, UIC and ERA

(DNV) for the period of 1.1.2005 -31.12.2010. Difficulty was experienced collecting the data to present in a single format due to the differences in the variety of data reporting criteria, consistency of reporting, structure of individual databases, classification of causes, etc. It was agreed to use the DNV classification of causes with slight modifications and concentrate on the summary number of derailments per year and costs, location of derailment (main line or shunting yard), and separation of single and multicauses derailments. Derailments were therefore categorized into the following groups:

1. Derailments caused by Infrastructure failures

2. Derailments caused by Rolling Stock failures

3. Derailments caused by Operation failures

4. Derailments caused by Weather, Environment and 3rd Party

5. Unspecified

88% of derailments were successfully categorized into one of these four groups. It was easier to categorise derailments into main groups than into subgroups. Initial analysis for world regions: USA, Russia and Europe was followed by individual European country analysis to help determine the main derailment causes in Europe. Within the world regions, Infrastructurecauses are responsible for 40% of derailments, followed by rolling stock (33 %), and operations (25%). The spread between countries is sometimes huge due to differences in operation, track, rolling stock, etc. For example,

USA's dominant cause is infrastructure at 50%. Track geometry is much more dominant in Europe as a whole (UIC and DNV) and other presented European countries, compared with USA and Russia, where the number of track causes is higher, and where it is the dominant cause is rail failures in USA. Within the infrastructure sub categories the following four causes are dominant in Europe: excessive track width, track height / cant failure, rail failures, switch component structural failure, excessive track twist.

Rolling stock derailment causes were more difficult to classify, although the following four groups could be identified: hot axle box and axle journal rupture, failure of bogie structure and supports, spring and suspension failure, wheel rupture. GB have had a higher percentage of derailments as a result of brake failings, the majority of which have been due to handbrakes left applied to wagons when moving on the mainline.

The most difficult category to classify was operations, where how to include the 'human factor' element presented a constant dilemma. Other dominant causes include: wagon wrongly loaded, point switched to wrong position, other mishandling of train including driver caused SPAD, brake shoe or other object left under train.

So overall the ranking of major derailment causes in Europe is:

1. Axle ruptures

2. Excessive track width

3. Wheel failure

4. Skew loading

5. Excessive track twist

6. Track height/cant failure

7. Rail failures

8. Spring & suspension failure

Most railroads and national agencies tend to report derailments as due to a single cause or to a primary and additional secondary causes. In reality many derailments are the result of combined causes (where the combination of several contributing factors are necessary), for the derailment to occur. Very typical of this class of combined cause derailments are those associated with track geometry defects. In many cases, key additional contributing factors to these types of defects are speed, often within a "critical speed range", non-uniform loading- which can include under loading of one side or end and overloading of the other side or end-, poorly performing bogies, and excessive wheel or rail wear, particularly when they form a shallow angle that makes it easier for a wheel to climb the rail in a curve. Likewise derailments in switches are very often combined cause derailments, with both the condition of the switch and the condition of the vehicle contributing to the derailment.

Analysis of shunting yard derailments, where costs of derailment are comparatively much lower, showed the main cause to be operational, with the 'human factor' as a significant contributor. It is not recommended that subsequent WP studies focus on this area any further. The main causes of derailments in shunting yards (data from France and GB) and mainlines (Europe) is presented in Table (5.3). Where derailment is noted as occurring in shunting yards, other causes are generally not noted – hence other unspecified causes are excluded from the table below.

Table (5.3) Comparison between mainline and shunting yard derailment causes

Mainline causes	Shunting yard cause
Axle ruptures	Human factors
Excessive track width	Backing movement
Wheel failure	Point switched to new postion
Skew loading	Darg shoe
Excessive track width	Switch component structural failure
Track height/cant failure	Excessive track width
Rail failures	Insufficient fastening of load
Spring & suspension failure	Excessive track twist

These two categories are completely different, therefore the recommendation to analyse them separately. In shunting yards, operations and human factors are dominant. When considering the technical causes, track width and twist and loading appear in both categorisations.

Based on this analysis of derailment statistics, we can conclude that: developing new technologies and improving existing ones to aid the detection of major causes, improved planning and optimisation of inspections, where greater risk causes are tackled first, would result in fewer derailments.

4) Recommendations

Based on the analysis of the derailment causes, it can be seen there is a commonality of causes and mechanisms between all of the reporting groups which include Europe, US and Russia (with some exceptions as noted earlier). These mechanisms are very often related to clear and measurable parameters, which either directly or indirectly contributes to the derailment mechanism and/or cause. Many of these parameters are directly measurable, so that by identifying the derailment mechanism and its key contributing parameters it is possible to develop measurable parameters and/or conditions - the elimination of which would significantly reduce the risk of derailments.

These parameters can be directly related to the derailment cause, such as axle box temperature and its associated axle box burn-off derailment or indirectly related such as vehicle/track dynamic related derailments which often can have multiple contributing factors but generally have to have a high Y/Q ratio for the derailment to occur.

By examining those categories of derailments that represent the most common derailment causes (e.g. the top 5 or top 10 causes), and examining in detail the mechanisms associated with these derailments, it may be possible to identify key parameters that are both directly related to the derailment condition and that are directly measurable, either via wayside or on-board measuring devices or via an inspection vehicle. By focusing on such key contributing parameters for the most common derailments and identifying both the parameter, its critical (or high risk) threshold, and a reliable and accurate measurement technology, it is possible to identify current or near term future technologies and measurement techniques that can be used to reduce the risk of these derailments. Furthermore, by understanding the fundamental mechanisms and the key influencing parameters, it may be possible to redeploy or modify existing technologies to more effectively reduce the risk of these derailments.

5.3 Recommendation

SRC in its specifications for locomotives request from bidders to give estimated fuel consumptions. For speed-time and trailing loads pulled by their offered locomotive the trips & loads specified are:

1) [1350 tons-1500 tons] Port Sudan – Summit.
2) [1500 tons] Summit- Atbara – Khartoum.
3) [1500 tons] Khartoum- Atbara- Summit- PortSudan.

We present here such requested information presented by one of the bidders with his offer.

SRC usually make test runs to the locomotive during its one year guarantee period to make sure of the figures of the fuel consumption & the timings & the speeds when tested with the specified loads in the specified sections.

Our recommendation to SRC or any colleagues who may take this recommendation as graduation project is to do these tests for:

a) Locomotives powered by medium speed diesel engines-(850-1100 rpm).
b) Locomotives SRC, SDD_1& SDD_2 (Chinese locomotives powered by cat3500 B high speed diesel engine (1800 r.p.m).

The offered locomotive performance of the said bidder fuel/time results are presented here under.

Locomotive performance

Train Resistance

The values of train resistance used in the performance calculations are based on the Davis formula and goods wagons having the following dimensions.

Tare	--	17 long tons
Carrying capacity	--	35 long tons
Overall length	--	41 ft. 9 ins.

Height above floor level	--	8 ft.
Overall height above rail level	--	11 ft. 6 ins.
Width	--	8 ft. 3 ins.

Using the general Davis formula:

$$R=1.45+\frac{29}{W}+.050\ V+\frac{.0005}{WN}AV^2$$

When R = Ibs/ton

W= Axle load – 13 tons

V = Mile /h

N = Number for axles = 4

A = head Area = 79 $ft.^2$

the formula thus becomes R = 3.68+ .050 v + 0.00076 V^2

The locomotive resistance has been calculated according to the Davisformula:-R= $1.45+\frac{29}{W}+.034\ V+\frac{.0024}{WN}AV^2$Ibs.f/ton.

For a 6 axle 99 ton locomotive of head area 110 $ft.^2$ this becomes:-

$$R = 3.21 + .03\ V + .00267\ V^2 \text{ Ibs.f /ton.}$$

Locomotive characteristics:

The Tractive Effort /Speed characteristic of the locomotive at maximum output is shown on curve No.8945T.

The maximum Tractive Effort developed is 70,000 Ibs.

The locomotive is designed for a maximum speed of 80 km/hr.

The continues tractive effort developed is 50,000 Ibs at 22 kph

The above characteristics are based on the engine developing 2400 h.p. at site at 900 r.p.m. which, after allowing 210 h.p. for the load of the auxiliary machines,gives an input of 2190h.p. for traction to the generator.

Trailing Load /Balancing Speed:

The load hauling capacity /speed on varying gradients is shown on curve No.1867747.

The above curve is based on the following parameters.

1. Axle load $16\frac{1}{2}$ tons per clause 3 of your specification.
2. Compensated curves per clause 5 of your specification.
3. Rolling stock details and \Train Resistance (Davis formula) as per Train Resistance Section and assuming the rolling stock brakes to be in good condition without drag.
4. Low weight transfer bogie with unloading characteristic on the axles of the leading bogie of 17% T on each axle.

Dynamic Brake Characteristic:

The braking effort/speed curve for the dynamic brake is shown on curve No.1856405.

Required Performance:

Operating on the Port Sudan /Summit line which has ruling grade of 1 in125 a locomotive with a trailing load of 1350 long tons will have a balancing speed of 37 km/hr, on ruling grade of 1 in 100 the train will have a balancing speed of 28 km/hr.

A train with a trailing load of 1500 tons will balance at speed of 26 km/hr on a 1 in 100 grade.

The foregoing calculations are entirely dependent on the train resistance being in strict accordance with particular Davis formula.

Calculated times for the train are given in the following tables together with the estimated fuel consumptions:-

1350 tons) Port Sudan – Summit

1500 tons)

1500 tons) Summit – Atbara – Khartoum

1500 tons) Khartoum – Atbara – Summit – Port Sudan

The fuel consumptions for the recorded runs don't allow for periods of idling at stations and the performance figures are subject at 5% tolerance. It has been assumed that 8.4 Ibs of diesel fuel is equivalent to 1 gallon.

The performance stated in the attached tables is based on the following parameters:-

1. Site rating – 2400 H.P. (2190 HP for Traction)
2. Stopping at every station but excluding stopping time.
3. Observing a speed restriction of 20 km/hr for 600 metres before each station.
4. Maximum speed of 60 km/hr.
5. Trailing load of 1350 and 1500 tons.

6. Train resistance as per Davis formula stated in the train resistance section.
7. Braking rate – 0.8 km/hr/s.
8. Locomotive weight – 99 tons.
9. Assuming all out performance but working to the relevant speed restrictions.
10. Assuming locomotive and train brakes not causing drag after lease.

For the speed – time – diesel fuel consumption tables we have used the maximum Tractive Effort available in the equipment for the computation. High a adhesion levels are needed to achieve these timings.

We would wish to discuss these aspects appertaining to your system and applicable to our offer.

Note: the profile supplied with the tender was incomplete over the section Khartoum north to 250 km post on the Khartoum to Port Sudan line diagram. For these runs we have used old profile data stored in our records. If the above profile data is supplied to us we will be pleased to re-calculate the performance figures for that section.

Speed – Timing Port Sudan – Summit (Distance 129 km)

Trailing load 1,350 tons

2400 H.P. Co – Co Locomotive

	Time (mins)	**Fuel (galls)**
Port Sudan	0	0

Astoriba	17.59	28.5
Tesyamti	22.28	36.4
Sallom St.	11.17	16.4
Edraweib	22.07	37.1
Obo	16.35	26.7
Ekwat	23.65	39.6
Komosana	16.62	27.0
Assut	20.65	34.9
Erebba	25.53	42.9
Gamateit	15.14	24.3
Gabeit St.	16.94	27.1
Edate	13.22	20.8
Sinkat	12.01	17.7
Summit	14.33	22.7

Total time **247.54** **Total fuel** **401.9 galls**

Average Running Speed 31.3 km/hr.

Speed – Timing Port Sudan – Summit (Distance 129 km)

Trailing load 1500 tons

2400 H.P. Co – Co Locomotive

Time (mins) **Fuel (galls)**

Port Sudan	0	0
Astoriba	18.8	30.9
Tesyamti	24.0	39.5
Sallom St.	11.69	17.5
Edraweib	23.88	40.5
Obo	17.64	29.1
Ekwat	25.53	43.1
Komosana	26.37	29.4
Assut	22.22	37.9
Erebba	27.64	46.8
Gamateit	16.23	26.4
Gabeit St.	18.21	29.5
Edate	13.95	22.3
Sinkat	12.75	19.0
Summit	15.13	24.3

Total time **265.59 mins** **Total fuel** **436.3 galls**

Average Running Speed 29.1 km/hr

Speed – Timing Summit – Khartoum (Distance 658 km)

Trailing load 1500 tons

2400 H.P. Co – Co locomotive

	Time (mins)	**Fuel (galls)**
Summit	0	0
Shakan	14.24	7.4
Barameyu St.	10.25	6.6
Barasait	14.83	8.0
Erheib	10.89	6.8
Thaymum	21.57	10.3
Haiya St.	19.02	10.5
Einha	14.55	8.9
Kass	17.69	13.2
Shediyeb	12.55	9.8
Tolghari	19.24	13.6
Miass	18.22	15.8
Musmur St.	18.03	16.5
Sigodeit	12.95	11.9
Rogal	17.71	16.6
Togni	25.82	25.5
Seteib	24.15	18.0
Ogrein	22.26	21.9
Hadika	25.77	25.3
Dugaya	25.15	24.2

Hudi	27.55	30.2
Zullot St.	18.19	19.5
Atbara St.	19.38	18.4

Subtotal410 mins subtotal 338.9 galls.

Eldamer	16.81	17.0
Zeidab	22.92	28.2
Aliab	21.48	25.6
Mahmya	25.47	29.6
Um Ali	14.42	16.8
Kaboushiya	22.28	26.5
Tragma	23.35	28.3
Shendi	20.22	23.9
El Goz	22.77	27.0
Wad Ben Naya	23.71	27.8
El Meiga St.	28.38	42.1
Gebel Gerri St.	31.09	38.7
Royan St.	23.38	24.1
Geili St.	18.97	18.7
Gubba St.	16.44	18.6

Kadaru St.	18.96		22.7
Khartoum North	17.50		20.9
Khartoum Central	7.76		9.3
Sub total	375.92		445.7
Total time	**785.92**	**Total fuel**	**1784.6**

Average Running Speed 50.2 km/hr.

Trailing load 1500 tons

2400 H.P. Co – Co locomotive

Speed – Timing Khartoum – Port Sudan (Distance 787 km)

	Time (mins)	**Fuel (galls.)**
Khartoum Central	0	0
Khartoum North	7.78	9.4
Kadaru St.	17.31	19.9
Gubba St.	18.98	21.5
Geili St.	16.43	19.8
Royan St.	19.36	26.6
Gebel Gerri St.	23.77	31.7
El Meiga St.	28.51	32.3
Wad Ben Naga	27.49	24.6
El Goz	23.75	27.6
Shendi	22.85	26.3
Tragma	20.09	23.1
Kaboushiya	23.44	26.1
Um Ali	22.27	25.3
Mahmya	14.47	17.5

Aliab	25.33	29.3
Zeidab	21.46	24.6
Eldamer	22.80	25.7
Atbara St.	17.03	23.7

Subtotal 373.12 mins **Subtotal 435.1 435.1 gall.**

Zullot St.	20.52	28.6
Hudi	18.54	24.2
Dugaya	27.72	33.9
Hadika	25.6	34.6
Ogrein	26.10	35.2
Seteib	22.58	29.8
Togni	25.49	38.8
Rogal	25.80	34.7
Sigodeit	18.40	25.7
Musmar St.	13.49	19.0
Miass	19.03	27.4

Tolghari	19.41	28.3
Shediyeb	20.64	32.4
Kass	14.04	21.4
Einha	19.27	30.3
Hayia St.	16.97	26.6
Thaymum	22.95	36.8
Erheib	27.14	44.9
Brasait	13.70	20.9
Barameyu St.	19.14	30.8
Shaken	14.0	21.8
Summit	17.92	31.9
	Subtotal 448.48 mins	**Subtotal 657.9 galls.**

Sinkat	12.66	7.5
Edate	9.07	5.8
Gabeit St.	10.25	5.3
Gamateit	11.84	6.0

Erebba	10.22	5.0
Assut	14.99	6.3
Komosana	12.47	5.0
Ekwat	10.57	5.1
Obo	13.77	5.9
Edraweib	9.68	4.7
Sulloom St.	12.63	5.2
Tesyamti	9.72	6.7
Asotriba	14.62	6.5
Port Sudan	13.37	7.0

Subtotal 165.87 mins **Subtotal 82.1 galls.**

Total Time 987.47 mins **Total fuel 1175.1 galls.**

Averege Running Speed 47.8 km/hr

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Appendix (Y)

Summary & conclusion on other medium speed diesel engine

VS

the high speed diesel engine

Appendix Y

Summary & conclusion

1. Introduction:

Medium & high speed diesel engines are in rail traction with medium speed engines dominating the field specially the field of main line locomotives for freight traffic.

The high speed diesel engines are used in shunting &switching locomotives. Also they are used in rail diesel cars (DMU) for suburban passenger trains & may be for inter-city passenger trains.

High speed in rail traffic, are many causes, of great commercial value and, therefore, the maximum power must be installed in the locomotive whilst still retaining light axle loads. To quote example, the maximum permitted axel load on British railway is 25 tones, but for high speed train (HST), (inter-city passenger train, which travels at speed of up 200 km/h), the maximum permitted axle load is restricted to 17.5 tones, in order to keep track stresses within acceptable limits.

The need recognize the severity of the traction duty cycle cannot be over emphasized. The traction mode of operations involves many hours at idling which can produce oil carry over, carbon build up, low temperature corrosion, and the other problems with fuel injection equipment.

In assessing methods of acceptance testing of new designs of diesel engines for rail traction duties, both the mechanical (high cycle) and thermal (low cycle) fatigue regimes have be considered. The cycle life of components depends upon the magnitude and type of strain and it seems that strain in the most significant factor governing the cycle failure mechanism.

However, the behavior which dictates the actual magnitude and type strain encountered is strongly influenced by the imposed temperature stress and engine cycling loads and hold times.

To this end, it is clear that A RAIL TRACTION ENGINE is an engine which has been developed SPECIALLY for this APPLICATION.

This project displayed, the effort made by three famous locomotives manufacturers to develop their engines for TRACTION.

For reasons, to be shown, the three started by medium speed 2-stroke & 4-stroke and HIGH rotational speed engines. Each has this inherited problems linked to his choice of design.

They started developing these engines to suit the traction duties.

Other engines which had not undergone such research & development work are still with their inherited problems.

The reasons which dictated on the three manufacturers are shown under:

1. Historical, general motor has pioneered the medium speed two-stroke cycle diesel engine, although they are not irrevocably committed to any specific type of engine. In the early thirties research at EMD/GMC had led to an eight cylinder two-stroke cycle diesel engine design which could be built with rating of 600 horsepower. The best 4-cycle, 8 cylinder of the same bore and stroke as EMD's 2-cycle design was then rated 400 H.P., SO it appeared that their 2-cycle engine had affair edge over the 4-cycle and then produced the 8-cylinder two-stroke cycle. The general philosophy of EMD/GMC is to solve problems related to 2-stroke cycle design. The development work for the enhancement of reliability, durability and engine performance. Lately a conservative attempted to update Series 645 engines was made (645FB).
2. The Ruston philosophy is philosophy of uprating. Development work is for solving the problems related to higher thermal and mechanical loading due to uprating and enhances reliability and durability. Redesigning for ease of maintenance was also looked at. The horsepower which and been available from 16 cylinder version (RK3) was extracted from a 12 cylinder version with the same bore and stroke (PKC). It now

possible to see 8 cylinder versions replacing the 12 cylinders at the same horsepower.

3. Maybach Motorenbau of Germany pioneered the design & manufacture of high speed, light weight and small size gasoline and diesel engine for airships. They found themselves bound to move continuously, extensively & expensively into research and development work to introduce their HIGH SPEED ENGINE in RAIL TRACTION.
4. CAT. Series 3500B engine shows:
    a. 1980s start electronic engine design.
    b. Later 1980s start 3500bB electronic engine production including 8, 12, 16 cylinder.
    c. Since 1995s 3500B engine start volume manufacturing.
    d. In 1998 first 3500Blocomotive engine sold in china.

1. some users, consultants & manufactures viewpoint:

1.1. British Rail:

First cost:

First cost has to be seriously considered at the purchasing stage, but it is of secondarily importance to technical assessments associated with maintenance and operating costs when taken over the twenty-five year life of the engine. The first cost less than 5% of the fuel life operation and maintenance costs, and inflation and the increase in fuel costs are taken into account, it becomes almost insignificant. In practice however, first cost can be decisive if engines of similar design and operating performance are offered at a significant in price.

Overhaul periodicity:

In cutting improved reliability as a major user's requirement, it's essential that this leads to an increase in the time between workshop overhaul periods, and hence reduced maintenance costs.

Consider the existing 2984 DMU engines which at the present time do not operate on preventative maintenance basis, but are overhaul when they fail.

The average of these engines appalling, at approximately 161 000 km (100 000 miles), 4000 engine hours. The major failures being pistons and bearings, which account for roughly 60% of total of failures. The causes of these failures are generally inadequate cooling, poor air filtration, poor lubricating oil filtration and low lubricating oil pressure. In specifying new engine, a minimum overhaul period of 450 000 km (280 000miles) will be required and a preventative maintenance schedule will be introduced to further reduce costs of expensive engine failures.

Locomotive maintenance costs are approximately ten fold these of the DMU engine and carefully specified maintenance and overhaul schedule is laid down. Depot maintenance procedures are fairly common between the various types of engine, but the overhaul periods dictated by the first component to reach its life or wear limit. This could be piston ring wear, piston ring groove wear, bearings etc. a further factor is duty cycle. Passenger train cycles for instance, are more severe than freight. Overhaul periods are therefore, determined by experience in service and the current periods. British Rail operated both medium speed and high speed engines and experience has shown that the medium speed engines can achieve an overhaul period of approximately 1500 hours.

The HST however, has to be considered in a different light, as it undergoes a much more arduous serves duty cycle, as shown in fig 4-3 the figure illustrated a typical locomotive hauled passenger train and an HST from which it can be seen that the HST engine experiences 8.2 load to idling thermal cycles per hour compared to 4.9 of a typical passenger train. The present overhaul cycle on the engine is 7000 hours.

The user, in looking to the future, must aim at a significant improvement in these figures and on a medium speed engine it should be possible to extend the overhaul period to 20000 hours to a top overhaul, that is piston removal, and 40000 hours to a major overhaul, that is crankshaft removal. A better alternative would be 30000 hours to a major overhaul with no intermediate repair.

The second alternative is favored, as the first option can lead to dirt ingress to the main bearing system during a top overhaul. If a 30000 hour overhaul period is achieved, it would reduce the annual engine maintenance cost by 25 to 30%.

A similar argument should apply to the HST engine and a 15000 hour period should be possible, which will give a similar reduction in costs.

In looking to these extended periods, it will be necessary to introduce Depot schedules for the changing of fuel injection equipment and turbochargers at intermediate periods.

It is worth noting that the French Railway have already achieved the above overhaul periods with their 67000 type locomotive and overhaul the 1500 rpm engine after 775 000 km (481 000 miles) which is equivalent to 24000 hours.

The engine of course, cannot be treated in isolation and cooler groups, transmission systems and all other equipment has carefully matched to ensure it gives the same performance.[21]

<u>Railway & fuel consumption:</u>

Traditionally BR have specified medium speed engines for all main line locomotives and few deviation to 1500 rpm engines have generally resulted in the engines of poor reliability. Together with the higher fuel consumption of the 1500 rpm engine.[21]

<u>Maybach Motorenbau (High Speed engine):</u>

Opinion at the time-which may still hold today – was that the high-speed engines despite their obvious advantages with the respect to weight and reduced space requirements would have shorter service life, have less favorable consumption figures, and be more prone to trouble. This opinion chiefly resulted from the fact the many firms tried to increase the output of their existent low speed engines merely by increasing the engine speed and neglecting any improvements in design. Naturally, this led to setbacks which in turn gave rise to such widespread opinion. The MAYBACH MOTORENBAU, which even then had many years of experience in the field of the high-speed Otto engines, a adopted new methods by designing a diesel engine especially for high speed and by aiming simultaneously at a corresponding increase of operation reliability and serves life, two requirements obviously necessary for airship engines.[22]

<u>The Henderson partnership (Consulting Engineering's</u>):

In the past many third world countries have purchased locomotives, which have proved less satisfactory that when running in developed countries, and which have shown their poor reliability when maintenance has not reached a sufficiently high standard.

For third world countries for many years in the future, there are advantages in having engines less sophisticated and less sensitive, but with a high degree of reliability.[23]

<u>CAT. Electro-Motive Diesel (EMD):</u>

<u>Idling speed</u>:

The idle speed of the model F3A engine is 200 rev/min the lowest idle speed of any model 645 engine to date this 200 rev/min idle speed reduces engine fuel consumption at approximately 15 per cent when compared with the 235 rev/min idle speed of the model F engine.

Prior to adoption of the 200 rev/min idle speed studies had indicated the engine performance at this very low idle speed is characteriesed by satisfactory combustion, invisible exhaust, and freedom from lubricating oil dilution.

<u>NB</u> the idle speed of cat. 3516B engine is 600 rev/min.[24]

<u>GanzMavag- Budapest:</u>

<u>Diesel engine:</u>

Type Ganz-Mavag S.E.M.T.-pielslich PAV-185.

Maintenance & repair instruction:

The schedules of maintenance in the instruction are given in engine revs. Which indicated that a high speed engine has shorter between overhauls than medium speed engine?

From all testimonies above and the content of the project we conclude the Chinese locomotives with their undeveloped engines for locomotives are inferior than any of SRC locomotives purchased to SRC specs.

There are no technical or economic grounds for the introduction of locomotives with high speed engines in SRC because:[24]

- Reliability.
- Durability.
- Fuel economy.
- Overhaul periodicity, etc…

Shall be inferior that of locomotives with medium speed traction engines purchased by SRC according to rules & regulations.

Appendix (Z)

1. A short description of the principles of wheel/rail interface

2. Vehicle &train dynamics.

3. Derailments.

4. A power point presentation of train track dynamics.

Appendix Z

1) The wheel on the real:

Rail way wheels sit on the rails without guidance except for the shape of the tyre in relation to the rail head contrary to popular belief, the flanges should not touch the rails.

Flanges are only a last resort to prevent the wheels becoming derailed a safety feature.

The wheel tyre is coned and the rail head slightly curved as shown in the following diagram Fig.1. The degree of coning is set by the railway company and it varies from place to place. In the UK the angle is set at 1 in 20(1/20 or 0.05). In France it is at 1/40.

The angle can wear to as little as 1 in 1.25 before the wheel is reprofiled.

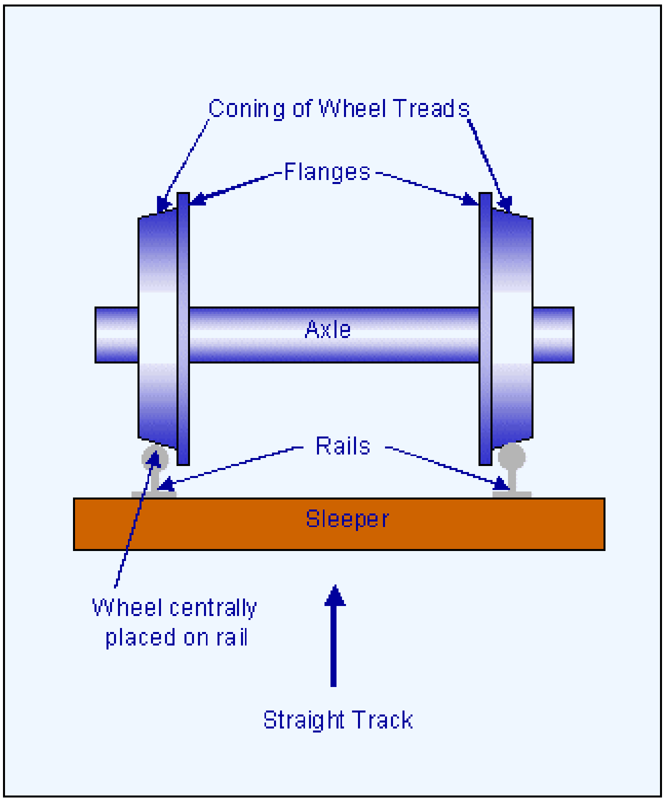


Fig 4.1: The shape and location of wheels and rails on straight track.

This diagram is exaggerated to show the principal of the wheel/rail interface on straight track note that the flanges do not normally touch the rails.

On curved track, outer wheel has a greater distance to travel than the inner wheel.

To compensate for this, the wheelset moves sideways in relation to the track so that the larger tyre radius on the inner edge of the wheel is used on the outer rail of the curve, as shown in Fig.2

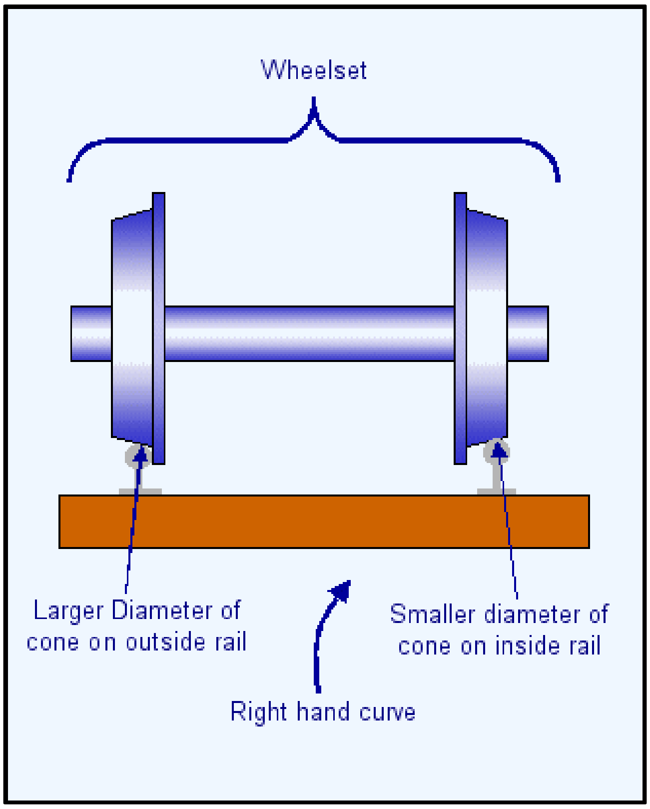


Fig.2: The location of the wheels in relation to the rails on curved track

The inner wheel uses the outer edge of its tyre to reduce the travelled distance during the passage round the curve. The flange of the outer wheel will only touch the movement of the train round the curved rail is not in exact symmetry with the geometry of the track this can occur duo to in correct speed or poor mechanical condition of the track or train. It often causes a squealing noise. It naturally causes wear.

Many operators use flange or rail greasing to ease the passage of wheels on curves.

Devices can be mounted on the track or the train. It is important to ensure that the amount of lubricant applied is exactly right too much will cause the tyre to become contaminated and will lead to skidding and flatted wheels.

There will always be some slippage between the wheel and rail on curves but this will be minimized if the track and wheel are both constructed and maintained to the correct standards.

Bogies (Trucks):

A pair of train wheels is rigidly fixed to an axle to from a wheelset .normally, two wheel sets are mounted in a bogie , or truck as it is called in US English . most bogies have rigid frames as frames as shown below (Fig 4.) .

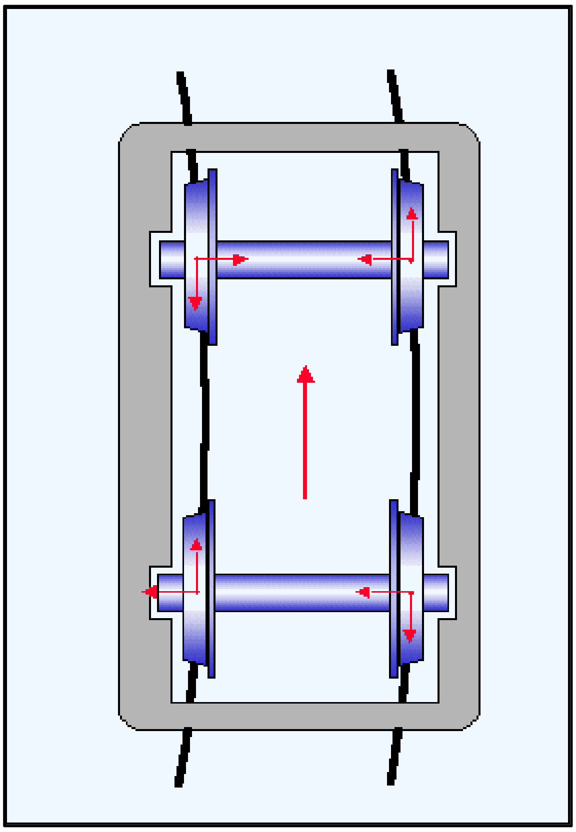


Fig.3: A standard rigid bogie on curved track.

The bogie frame is turned into the curve by the leading wheel set as it is guided by the rails however is a degree of slip and a lot of force required to allow the change of direction the bogie is, after all, carrying about half the weight of the vehicle it supports.

It is also guiding the vehicle, sometimes at high speed, into a curve against its natural tendency to travel in a straight line.

Steerable bogies:

To overcome some of the mechanical problems of the rigid wheel set mounted in a rigid bogie frame, some modern designs incorporate a form of radial movement in the wheel set as shown below (Fig 4.)

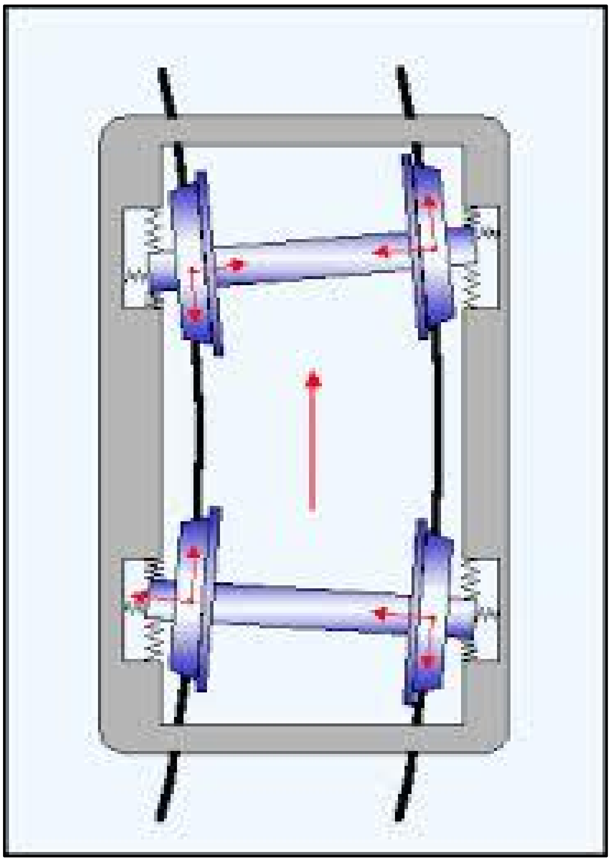


Fig .4 A bogie on curved track with radially steering wheelsets.

In this example, the wheelset ‘’floats’’ within the rigid bogie frame the forces wearing the tyres and flanges are reduced as are the stresses on the bogie frame itself. There are some designs where the bogie frame is not rigid and the steering is through mechanical links between the leading and trailing wheelset.

2) Vehicle and train dynamics

Vehicle oscillations

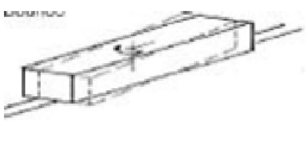
The motion of any kind of vehicle at speed along the track causes various movements and oscillations, some of which are undesirable. Their amplitude and effect depend on speed, mechanical conditions, suspension design, vehicle geometry, height of centre of gravity and the condition of the track.

These oscillations arise from several causes, mostly originating from track irregularities, and are accentuated through wear, breakage and excessive clearances on the parts of vehicles, such as axle end-play in axle-boxes, clearances between axle-boxes and frame, broken springs, etc.

Nosing or yawing

These are transverse or lateral out of phase rotational movements of the front and rear of the vehicle about a vertical axis through the centre of gravity of the vehicle.

Vehicle nosing or yawing can induce heavy side thrusts on the rail head by lateral impacts, resulting in a variation of wheel loading to such an extent that the wheel flange will be forced over the rail by the heavy side thrust acting coincidentally with a reduction in vertical wheel loading.



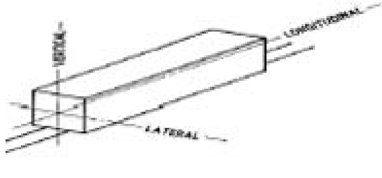


Figure - Nosing or Yaw Harmonic Roll

This is a transverse rotational oscillation of the vehicle body at its centre of gravity about a longitudinal centre line located at the level of the suspension system.

Roll is excited by irregularities in superelevation or versine. Wagon weight is shifted alternately from one rail to the other. The roll will normally be accompanied by lateral movement of the wagon centre of gravity (sway).

In most cases, vehicle oscillation tends to be minimal at speeds less than 50 km/h. The main exception to this is low speed "rock and roll" motion, which is primarily caused by cyclic track irregularities coupled with the natural roll frequency of loaded vehicles. This severe response occurs in a narrow speed band of about 3 km/h either side of the resonant speed. At speeds around 30 km/h, quite violent "rock and roll" motion can result in some cases actually lifting a wheel clear of the rail. On straight track, this motion is unlikely to cause derailments as there is no simultaneous lateral flange force when the vertical wheel load is reduced. On curved track, however, "rock and roll" motion is highly undesirable, as when the vertical load is reduced on the high rail, there is a simultaneous lateral flange force occurring which can cause the wheel to derail.

As train speed increases above 50 km/h on any piece of track, vehicle oscillations become more and more pronounced with corresponding increases in rail loadings. Some vehicles and bogie combinations can have critical speeds in the 70-80 km/h speed range.

These are generally high centre of gravity vehicles which roll badly under certain track conditions.

Harmonic roll is clearly a complex phenomenon involving many factors, particularly condition of track, design of wagon, spring static deflection, height of centre of gravity, plus loading and operating speed. A susceptible wagon must be operated at its resonant speed over track with consecutive cross level defects. A wagon will not roll harmonically if these factors are missing.

Figure - Roll

If harmonic roll is suspected as a cause use the following checklist to help ensure important facts are learned:

Derailment Situation

- train was travelling between 20 & 35 km/h on equal stagger jointed rail track or welded rail track with several consecutive low spots,
- first vehicle off was a heavy, high centre-of-gravity wagon,
- first vehicle derailed to outside of curve,
- derailment occurred in transition or in curve of 400 m radius or less.

Rolling stock

- bogie centre distance close to spacing of joints or low spots,
- stiff wagon body,
- broken body centre plates,
- evidence of springs going solid,
- broken or missing springs,
- defective or missing hydraulic or friction snubbers,
- sidebearer damage,
- shifted loading in lateral direction

Track

- Track prior to point of derailment had several low joints or "soft" spots in the loaded condition at a spacing commensurate with the train speed and the roll natural frequency of the first vehicle derailed.
- All of these factors are not required to induce harmonic roll:
- 10 mm cross level deviation on at least three consecutive joints on one rail will induce roll, but generally will not cause wheel lift,
- 15 mm cross level deviation on at least three consecutive joints may put a wagon on the verge of wheel lift,
- 20 mm cross level deviation occurring over at least three consecutive joints will probably create lift,

- difficulty maintaining track top,
- broken track components indicating excessive vertical forces from springs on rolling stock going solid,
- dry rail making flange climb easier.

Bogie Hunting

Hunting or harmonic yaw is oscillation in the plane of the track surface. At speed, wheel-rail interaction forces excite the bogie so it rotates about the centre pin and causes the wagon to move laterally. These periodic motions may grow until the wheel flange contacts the rail.

Bogie hunting is probably the most violent oscillation that occurs on vehicles. Sustained hunting can lead to flange and rail wear, bogie component wear and gauge widening. In extreme cases, any wheel of either bogie may climb the rail. Infrequently a wheel can momentarily widen gauge to allow an opposite wheel to drop in. However the track would need to be in a weakened state, or extremely high lateral forces would be necessary. The high lateral forces may also trigger a track buckle.

The extent of hunting is dependent on many factors, e.g. bogie type, centre pivot type, sidebearer type, condition of wheel profile, mass of vehicle and speed. Three-piece bogies with worn wheel profiles under empty vehicles can hunt violently at speeds as low as 80 km/h.

Hunting is far more likely to occur on straight track than on curved track as the flange forces during curving tend to damp out hunting. It can be triggered by transition from a curve, rail joint or minor misalignment. Hunting generally will not occur in the rain because wheel tread-rail head frictional forces are reduced. Friction journals (plain bearings) are much less likely to hunt than roller bearings because wheel-rail excitation forces are not as readily transmitted from the wheelset to the bogie sideframe and bolster. The likelihood of hunting increases as wheel tread wears past cylindrical to hollow.

Bogie centre spacing and CoG height do not affect hunting. Constant contact side bearings raise the threshold speed. Bogie hunting is more likely on track with tight gauge but the effect is more severe on track with wide gauge, i.e. lateral forces generated will be higher.

When an obvious cause cannot be established for high speed derailment, use the following checklist to determine if hunting was the cause or contributed. The presence of some of these factors will generally not be sufficient to determine whether the bogie was hunting and whether that caused the derailment:

- empty or lightly loaded wagons, or flat wagons with one trailer or container,

- train speed of at least 80km/h,
- dry rail,
- straight track or curvature of 1000 m radius or greater,
- roller bearing bogies,
- mode of derailment is usually wheel climb,
- wheels have substantial service wear and visible tread hollowing. Similar flange wear on both wheels of a wheelset,
- contact band on rail close to gauge corner giving higher effective conicity by contacting on more steeply tapered part of wheel,
- excessive wear of friction wedges making it easier for lozenging,
- lack of or inoperative hunting control devices such as constant contact sidebearers. (Check for loss of preload or excessive longitudinal free travel),
- excessive wear of bolster gibs, centre bowl vertical wall or roller bearing adaptor backs; evidence of springs rotating indicating wagon has a history of hunting,
- excessive wear of coupler carrier wear plate coupler underside and draft gear support plate indicating wagon has a history of hunting.

Hunting is not likely to be the sole cause of a derailment. It can be a significant factor. Hunting is not an acceptable derailment cause when no speed restrictions were violated and no critical bogie components were excessively worn.

Lurching

This is a sudden lateral transient movement in the action of the vehicle such as a single deflection from the centre towards one or other of the running rails.

Lurching is not really an oscillation since it does not continue and is usually damped by the suspension and gravitational forces. The cause can usually be identified as a lateral deviation in track alignment or a sudden change in curvature. Obviously, this results in lateral forces between wheel flange and rail that could possibly contribute to wheel climbing.

Figure - Lurching

Surging or shuttling

This is a backward and forward cyclic movement along the track.

Surging or shuttling of vehicles in motion is generally caused by poor design where the placement of bogie traction rods is not on the same plane as the bogie pitching centre
(e.g. on single deck interurban and 1955 suburban cars). This motion is very unlikely to be the primary cause of a derailment.

Pitching

This is the front and rear of a vehicle or bogie alternately rising and falling.

Vehicle body pitching usually occurs in conjunction with bogie bounce where the natural bounce frequency coincides with the body pitching frequency. This usually only occurs where there is insufficient vertical damping in the bogie suspension. Bogie pitching indicates insufficient primary suspension damping or the natural frequency of the primary suspension is coincidental with a cyclic track borne disturbance such as square mechanical rail joints. At speeds above 60 km/h pitch and bounce are commonly caused by abrupt changes in vertical track stiffness such as bridges and road crossings.

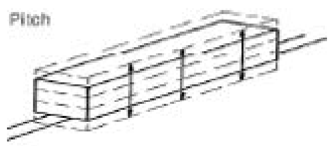


Figure – PitchingFigure - Bounce

Longitudinal Forces

This section deals with the longitudinal dynamics leading to the development of high longitudinal forces in the train. Forces are transmitted longitudinally throughout the train via the couplers, draft system and centre sill of each vehicle.

When a train is operating on straight track with the slack stretched i.e. "in draft", the drawbar forces act along the centre line of the track. When a vehicle negotiates a curve, the couplers become angled to the vehicle centreline and there will be a lateral component of the longitudinal force.

High draft forces tend to stretch or "string line" the train, and tend to pull the wheel flanges against the inside rail of the curve. If the force is increased further there is a tendency for the inside wheels to pivot about the low rail with the outer wheels lifting off the high rail.

If the slack is bunched instead, buff forces tend to buckle the train outward, pushing the flanges against the outside rail and adding to the usual outward curving forces. Additional lateral force is developed between the wheel tread and the top of the rail when the angle of attack is not zero. If the lateral component of the longitudinal force is high enough, the inside wheels may be lifted off the low rail, which effectively reduces the contact angle of the wheel flange to the side of the high rail lowering the L/V ratio for wheel climb.

Lateral forces also may cause a lateral shift of the track, particularly under light wagons.

The angle of the coupler to the longitudinal centreline of the vehicle is influenced by the geometry of the vehicle and of the vehicles to which it is coupled. The longer the vehicle, the greater the coupler angle. Combinations of long and short wagons can cause even greater angling of the couplers and hence a greater lateral component of the longitudinal force.

Jack-knifing

Jack-knifing derailments are caused by heavy buff loading. Jack-knifing always involves at least two vehicles when a unit wheel climbs or overturns the high rail. Lead consists are most likely to jack-knife when making backup movements, during heavy dynamic braking, during heavy application of the independent brake at slow speed or on high curvature track, or in turnout or crossover when one or more trailing units lack auto coupler limiting blocks or have worn alignment control draft gear. On locotrol trains, the lead portion may jack-knife as a result of lead locomotive wheel slip or power failure with the remote portion still pushing. Jack-knifing may occur after a train separation, burst hosebag, heavy application of independent brake or emergency brake application, particularly with heavy loads towards the rear of the train. It is more likely at slower speeds under these circumstances.

Couplers of the two vehicles involved are usually cocked to one side of the curve. Consequently, as in stringlining, the marks found on the rail top in a Jack-knifing are usually short. If the high rail overturns, wheel flange marks appear on the web or flange of the rail.

Stringline

Stringlining derailments may occur in the head one-third portion of a train accelerating from less than about 25 km/h in high curvature territory. This type of derailment is characterized by:

- derailed wagons are pulled over the low rail, usually in a straight line, or sometimes the low rail overturns and the high wheel drops in,
- derailed wagons are generally empty, lightly loaded, long overhang or long wagon/short wagon combinations,
- a short flange mark on top of the rail.

Speed

The maximum permissible speeds allowed on the various lines are shown in RailCorp's TMR 001 – OS 001 1M – Train Operating Conditions Manual TOC or RailCorp's Working Timetable.

There are two key aspects of speed which affect the behaviour of the track. One is EXCESS SPEED and the other is CRITICAL SPEED.

Loads on track increase with speed for any vehicle. Different components or conditions in the track mean different rates of increase. Speed and load limits are set to match what the track structure can sustain. For example, most deflections of the track are elastic and not destructive, and stresses are kept within proper limits for the materials used in the track structure.

EXCESS SPEED, i.e. speeds exceeding these limits, damage the track and add energy to any vehicle that is misbehaving.

The effect of speed in the presence of a track irregularity is to increase the impact forces between wheel and rail. The magnitude of the impact force is directly influenced by wheel load and unsprung mass which, if high enough, will result in damage to the wheel, axle bearings and track structure.

Excessive speed on curves, i.e. over that for the ruling superelevation deficiency allowed for the curve, will result in increased throwover and wheel load transfer due to centrifugal force.

Excessive speed may be evidenced by damage to the track, misalignment, etc. before the point of derailment.

Every vehicle has a natural rhythm or frequency for each of its normal movements (roll, pitch, bounce and yaw). The vehicle responds when the track condition disturbs or "excites" the vehicle. The vehicle will respond most when the "exciting" rhythm matches the vehicle's natural frequency. The speed at which this happens is the CRITICAL SPEED.

The question of whether or not the permissible speed was exceeded over that part of the track where a derailment occurred may be an important factor in assessing the cause. In certain cases where speed records are not available, an estimate of the speed can be obtained from the distance run by the train, or any part of it, after the derailment occurs. If necessary, seek expert advice from Rolling Stock experts. Many factors are involved in forming a reasonably close estimate of speed from the distance the train travelled after an emergency application of the brakes, and this can only be done satisfactorily by an expert.

3) Derailmentsan analysis of defects in permanent way and rolling stock

Synopsis:

This monograph deals with the analysis of various forces andcauses that lead to derailments. Theoretical considerations of the various formulae adopted for the analysis of derailments and actual derailment tests conducted with test vehicles on experimental tracks 1 some of the foreign railway have been discussed. The various defects irregularities that normally cause or contribute towards the derailment؛ the usual defects of some of the important components in the rolling stock are discussed in detail for (i) 4-uheeler stock (ii) bogie vehicles of both goods and coaching stock. On the permanent way side, the importance of each parameter of track geometry and its contribution towards derailments has been discussed in detail The coefficient of rail wheel friction which has a definite influence on the quantum of the derailing forces, the effect of tyre wear on hunting, and the effect of uneven loading (payloads)have .also been touched upon. The duration through which a derailing force could be sustained without mounting for various flange force wheel load ratios has been indicated. Although not connected directly with the analysis, of causes of derailments, the criteria which the locomotives and rolling stock should satisfy for general operation on the Indian Railways and the ride index criteria have also been indicated. At the end a few suggestions for improvements have been made which may bring down the incidence of derailments.

1. **Introduction:**

1.1 The causes of derailments are often unsatisfactorily established.

A mathematical derivation into the causes of derailments is almost impossible in view of the innumerable and instantaneous variations, in the wheel loads, relative positions of the wheels to the rails, conditions of track, and the dynamic conditions of vehicle, which depend not only on the instantaneous track condition but also on the history of the vehicle travel upto that instant. The only possible way of coming to a conclusion is, therefore, to assess as best as possible the conditions of the vehicle and track in the .near vicinity of the point under consideration and the history of the vehicle travel immediately before.

1.2 It is for the following reasons that the midsection derailments are hard to explain.

(i) There are too many factors that may constitute a cause and the derailment may be a result of the combination of many unfavorable conditions. Derailment happens in many cases when neither the rolling stock nor the track is defective, and the wear and tear or deformation is within the specified permissible tolerances or sometimes even better than average; it seems to result from coincidence of many casual factors none of which individually counts much and constitute the primary cause of the given case.

(ii) It is difficult to reproduce/reconstruct the scene and test again as, when the same vehicle that derailed before is operated on the same track at the same speed as before, the derailment may not this because it is impossible in all probability to reproduce the aforesaid factor in exactly the same condition.

1.3 The Permanent Way Engineer is particularly at a disadvantage as the damage done to the rolling stock involved in derailment very often removes the clues of defects in the rolling stock, and if found are sometimes debatable as to whether they are the causes of the derailment or as a result thereof.

1.4 While the rolling stock have been prescribed with rejectablelimits, no such tolerances have been laid down for the permanent way. The Engineering section of the Indian Railways Conference Association (19th meeting) were of the opinion that is was not possible to draw up limits of track irregularities, infringement of which would result in derailments, because of the very variable limits that could be permitted under different conditions and it was not practicable to determine any specific demarcation between safe and unsafe conditions.

1.5 The introduction of formal standards places a heavy responsibility upon the authorities formulating them. In the absence of such Standards, however, those carrying out the work of maintenance had to make their own assumptions and moreover responsibility for an accident tends to be assigned in accordance with indefinite standards tacitly assumed by someone after the accident. fixing up of such tolerances has been under the consideration of railway administrations from time to time Guide lines for such tolerances, if indicated, would be the framework on which could be discussed as to how far a particular defect, could have contributed to a derailment under the circumstances.

1.6 An attempt has been made to analyse the various factor that come into play and their effect in causing the derailments, along Suggestions for improvements which may bring down the incidence of derailments.

2Theoretical consideration

2.1 forces acting at rail-wheel, contact

The wheels of railway vehicles move over rails safely in sinusoidal waves due to conicity of the wheels and are held within the rails by their flanges. The forces acting on the rail wheel contact rails by their flanges. The forces acting on the rail wheel contact are:

(i) The weight of the vehicle, transmitted by the wheel on the rails which keeps the vehicle onto the track.

(ii) The flange forces exerted by the vehicle on rails.

Before the wheel can mount the rail the forces acting on the contact, surface should be such that their vertical component continues toneutralise the weight being exerted by the wheel and their horizontal component continues to push the wheel across the rails the wheelflange is being lifted over the rail over its entire height and derails.

2.1.2 When the wheels of the vehicles roll parallel to the rails i.e., there no angularity, the forces acting on the wheel contact (when the flanges make side contact with the rail) are as under:

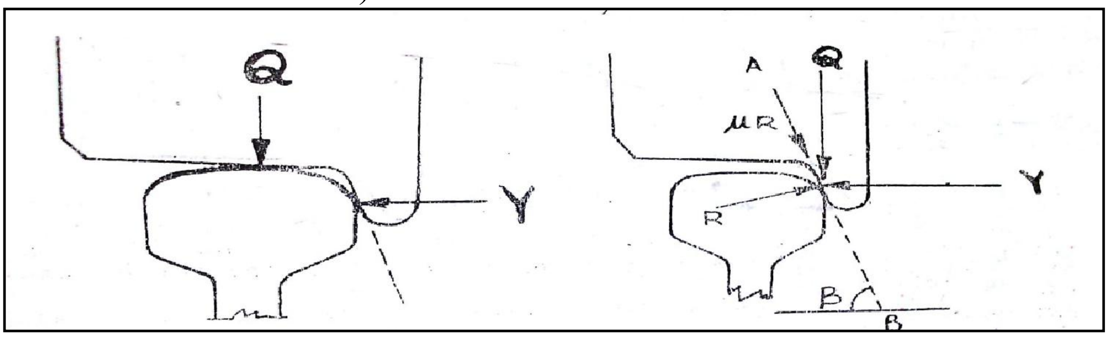


Fig. 1 Fig. 2

When the wheel flange just makes side contact with the rail head, there are two points of contact one at the top of rail head another at the Side by the flange (as indicated in fig. 1) where the weight Q and the flange force Y act. As soon as the Wheel tread slightly lifts over the rail head, taking the case of a jump before derailment, the point of application Q is transferred to the Conical part to the point where Y acts and the force diagram will be as shown in Fig. 2 (The force diagram will be the same as in Fig.2 for the trailing contact when the wheel moves with angularity, while for the leading contact it will be as shown in Fig. 4). As the Component of Y starts moving the wheel along the contact plane in the direction- BA the frictional force acts in the opposite direction AB and opposes the motion. For the wheel to derail the forces along BA should overcome the forces in the direction AB. In the equilibrium condition.

$$Q + \mu \sin \beta = R \cos\beta \ ; \ Y = \mu \cos \beta + \sin \beta$$

$$\therefore \frac{Y}{Q} = \frac{\mu \cos \beta + \sin \beta}{\cos \beta - \mu \sin \beta} = \frac{\mu + \tan \beta}{1 - \mu \tan \beta}$$

2.1.3 Taking the rail wheel fictional coefficient as 0.25 and the- flange angle β for Indian Railways rolling stock as 86°, i.e. a slope of 1 in 2½ (except for steam locomotive wheels for which β = 60°), $\frac{Y}{Q} = 7.15$ at the point of mounting. Such heavy flange forces do not occur in practice and there is no possibility of derailments when the motion of the wheels is *parallel* to the rail without any angularity.

2.1.4*Nasal's formula*: In practice, however, the axles run with slight angularity due to various clearances in rolling stock, the play between rail and wheel, flange, misalignment and curves. The position of the wheel moving with angularity is indicated in Fig. 3 which is the general case.

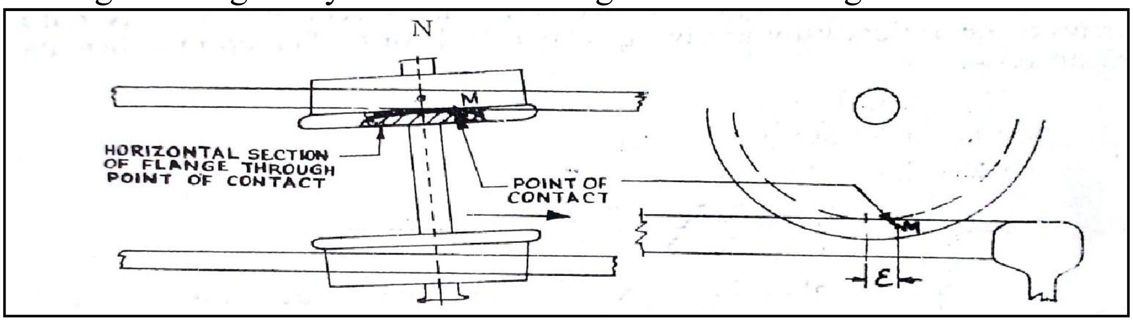


Fig. 3

Due to angularity, the point of contact 'M' of wheel flange and rail is ahead of the point of Contact 'N' of wheel tread' with the rail and the wheel can lift off the rail table due to rotatory motion of the wheel. As the rotatory motion of the wheel flange at the point of contact is downwards the frictional force **µ**R now acts in the Opposite direction i. e. upwards.

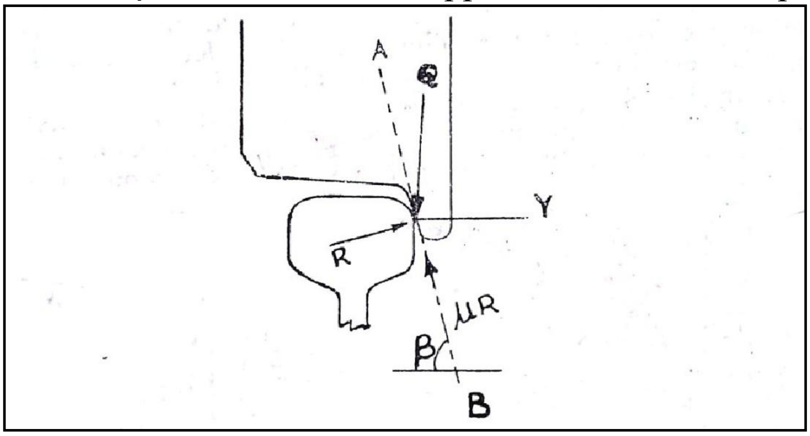


Fig. 4

Nadal has worked out the flange force - wheel load ratios for equilibrium conditions as under (Fig. 4).

$Q = Rsin\beta + \mu R \sin\beta;$

$$Y = Rsin\beta - \mu R \cos\beta$$

$\therefore \frac{Y}{Q} = \frac{1-\mu\cot\beta}{\mu+\cot\beta} = \frac{\tan\beta-\mu}{1+\mu\tan\beta}$ Nadal's equation

With $\mu = .25 \ and \ \beta = 68°, \frac{Y}{Q} = 1.4$

Nadal had established that derailment conditions are created when the flange force exceeds 1.4 times the instantaneous wheel loads.

The normally occurring flange force level gives a ratio which is much lower than the limiting value of 1.4.

2.1.5Y.Satohas sought to overcome the discontinuity in Nadal's formula theoretically by applying Garter's 'theory of creep in a formula of the type,

$$\frac{Y}{Q} = \frac{\tan\beta - f}{1 + f\tan\beta}$$

Where f = k sec β tanα

And k=3.5$\sqrt{\frac{wheeldia.(inches)}{wheelload\ (tons)}}$

This formula takes into account the angle of attack α also, and the ratio of Y/Q for equilibrium conditions decreases as the angle of attack 1 increases within certain limitations. As the angle of attack increases the predisposition to derailment also increases, as the equilibrium condition is reached with lower values of lateral force Y.

2.1.6 Charter's theory:Nadal has not considered the effect of the other wheel on the axle on the derailment consideration. Since both the wheels are integrally connected with the axle, Chartet took into consideration the effect of the other wheel and coning of wheels. Healsoe stablished that the frictional coefficient at the wheeltread - rail head contact of the nonderailing wheel is different from, that at the wheel flange-rail head contact of the derailing wheel.

As the wheel 1 (Fig. 5) moves i. e. rolls along with the axle in the direction AB the wheel flange hugs the rail and is forced to move in the direction BG ultimately making it .move in the direction AC. Thusthe wheel rail contact point moves in the transverse direction Be, the point of contact on the 'tread of wheel 2 moves in the direction B' C' and the frictional force in the opposite direction (μ_2R_2) as indicated above.

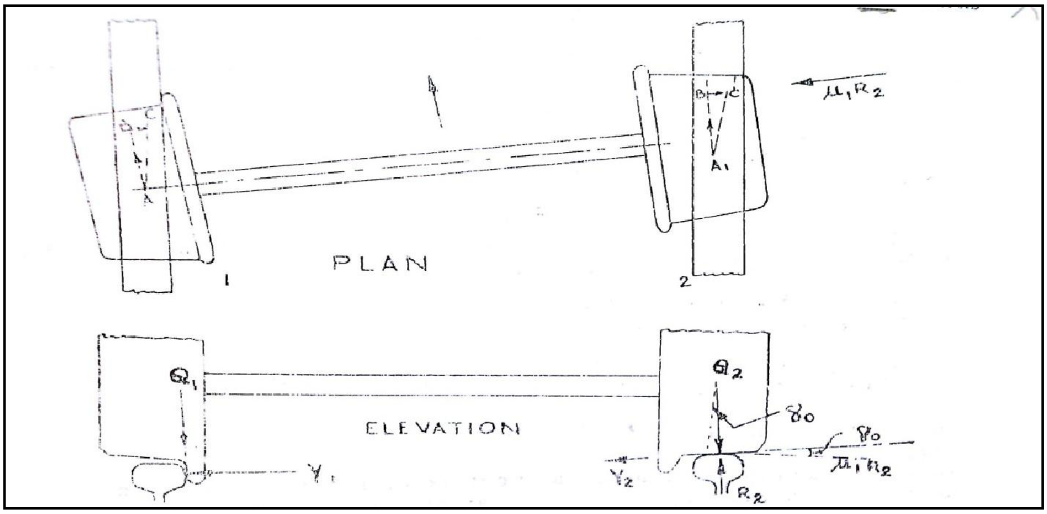


Fig. 5

The effect of forces on wheel 2 on the mounting of wheel 1 has also been considered by Chartet,

$\frac{Y}{Q} = K1 - K2.\frac{T/4}{Q}$, where $K1 = \frac{\tan\beta-\mu}{1+\mu\tan\beta} + \mu_1 + 8\circ$

and $k_2 = 2\ (\mu_1 + 8\circ)$

$\mu = \sqrt{2.\mu}$ approximately, μbeing the coefficient of friction between two cylindrical surfaces (i. e. rail- head and wheel tyre), μ being the coefficient of friction between cylindrical and plane surface (rail head and wheel flange).

With $\mu = 0.2$, $\mu_2 = 0.3$approximately and $\beta = 68°$

$K_{1=}$ 1.873| or 2 and K_2=0.7

$\therefore \frac{Y}{Q}$= 2-0.7 $\frac{T/4}{Q}$(4 being the nominal wheel load)

$\frac{Y}{Q}K_1$-K_2. $\frac{T/4}{Q}$;this formula may be rewritten as when $Y=k_1.Q - k_2T/4$

When Y = 0, Q = $\frac{K2}{K1}.\frac{T}{4}$ which is the minimum value of wheel load whenno lateral force acts. The component of this minimum load of the wheel when in leading contact, is no longer capable 'of -balancing the sum of frictional forces including upward movement of the flange.

Thus $Q_{min}=\frac{0.7}{2}x\frac{T}{4}$ i. e. 35% of the nominal wheel load, or maximum unloading of 65%.

2.2 Effect of track depressions and rolling stock defects on wheel load

2.2.1 Maximum variation in spring camber: Spring, deflection 'x' (Fig. 6) at which a wagon will unload one wheel completely is given by $X = \frac{TGd}{2L}$; the effect of a track depression under one wheel only is tounload and increase the camber/decrease the deflection of a spring. If there la depression in track of '$\frac{1}{2}Z_0$' and the unloading caused by thespring is x then

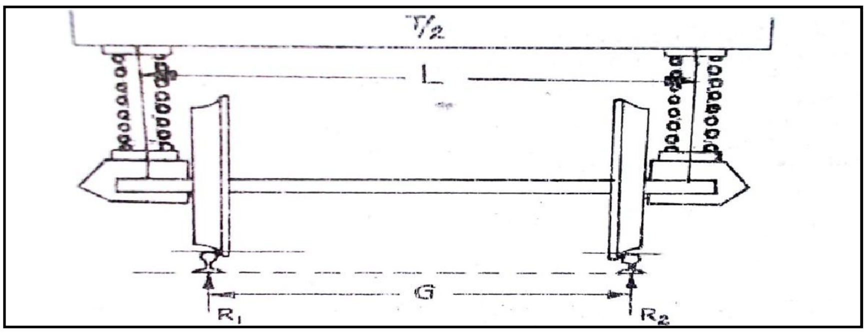


Fig. 6

x = $\frac{1}{2}Z\frac{L}{G}$ hence a track depression of $1/2\,Z = G/L\ x$is Xwill cause unloading of the spring, but as x =$\frac{TGL}{2L}$; $\frac{1}{2}Z_0$ =$\frac{TGD}{2L}.\frac{G}{L}$ =$\frac{Td}{2}(\frac{G}{L})^2$.As indicated earlier the unloading that can be permitted is 65%. Hence the track depression for 65% unloading or $\frac{1}{2}Z_0$= 0.65$\frac{Td}{2}(\frac{G}{L})$. As soon as the spring

unloads, around the diagonaljoining the two springs, the frame itself will rock since one corner is no longer supported. As the spring opposite the other diagonal is still loaded the body will rock until this opposite spring is unloaded. There must, therefore, be a drop of 2x$\frac{1}{2}Z_0$ =Z_0 for the wheel, to unload to' the extent of 65%. The track depression required to create this condition is. $Z_0 =$ 0.65Td $(\frac{G}{L})$.^{2}

From this it would be seen that there is greater danger for derailment when,

Weight of the vehicle T is lower, which explains the reason for empty wagons derailing more.

(2) $\frac{G}{L}$Is lower-this value for MG is 0.71 and for BG it is 0.78 explaining the reason why MG wagons figure more in derailments.

(1) If the specific deflection of the spring'd' is lower, which explains why vehicles with stiff springs, figure more in derailments.

2.2.2 Effect of spring camber difference: The bearing spring in the rolling stock transmit the load from the underfarme on to the axle and keep the wheels on to the rails by adjusting themselves to the variations in loads and track irregularities. If the eyes of one spring are not in the plane determined by the other three springs (due to difference in spring camber and/or spring seats being at different levels) the wheel loads will be altered. If there is a difference of x′ in spring camber/deflection of one spring over the other (Fig.7), say between 1&3 then the between 1 and 3.

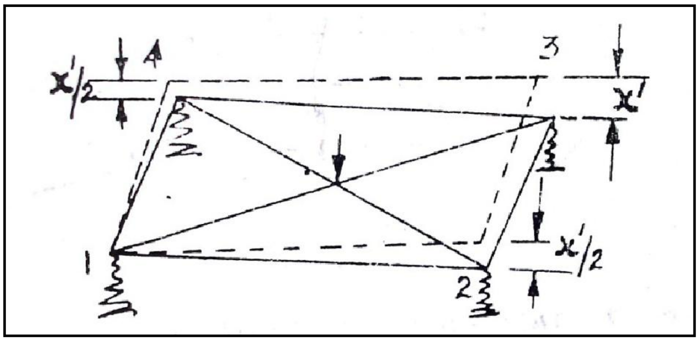


Fig.7

Since this makes the plane distorted, the underframe rocks till the value at 2 and 4 is $\frac{X'}{2}$, so that all the 4 points of suspension of the underframe remain in one plant. Thus the effect of camber difference effecting the unloading by tilting the underframe is x' /2. The corresponding drop in the safety depth at rail level is $Z_2 = \frac{x'}{2} \cdot \frac{G}{L}$

The worst effect on value of Z_2 would be when two diagonally opposite springs have this defect and the value of Z_2 will be more due to cumulative effect.

2.2.3 Effect of torsion of underframe : The vehicle is supported on the 4 Points of suspension below the underframe. The body of the vehicle is flexible due to elasticity. As soon as any wheel is unloaded the point of suspension becomes free and the torsional moment due to the weight of the body deflects the underframei.e. point of suspension downwards and to that extent keeps the wheel down and adds to the safety depth. Kereszty has given the corresponding safety depth at rail level as

$Z_4 = (\frac{G}{L})^2 T. \frac{\psi}{4}$; ψ being the inverse value of the factor of torsion.

No specific studies have been made on Indian Railways stock to determine their flexibility. Based on similar trials carried out by German Railways this safety depth may be assumed as 5 mm for MG and 13 mm for BG, the wagons with longer wheel base being more flexible. Rivetted and open wagons would be more flexible than welded and covered ones respectively.

2.2.4 Effect of cant gradient or twist: on a level track or gradient track or on a circular track with equilibrium superelevation the load is uniformly distributed on the springs as the points of rail wheel contact are in one plane. But on transitions or twisted tracks the four points of rail wheel contact are not in one plane (Fig. 8), one point being higher or lower than the plane formed by the other three points. This Wheel is either gradually raised or lowered causing unloading of springs, which is equivalent to a track depression of $b \times i$where b is the wheel base, and $i$the cant gradient.

2.2.5 Factors coming into play for derailing the vehicle and the stabilizing *forces:*The various factors that come into play in causing conditionsfor derailments have been discussed above as worked out by Nadal, Chartet and Kereszty. Before a wheel can derail by mounting it has to unload itself to the extent of %65ofthe nominal load coming on it. The following safety depths are available (Fig. 9).

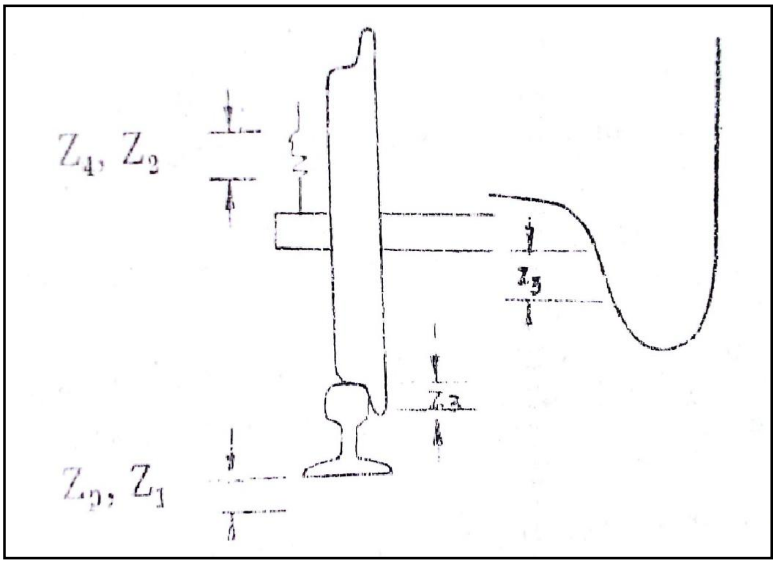


Fig. 8

(i) The extent to which the rail can be allowed to be depressed for unloading 65% of the wheel load=Z_0.

(ii) The safety depth of the conical portion of the wheel flange by which the wheel is to be physically lifted–'Z_3'.

(iii) The torsional deflection of the vehicle preventing the wheel from mounting –'Z_4'

The above are, however, offset by the reduction in safety depth due to spring defects like camber variation –'Z_2'and the cant gradient or twist–'Z_1'. Hence further depression in track that is still permissible before a derailment is possible.

Z permissible = $(Z_0+Z_3+Z_4)-(Z_1+Z_2)$

These permissible values for the 4 wheelers of BG (Two types each) on cant gradients of 1 in 720 and 1 in 360 are indicated in Table 1 below

Table 1'Z permissible' value for empty wagons.

Type of wagon	Weight in tons 'T'	Wheel base in feet 'b'	Spessific deflection Of spring 'd' mm/ton	Z_0 Z3 Z4 Z1 Z2 (in milllimeters)	Z premissible=Z_p	
					Considering Z_4	Excludin Z_4
					$(Z_0+Z_3+Z_4)-(Z_1+Z_2)$	$(Z_0+Z_3)-(Z_1+Z_2)$

A Cant gradient of 1 in 720

(i) Broad gauge

C	10.25	15	6	25	9	13	6.4	5	35.6
KC	10.15	11.5	6	25	9	13	5	5	37

(ii) Metre gauge

C	5.55	10	5.2	10	8	5	4	5	14
KC	5.75	12	5.2	10	8	5	5	5	13

B Cant gradient of 1 in 360

(i) Broad gauge

C	10.25	15	6	25	9	13	12.7	5	29.3

KC	10.15	11.5	6	25	9	13	9.7	5	32.3
(ii)	Metre gauge								
C	5.55	10	5.2	10	8	5	8.5	5	9.5
KC	5.75	12	5.2	10	8	5	10.1	5	7.9

Note:- (i) the above are for BG, G=5.75 ft.
L=7.33 ft.
For M, G=3.43 ft.
L=4.83ft.

(i) And when one spring has a camber difference of ½" or 13 mm. the value of Z_2 is to be doubled when two diagonally opposite springs have a higher camber/lesser deflection of 13 mm.

(ii) The above value exclude the depth of semi-circular portion at the root of flange and 5 mm may be added to the 'Z permissible' value for this.

From the above it is seen that on a track with a cant gradient of 1 in 720 and with one spring having an irregularity of ½" the safety depth i.e. 'Z_0' is reduce to about 1" (24 mm) for BG, and $5/_{16}$" (8 mm) for MG if the torsional effect is not considered; and to about 1½" (37 mm) and ½" (13 mm) respectively for BG and MG if torsional effect in wagon is considered. On a track with cant gradient of 1 in 360 these safety depth are further reduced as will be seen from the Z_p values given in Table 1. It is thus seen that while the BG wagons have great safety margin, MG wagons have much less explaining the reason for more number of derailment on the metre gauge.

2.3 Preconditions for a derailment: It has been shown that a derailment by mounting is probable only.

(i) when the unloading of wheel is to the extent of 65% of the nominal weight i.e. reduction of Q.

(ii) the vehicle moves in such a way that the wheel tracks angular position with respect to the rails instead of rolling parallel to the track so that the leading wheel flange is biting against the rail gauge face i.e. increase of angularity α,

(iii) the flange force Y acting on the wheel exceeds twice its instantaneous wheel load minus 70 percent of nominal wheel load $\frac{Y}{Q} = 2.0 - 0.7\ \frac{T/_4}{Q}$; i.e. increase of Y,

(iv) these condition continue to prevail till the wheel flange completely mounts and derails i.e. time phase or the duration of time through which the derailing force is sustained, and

(v) the point of contact of wheel flange with the rail head i.e. eccentricity 'ϵ' by which the flange bites the rail.

Summarising, the factors that contribute to derailment are,

(i) reduction of wheel load 'Q'

(ii) increase of flange force (i.e. lateral force) 'Y'

(iii) increase in angularity of wheel with the rail 'α' (see para 3.2.4).

(iv) increase in eccentricity of the wheel flange contact 'ϵ'

2.4 Wheel load variations – dynamic conditions

2.4.1 Parasitic motions of vehicles on track: It is due to the parasitic motion of vehicles on track that the horizontal- vertical wheel load ratio changes.

Due to the irregularities in the geometry and elastic characteristics of the track, variations in the load from the design value in the case of vehicles and in oscillational characteristics, the vehicle execute a number of parasitic motions, which are oscillational in character. The definitions of these given by the Pacific Locomotive Committee are as under:

Nosing: Transverse oscillation of the engine on its springs about a vertical axis (i.e. about z axis) pursuing a sinuous path along the track.

Rolling: Transverse oscillation of the engine on its springs about a longitudinal centre line (about the x axis).

Hunting: The two movements defined above rarely occur separately but are generally found acting together in varying proportions. The resulting oscillation is described as hunting.

Lurch: One semi-amplitude of movement in the action of hunting, viz, an individual deflection from the centre line towards one or other of the running rails (i.e. along the y axis).

Pitching: The front and back ends of an engine alternately rising and falling about a transverse horizontal centre line (i.e. about the y axis).

This is sometime referred to as galloping.

Bouncing: The movement of vehicle having up and down linear oscillations along the z axis is called 'Bouncing'.

(Note: x axis parallel to the rails, y is parallel to gauge and z is vertical axis).

2.4.2 Wheel load variations occur as result of both dynamic and pseudo-static phenomena. The horizontal vertical force ratio $\frac{Y}{Q}$ may become unfavorable (i.e. high) either owing to a high value of flange force or low value of vertical wheel load or owing to combination of these. The following are a number of causes which result in load variation.

a) Weight transfer owing to traction forces.
b) Load fluctuations owing to bogie and body oscillations on the suspension.
c) Load variations owing to track conditions by which the contact points between the wheels and the rails cease to be
d) co-planer and the inability of vehicles to accommodate them.
e) Asymetrical construction or initial distortion of the vehicle or uneven distribution of the payload.

At low speeds up to about 15 to 20 m/h, causes (c) and (d) are probably almost entirely responsible for derailment, while at higher speeds the causes are the combination of higher lateral forces owing to body nosing and wheel load reduction as a result of inability to accommodate track conditions or some conditions of asymerty in the vehicle.

In most cases reductions in load at certain wheels are accompanied by similar increases at other wheels. The consequence of this is that corresponding springs are called upon to support loads in excess of nominal values and the dynamic loads are superimposed on these.

2.5 Derailment tests with experimental track

2.5.1 Tests by JNR: Japanese National Railways conducted running tests in 1967 on freight cars on 11 Km experimental track in Hokkaido, which was to be abandoned as a result of route re-location. The result of these tests generally supports the theoretical considerations discussed above. The experiment conducted and the result of the tests are extracted below from "Japanese Railway Engineering−Vol. 8 No. 4 December '67."

"First experiment: The first experiment at this test track took place last July. To cause a derailment in this experiment, extreme irregularities to an extent almost inconceivable in actual operation were put to both track and rolling stock, and a single freight car of representative type in JNR was made the test car. The condition set were as follows:

Track: The following three kinds of track irregularities were made each at an interval of 50 m.

(a) Combination of alignment irregularity of a 20 mm wave height and cross level irregularity of 5 mm.

(b) Alignment irregularity of a 30 mm wave height.

(c) Combination of alignment irregularity of 30 mm wave height and cross level irregularity of 5 mm.

Each kind of the irregularities is made to from a sine wave of a 20 m wave length and repeated three times so as to make a continuous wave.

Rolling stock : Three representative loaded freight cars were picked out, the WARA 1 type for two-axle box car with double link suspension device, the TASA tank bogie-car for the high rigidity of its car body and the SEKI coal hopper bogie-car for its high centre of gravity. The conditions are set as follows.

(a) Stander condition ……… WARA, TASA and SEKI

(b) Difference of diameter between the right hand and the left hand wheels 2 mm (in reverse relative position longitudinally)………….. WARA and SEKI.

(c) Ratio between wheel weight in sum total in diagonal line, 30% …………WARA

(d) Deviation of centre of gravity (towards right), 30%.........WARA

(e) Side bearer clearance reduced to half (from 12 to 6 mm)……….TASA

(f) Overloading, 10%...........SEKI

Under these condition, the experiment run was made on the straight section in July, and on the curved section in September. In these tests, an aggregate of about 150 runs were made and derailment occurred 15 times."

Result of test: The following conclusions were generally reached.

(a) The lateral force-wheel load ratio does not rise much up to a certain point, but once this is exceeded, it change by a leap for a slight increase of speed.

(b) The drop-off of wheel load has a greater effect on the derailment than the maximum value of the lateral force.

(c) The wheel load decreases remarkably when the track is given big and continuous irregularities. When the residual wheel load becomes close to zero, the probability of occurrence of derailment increase. In such a case the derailment coefficient (lateral force versus wheel load ratio) generally increases. The derailment occurs when the wheel load remains zeros until the vehicle comes to a time when lateral force rises, while it does not if simply the wheel load becomes zero and that is all. It means that what counts for much is the duration of this drop-off of wheel load, and another important problem is the time phase between the generation of lateral force and the drop-off of wheel load.

(d) Consider the values of the derailment coefficient actually measured, the critical value, 0.8 hitherto prescribed seems adequate. It is observed that, over 0.8, the rate of rise of derailment coefficient rapidly increase.

(e) The decrease of wheel load and the increase of derailment coefficient are conspicuous where the track has heavy, continuous irregularity in alignment. The effect is augmented when cross level irregularities are added to this, and a derailment takes place when the amount of these irregularities is great.

(f) Even when the alignment irregularity and cross level irregularity coincides and their amount is big, they do not have so much effect on the wheel load drop-off, or derailment coefficient, if their extension is not longer than something like a half wave length.

(g) The difference in diameter between the right hand and left hand wheel does not seem to have an adverse effect on the running safety if it is within an extent of 2 mm.

(h) The difference in weight between the right and left wheels and between diagonal wheels augments the drop-off of wheel load and the derailment coefficient.

(i) Overloading to the extent that the centre of gravity shifts to a higher position also aggravates the wheel load drop-off and the derailment coefficient.

(j) The amount of elastic displacement caused by the lateral force immediately before derailment, is usually about 3 mm."

2.5.2 Tests by ORE: The Office of Research and Experiments of the International Union of Railway have set up the "B-55 specialists 'committee on the question of 'prevention of derailment on distorted tracks'. This committee have conduced 100 derailment test on empty goods wagons, the results of which are discussed in para 6.3.3.

2.5.3 The conclusions of the above tests generally support the theoretical consideration as discussed in foregoing paragraphs.

3. Defects in four rolling stock

3.1 Wheel: The profile of a new BG IRS wheel is given in fig.10.

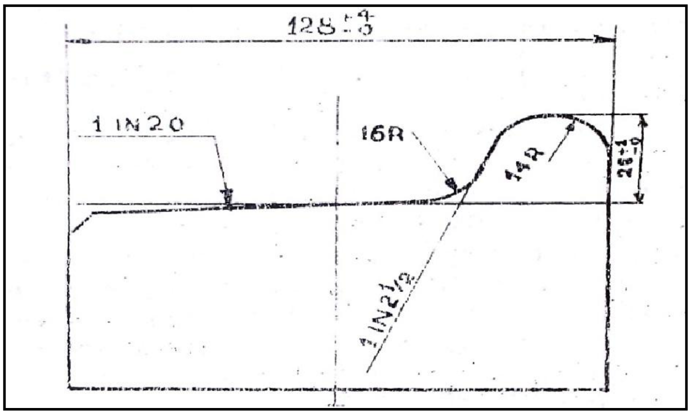


Fig. 10

3.1.1 Wheel flanges see Fig.10 (a). The defects in flanges are

(i) Thin flange i.e. thickness less than 5/8 " for both MG and BG,

(ii) Sharp flange i.e. the root of the flange having a radius less than 3/16".

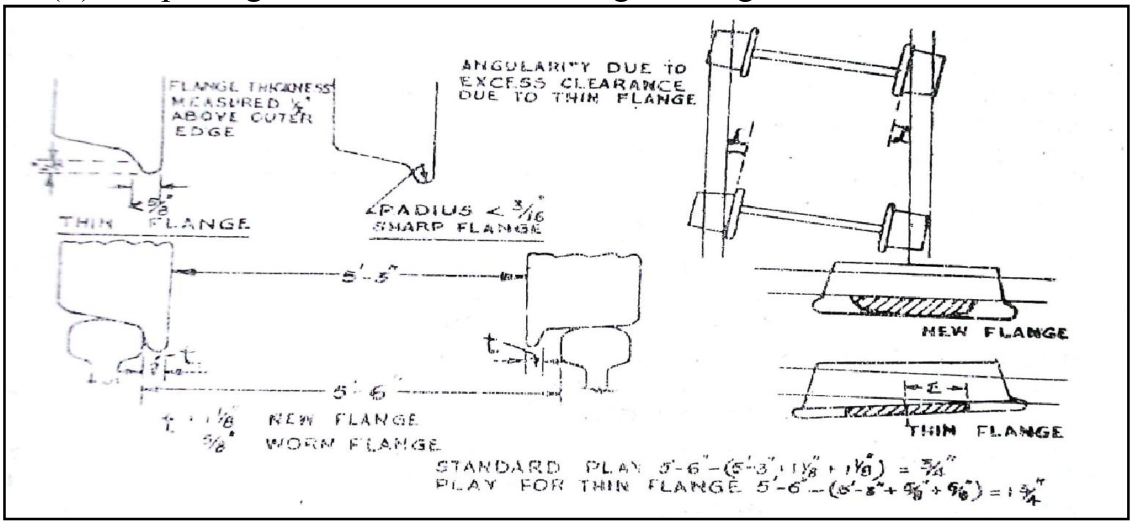


Fig.10 (a)

Thin flange: Wheel flange of less than 5/8"is rejectable. As the flange wears, the rail-wheel flange clearance will be increased. These extra clearances will create conditions oscillations, as these are proportional to the square of the amplitude and the flanges hit the rails with greater force-Y, and increased kinetic energy. Thin flanges enable the wheels to take up greater angularity and also increase the eccentricity of wheel flange contact with the rail from the point of tread contact i.e. increase of 'α' and 'ϵ'. The standard play on the Indian Railways is3/4" (19mm)for new wheels, which is higher than on many foreign railway. Reduction of standard play reduces the angularity, eccentricity and oscillations and is a step in the right direction. This is further discussed in para 6.2.5.

Sharp flange: this is dangerous as the flange is worn in such a way that the profile becomes vertical, and the flange forms a fine sharp edge. This increases the eccentricity of the point of contact (fig. 10b) of the sharp edge of the flange with the rail further in advance of the tread contact thus enhancing the predisposition to derailment due to the bitting action of thin flange which tries to ride over the rail. Sharp flanges may also split open slightly gaping points.

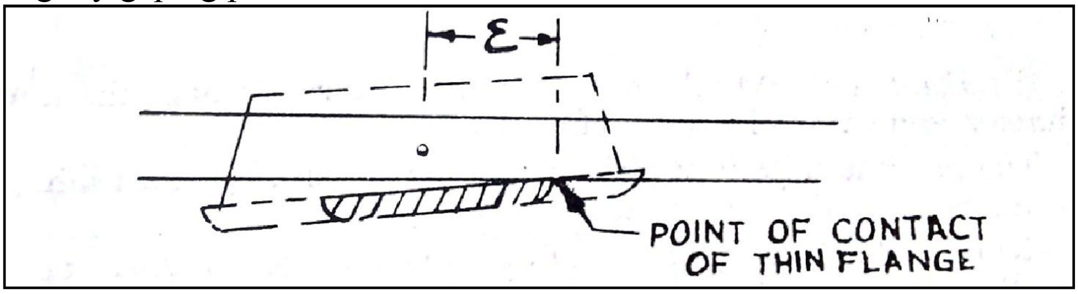


Fig.10 (b)

'ϵ'= Eccentricity of point of contact with thin flange.

3.1.2 Tyres: The deflect in tyrs are (i) hollow tyre; (ii) flat tyre; (iii) variation in diameter of wheel due to unequal wear and (iv) cracked or broken tyres.

(i) Hollow tyres: They neither effect the wheel load Q nor increasing the flange force directly. There is no angularity or eccentricity. But the wear on tyres has the effect of increasing the conicity of the wheel tyre, which reduces the critical speed of the wagon beyond which excessive hunting and oscillations take place,which in effect increase the flange force Y, increasing the change for derailment. The effect of wear on tyre has discussed in para 9. Hollow tyres, however, have the danger of developing a false flange. The false flange while moving over points and crossing in the trailing direction (fig.11) can enter between tongue and stock rails and exert excessive thrust on stock rail fittings causing the fittings to give way in some cases split the switch open after bending the stock and tongue rails.

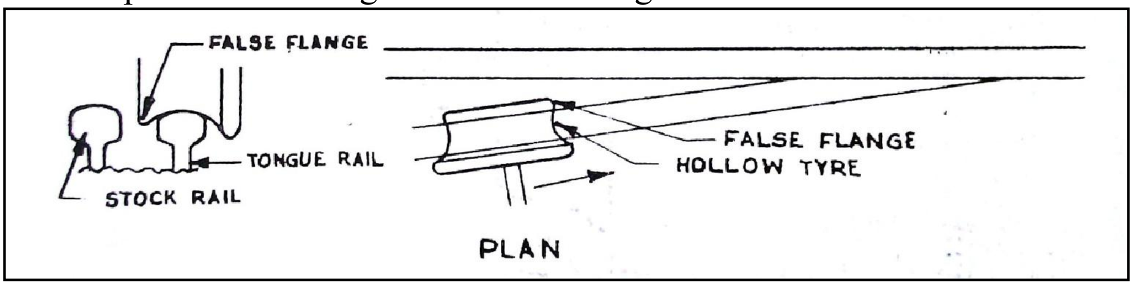


Fig. 11

The false developed due to hollow tyres can also be dangerous at the wings of the crossing (Fig.12) as the same rides over wing rails thus lifting the wheels and creating conditions favourable to derailments in addition on causing damage to track due to its lifting and sudden dropping.

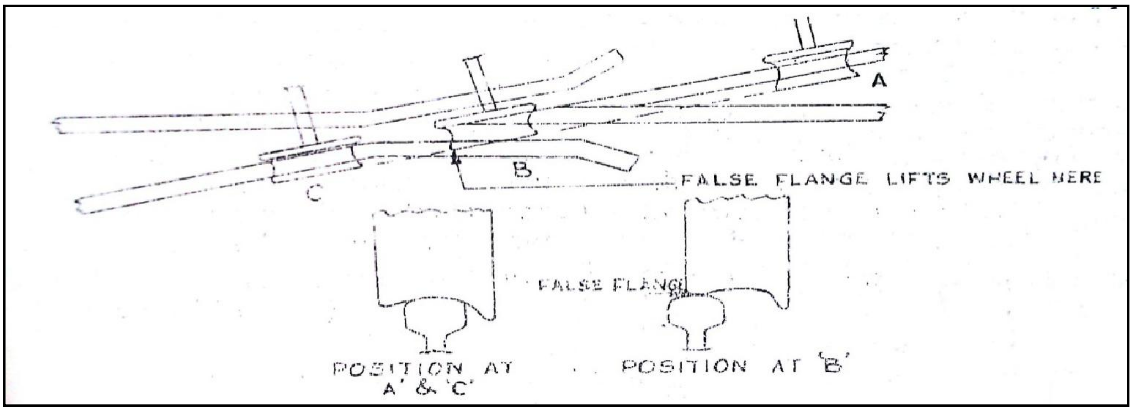


Fig. 12

(ii) ***Flat tyres:*** Wheels having flats on tyres damage the rails due to impact and cause high contact stress.

The maximum permissible values are as under, and rolling stock exceeding these must be rejected.

{Rule 56 (clause 88) and rule 57 (clause 73) of IRCA conference Rules Tart 111 and correction slip no· 56 of 1-3-1966}.

Flat places on tyres—Permissible maximum

(a) Broad gauge ………………………………75 mm (3") For BOX, CS, BCX, BOBX, BOBS, BWT, BWH, BWL, BOI, BOX Mk I& II, BRH and BRS… 63 mm ($2\frac{1}{2}$").

(b) Metre gauge …………………………. 51 mm (2")

(iii) Difference in wheel diameters:

(a) The wheel diameter should not differ in the same axles differ in the same axle distance of travel varies and the axles take angular position increasing the angularity 'α' and creating conditions for derailment. The diameter is not be carefully insured at quarter points.

This should be measured at a distance $2\frac{1}{2}$"of from the wheel gauge face. (fig.13)

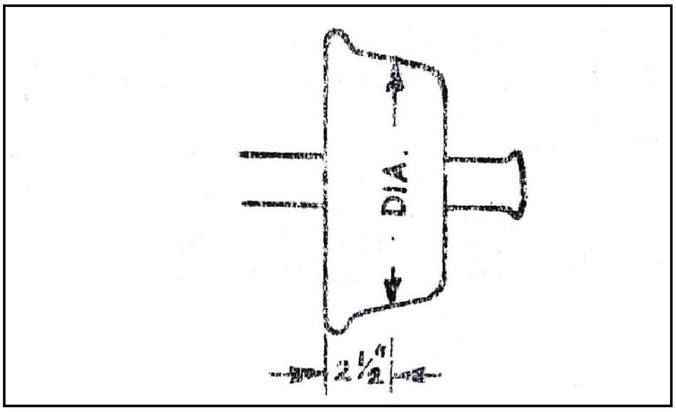


Fig. 13

(b) Difference in wheel diameters permitted in different axles of the same vehicle.

{Rule 10 (10) of lRCA[27]--C.S.NO.66 of 25.88.1996}

Permissible maximum variation.	BG	MG
(i) Four wheeler stock	25 mm (1)	13 mm ($\frac{1}{2}$ ″)
(ii) Four wheeled bogie trucks (IRS & conventional)	13 mm ($\frac{1"}{2}$)	10 mm ($^3/_8$ ″)
(iii} Four wheeled bogie trucks (schelirn ICF, BEML & ICF)	5 mm ($^3/_{16}$)	5 mm ($^3/_{16}$)

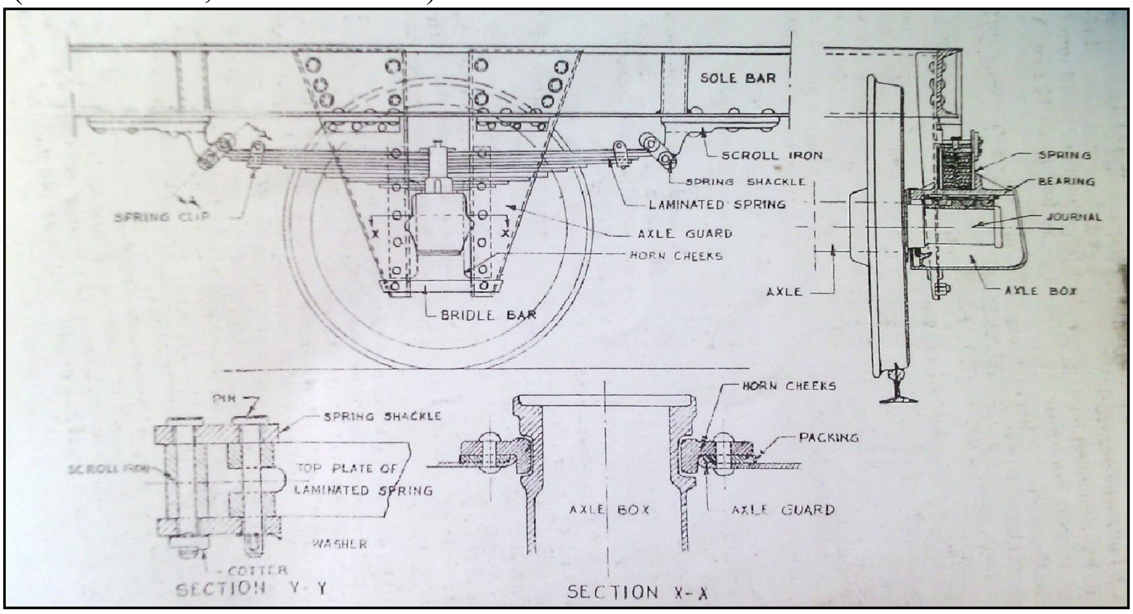


Fig. 14

Any excess difference will have an adverse effect on buffer heights.

(iv) Wheel gauge: The wheel gauge is to be checked at quarterpoints. If the gauge is different it indicates the bending of axle and the axle cannot stand fatigue. Either it causes fracture or develops hot axle due, to wobbling. This is a serious defect if the wheel gauge differs at quarter points and no tolerance is permitted.

Rule 56 (97) IRCA--Conference Rules-Part 111, however permits wheels tight or slack to gauge upto$\frac{1''}{6}$(± 1.5 mm) but no

Variation is permitted at quarter points.

Correction slip No. 36 of 15.3.65 States as under:

Vehicles or wagons may be refused if the gauge of any pair of wheels is on Broad gauge, less titan 1598.5 mm or more than 1601.5 mm. on Meter-gauge, less than 928.5 mm or more than 93'1 .'5 mm.

New wheels are now being worked to ± 1 mm tolerance.

3.2 journals and journal boxes

The running gear and suspension system of a four wheeled BG coaching stock has been shown in fig. 14.

3.2.1*Journals:* If the journal center lines are not coaxial or there isunequal (elliptical type) wear, then the journal box goes up and down when the axle rotates due to difference in height effecting the springs. This defect is to be examined along with the behavior of the springs.

(1) *Journal boxes*

(i)Bearing brass: The bearing brass should be of the same thickness and should bear uniformly on journals without riding over the collar. If the thickness of bearing brasses is different, then the difference must be made good by packing plates to see that the springs come into action simultaneously and share the load equally.

(ii) Hot axles: When any hot axle is developed and the bearing brassis burnt or displaced, then the wheel is completely free of load. Further the journal is free to move in the box and the wheel takes excessive angularity and combined with, the condition of unloading, this wheel in all probability will derail by mounting.

(2) *Transverse and longitudinal clearances in journal bearing and journal boxes:* In addition to the wheel flange clearance ‘c'(fig 15) betweenthe wheel and the rail (i. e. standard play +wear of flanges+ slack -gauge) which sets up oscillations with greater amplitude and greater flange force, the flange forces are futher increased due to clearance between the brass and journal i. e. clearance'b' and further lateral (transverse) clearance between the axle box groove and the horn, cheek i. e. clearance 'a'. The effect of the clearances 'a' and 'b' is to allow the body of the vehicle

move and oscillate transversely, till these clearances are used up to and fro and causing variation loads and excessive flange forces Y.

(3) Maximum permissible lateral clearance 'a' between the axle guard and the axle guard groove or between horn cheek and axle box for broad and meter gauge and for both bogie & 4-wheelers is 10 mm. ($^3/_8$ ″){Rule 10 (3) iii-IRCA Conference Rule}[27].

(4) Lateral play between .brass and journal not to exceed on BG and MG

Bogie- 6 mm ($\frac{1}{4}$ ″)

For wheeler – 10 mm ($^3/_8$ ″)

It is preferable to reduce the lateral clearances to the minimum possible as this will reduce oscillations hunting and flange force. The effect of these Clearances on the angle of attack ‘α’ is given below.

3.2.4 Determination of the maximum angularity of the wheel with respect to raill:[34] If a Vehicle, with a wheel base 'b' runs on a curve of radius ‘r’ with the leading and trailing wheel flanges in contact with the outer rail, the angularity of the leading wheel to the rail is $\frac{b}{2r}$.

If then the frame is swivelled, maintaining the flange contact of the leading wheel against the outer rail but making the trailing axle wheel flange bear against the inner rail of the curve i e. swivelled through the total gauge/flange and axle box clearances cumulatively equal to ‘C', then the angularity .is increased by $\frac{c}{b}$ .The total angularity with respect to rail will, therefore be$\frac{b}{2r} + \frac{c}{b}$. Hence the .angle of attack wil1 increase with the play between rail and wheel and the lateral clearances in the axle box and axle guards.

3.2.5 Effect of lateral clearance of on critical speed and lateral force.

Due to spring suspension, which is ideal, the lateral motion produced by track irregularities is damped below the critical speed, while above it the amplitude of oscillatory lateral motions grows without limit.

Gilchristhas reasoned that the effect of journal clearance is depression of critical speed comported with the same vehicle without clearance. Experiments establish that the lateral force Y due to hunting is much higher for a vehicle with journal clearance than one without these clearance. Flange forces recorded with and without clearance are dictated in (fig. 16).

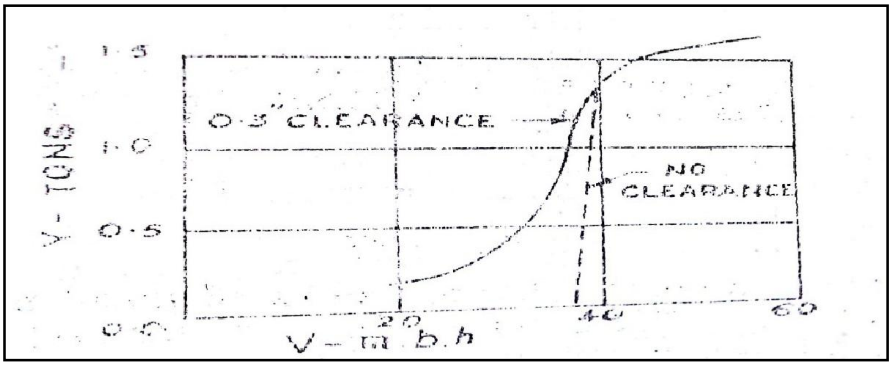


Fig. 16

3.2.6 Clearance between journal and underside of axle box top: the slipper plate and bearing brass should be in position as otherwise the transfer of load seriously effected. Limited of badly worn brass should be such as keep the following minimum clearances above the journal collar-Rule 65(16) IRCA (This does not apply to stock fitted with roller bearing axle boxes)

Dimension 'x' (fig.17) not less than' $\frac{1}{4}''$ for 9″ journal

Do $\frac{3}{8}''$ for over 9″

Do $\frac{5}{8}''$ for IRS and over 10″x5″ Journal.

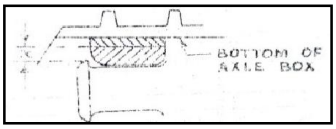


fig. 16

3.3 Axle guard and horn Cheeks: Bent, broken, cracked axle guards are not permitted—(Rules 56 and 57 IRCA). If the axle guards bent the axle box will not be able to move up and down freely and box will get jammed making the spring assembly ineffective and causesconditions for derailment. Bridle bars, horncheeks broken or damaged are not permitted.

The axle guard sustains heavy stresses due to traction, braking and rough etc. Hence the connections to the sole bar should be strong and bolting is not permitted. -Missing or loose rivets are not permitted.

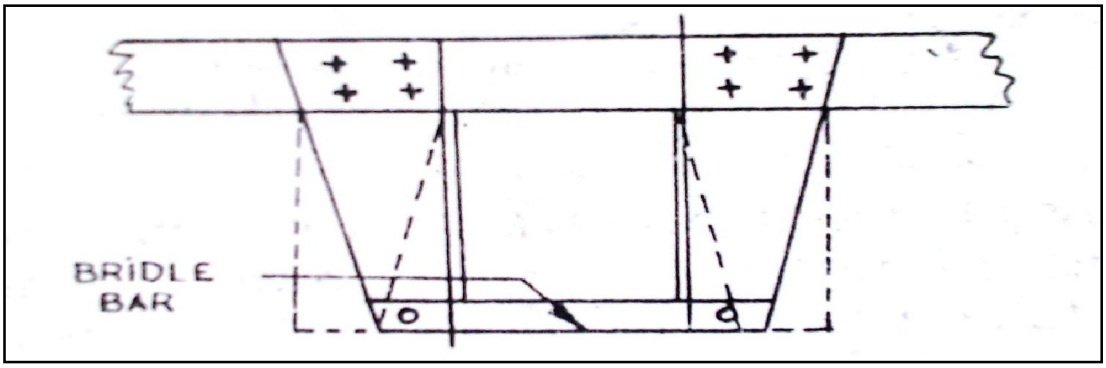


Fig. 18

Broken bridle bar is serious deflect and when it is missing or ineffective the axle guards being fixed at one end to the sole bar take all the forces from the axle boxes and deflect elastically (as shown in dotted lines in-Fig. 18) increasing longitudinal clearances and angularity. This will not only -set up additional oscillations but effects the free working of the axle box within its horncheeks.

3.4 ***Springs and spring assembly***

3.4.1 In four wheeler stock, normally the laminated springs are used. The defects in these can be classified into 3 main categories.

(i) Defects causing interference with working of springs.

(ii) Variation in performance of different springs on the same vehicle.

(iii) Failure of springs

3.4.2 Defects causing interference with working of springs.

If the defects are such that normal and free functioning is interfered with, their the load transmission system is badly effected and that particular wheel on which this spring bears cannot adjust itself to any unevenness in the track and the wheel will' be prone to derailment, as there will be no stabilising force keeping it on to the rail.

Spring assembly

(i)Shackle plates: If the shackle plates (Fig. 19) are longer thanpermissible or .the holes are elongated due to being worn out or the bolts bent and worn out or if the scroll iron is fixed wrongly then the shackle plate or the spring ,eye bear against the sole bar effecting the suspension system increasing the liability to derailment. If there are any indentation marks under the sole bar showing signs of the spring eye or shackles hitting the sole bar it is a clear indication that the free functioning of the spring, is affected under dynamic conditions (Fig. 20&21) even' though under static conditions the sole bar may not be riding over the springs. Flange of any wheel within 1" of the bottom of wagon is rejectable {Rules 56 (78(and 57 (61) IRCA}.

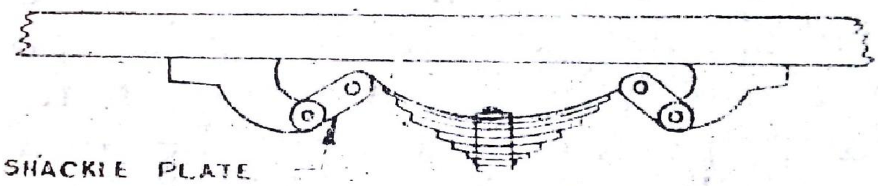


Fig. 19

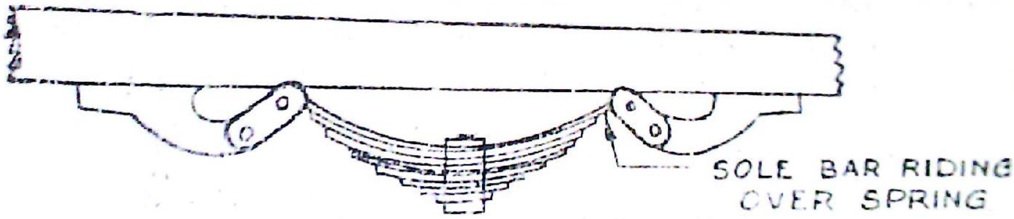


Fig. 20

Fig. 21

The permissible wear between shackle pin and shackle plate or shackle pin and spring, eye should be a maximum of $\frac{1}{8}''$ during service, while the shop tolerance after P.O.H. be 1/32″as recommended in para 537 of Latham and Isaacs report. Clearance along the length of the pin is not exceed 1/8″ [para 539 of Latham & Isaacs Report on derailment and Rule (14) IRCA].

(ii)Spring buckle: Shifting of a buckle, or a broken buckle is a serious defect as then the plates not act in unison. The shifting of a buckle along spring assembly is rejectable as recommended by Latham and Isaacs in their report vide para 532. A deformed spring is an indication of a loose buckle. Any plate or buckle displaced from its central position by $\frac{1}{2}$" or more is rejectable {Rule 56(77) and 57(60) IRCA} [27].

3.4.3 Variation in the performance of springs: The following permissiblelimits have been laid down.

(i) The difference in working camber of any 4 springs under a unit under load should not exceed $\frac{1}{2}$" (para 532 of Latham and Isaacs Report)

(ii) A variation in the free, camber of a spring should not exceed + $\frac{1}{8}$"for loco and carriage springs,$\frac{1}{4}$" for BG wagon springs and +3/16" for MG wagons and –O for all (Paras 519 and 526 of Latham and Isaacs Report) [6]. The -effect of springs having different cambers/deflection in reducing-the safety depth has already been dealt with in para 2.2.2. With an initial camber difference of $\frac{1}{2}$"between two springs on the same axle, while the wagon floor remains horizontal, there is strong liability to derailment with a low .joint of$\frac{1}{2}$ " on MG and on the broad gauge when the track conditions are poor (i. e. a low joint of 1"). If only one spring under a wagon has reduced camber, it is probable that in most cases the frame would adjust itself so that one wheel does not especially tend to lift, but where similar conditions exist on diagonally opposite wheel of a. four-wheeled wagon, then this adjustment is not possible, and the wagon would - be in a dangerously unstable condition. When such conditions are combined with movement over a curve oscillation for any cause, additional forces would in certain cases add to the danger.

3.4.4 Specific deflection of the springs. The specific deflection of all 4 springs for any one unit should be the same so that the- loads are shared uniformly and the vehicle is in level. If the specific deflection of one sprang is more than the other then the wheel under the weaker spring is likely to derail due to its carrying less load. Further, it is, desirable that the specific deflection for 4-wheeler roiling stock is as high as possible. The difference in load and free camber for a 4 wheeler empty wagon on .MG/ with a specific deflection of$\frac{1}{4}$ " per ton is only $\frac{1}{4}$ " (taking sprung mass as 4 tons) and the springs tend to get off loaded easily because of this small .range of deflection to take up track irregularities. It is therefore, suggested-that provision of springs with greater specific deflection for 4-wheeler MG stock be considered. There are, however, other factors which bring in limitations to the increase of specific deflection of springs *viz.,*the permissible maximum and minimum buffer heights, the ratio of the :tare weight to maximum-gross weight of the stock for economic .haulage of pay loads, the permissible tyre wear etc. The critical speeds, which govern the maximum permissible speeds, wil1 be higher with increase in spring stiffness from the center pitching and bouncing considerations needing stiffer springs for stock of fast trains.

3.4.5*Replacement of springs:* When any spring is replaced in a vehicle either on sick line or at any other time it is suggested that load deflection tests are made to satisfy that the spring is identical to the other springs with the same free camber/specific deflection. It is suggested that piecemeal replacement by springs of various manufacturers be prohibited and an entirely new set of spring, identical in size and characteristics and previously tested, be fitted to the rolling stocky. Rules 56(79) and 57) 62(IRCA Conference Rules-Part III permit top plate of one - spring thicker by " than the top plate of other springs. It is suggestedthat this be deleted and a stipulation made that the spring be identical in size shape and load deflection characteristics.

Examination of free and load cambers: In normal train passing examination only the load cambers are checked for any possible variation and with the present facilities and time available nothing more is possible. The handicap of this has been that variations in free camber go undetected till the wagon derails and no staff responsibility for such to variations could be fixed as the Train. Examining Staff are unable to check it—quite a few derailments were attributable to this on the Lumding-Badarpur. Hill section. Although the working cambers were within permissible limits, free cambers were widely varying with the result certain sprigs do not carry any load and these wheels mount on curves. Since all trains including goods trains are examined on pit lines when they enter ghat sections facilities for examining the free camber, by hydraulic lifting arrangements to make the springs free, would enable detection of such defects. The initial-‘ and recurring costs and the extra time involve in checking the free, cambers also will perhaps be worthwhile as it will prevent many derailments.

3.4.7*Packing plates under spring:* Packing plates are provided under the springs and above the axle box to make up for the difference -in brass thickness etc. by the train examiners and to make the working camber equal. It is often found that flimsy thin metal sheets are put in sometimes which drop off on run and the springs do not take any load, with the consequence of a wheel running light and probable derailment.

Load and free cambers of wagons fitted with such packing plates should be specially checked in case they figure in derailments.

3.5 Effect of spring friction

3.5.1 The laminated bearing springs have a spring friction as the plates slide one over the other The spring friction is found to the extent of 7 to 13% for 4 plated springs and 12 to 18% for six plated springs, the greater number of plates tending to contain proportionately more friction. When the vehicle is standing on a track the top of which is perfectly level, the wheel loads will not necessarily be equal even though the vehicle may be symmetrical, in all respects, due to the spring friction. The frictional force required to overcome this resistance must be added to the elastic force under loading conditions and subtracted from it under unloading conditions. .The hysteresis diagram for a spring after 4years service as investigated by ORE is shown in Fig. 22.

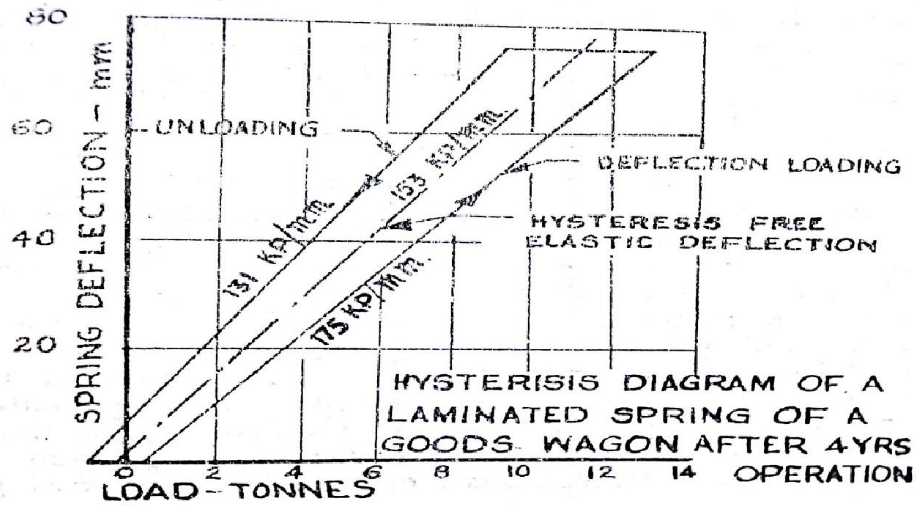


Fig. 22

It is thus seen that when a spring is deflected by x due to a transfer of load, the spring does not come to the original camber position when this load is relieved but to about 10 to 15% less of this deflection. The internal friction is thus a disadvantage as the spring does not regain the original camber in case of relief of load and this reduces the safety depth ‘z permissible' to some extent.

SNCF among others have succeeded in reducing this friction to about 60% by centralised maintenance and reducing initial gaps between leaves and preventing dust which increases fiction. The spring friction generally increases during service in course of maintenance interval.

Springs being the major cause of derailments as far as rolling stock are concerned, it is necessary to improve their quality, by manufacturing them at centrally located workshops, with reduced spring friction and also to see that the load deflection characteristics are identical for similar springs.

3.5.2 ***Spring friction—*stiffness effect *on deflection:*** The laminated springs are a Simple design and are suited for force transmission and guidance in longitudinal as well as in transverse direction. The friction between the leaves obviates the need for additional shock absorbers so that the total cost of suspension is low. On the other hand, the frictional forces depend on the maintenance and .lubrication condition of the leaves and vary correspondingly.

3.5.3 The- specific deflection for a laminated spring is given by the formula deflection per ton in inches, d $= \frac{0.1s^3}{\mathrm{BN}t^3}$

where N is the number of plates, B is the width of plates in inches, t is the thickness of the plates in 1/16 ths inch and s is the span in inches, the coefficient 0.1 being arrived with E=13,000 tons/Sq. in. The maximum variation of E (Modulus of Elasticity) in various steels does -not exceed 15%. If the thickness of a plate is increased by 25% the stiffness of the same will be increased to double the value as stiffness is proportional to the 3rd power. Since specific deflection of a spring differs with the variation in thickness of the plates it is suggested that $\frac{1}{8}$" variation permitted in IRGA. Conference Rides—Part III, for top plate thickness in a hearing spring {i. e. for the .top plate of. a bearing spring, to be thicker by 3 mm ($\frac{1}{8}$") than 'the top plates of the other springs other springs of a vehicle/wagon} be deleted (as already discussed in para 3.4.5) as this spring is likely to have a different specific deflection.

3.5.4 Normally the cause of a different specific deflection is of a nonstandard spring having variation in the- width, thickness, span, material and difference in the number of plates. The above formula (Para 3.5.3) ignores the interleaf friction. When the springs are manufactured the plates, are to be painted i. e. lubricated with a mixture of oil and grease before assembly, as per clause 4.1 of IRS specification R8-62. This friction will be very smal1 at the time of lubrication and is negligible. Further as per these specifications, clause 10.5, only 5% of the springs are tested for determining the range and deflection per ton and the characteristics may vary even though the springs comply with the dimensions mentioned in the drawing.

Further under Service conditions, the inter-leaf friction is likely to increase: After some service, the springs are likely to be stiffer than the new ones, and sometimes the stiffness may go as high as double the value of original stiffness endangering the safety. In addition normally there is a slight loss of camber in service due to permanent set. This increase in stiffness may be due to the loss of lubrication entry of dust and moisture due to slight opening of the spring plates under service etc. It is, therefore, desirable to check the stiffness, byload deflection tests, of all or a higher percentage of springs at the time of accepting from manufacturers, or when overhauled/manufactured in workshops. To this effect, it is suggested, that IRS, specification R-g be modified to provide for a greater percentage check. It is also necessary that springs of locomotives, laminated springs coaches and if possible that of all rolling stock are periodically tested for any variation in designed stiffness.

3.6 ***Defects in underframe***

3.6.1 The underframe is to be examined for horizontal and vertical twists. The horizontal twist can be measured by marking points on pegs vertically below the 4 corners of the underframe (Twisted frame shown dotted in Fig. 23) after keeping the vehicle on level track and measuring the diagonal distances. .If there is a longitudinal twist the distances will not be equal. The effect of this distortion will be to keep the axles always angular to the track (i. e. the direction of tractive force) and the wheels bite against the rails due to angularity making the vehicle prone to derailment.

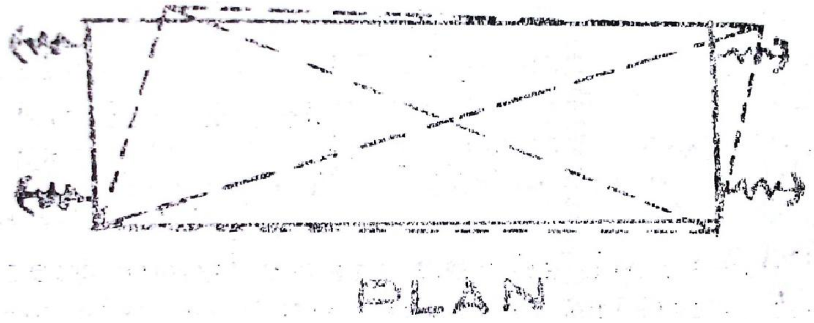


Fig. 23

6.3.2 ***Vertical twist:*** The vertical twist may be assessed by measuring the height of sole bar, keeping the vehicle on a level track If there is a vertical twist in the vehicle i. e. all 4 points of load suspension are .not in one plane then one of the springs has to adjust itself first and then transfer this to the others and this causes excess loading in some and less loading on other springs. The lightly loaded ones i. e. where the twist of the frame is upwards tend to derail first. A vertical twist of 33/4" can contribute appreciably to derailment.

3.7 ***Buffers***

3.7.1 Latham and Isaacs in their report on "Derailments" state that "it is to be feared that wagons are not stopped as frequently as they should be for infringement of buffer heights. A closer check of the buffer heights at the - time of train passing is advisable. The buffer height variations permissible for BG and MG stock are given below.

		BG	MG
Height of buffers Max. (for unloaded stock)		2'---7$\frac{1}{2}$"	1'--11"
	Min (for loaded stock)	3'--4$\frac{1}{2}$"	1'--9"

If the buffers of adjacent wagons are not .in the same level due to different conditions ofloading or spring characteristics the buffer drawgear takes an inclined position. In case of braking on down grades and in sags when the buffers exert compressive forces the lighter vehicle will be derailed by lifting due to the vertical component of the buffing force (Fig. 24). This force will be considerable when these wagons are trailed by a good number of wagons and is augmented on downgrades.

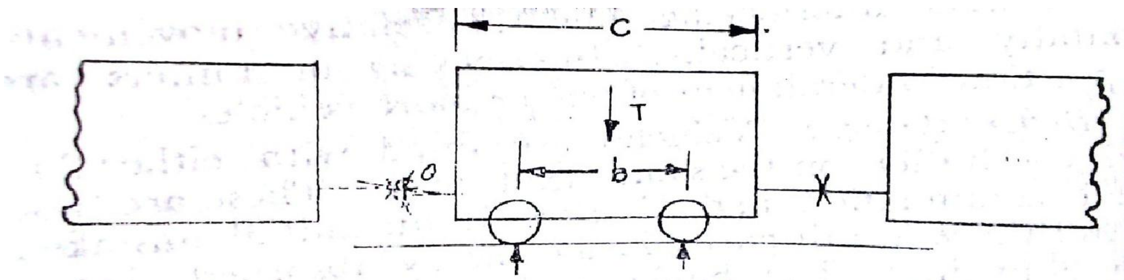


Fig. 24

The buffing force F required to make the wheel load zero is givenby

$F == \frac{b}{b+c} . \frac{T}{\sin\theta}$; T being the weight of the wagon. It will be seen that less force will be required to lift the wagon with increase of or increase in difference in buffer heights. F will be least when θ is maximum i. e. a lightly loaded wagon has a fully loaded wagon adjacent to it. A light wagon with a fully loaded wagon on .one side only is more vulnerable than one with fully loaded wagons on both sides[5]. The vertical eccentricity of buffers due to difference in buffer heights thus causes reduction in Q and makes the vehicle prone to derailment.

3.7.2*Horizontal eccentricity of buffers for BG stock:* If the head stock is bent so that the center of buffer face is displaced in any direction more than 38 mm ($1\frac{1}{2}$") from its normal position it is rejectable {Rule 56(92) IRCA}. In the case- of horizontal eccentricity the reaction on buffers not only exerts lateral force but also makes the wagon angular to the direction of motion (Fig. 25). Due to excessive flange force Y and angularity α wagon will be prone to derailment.

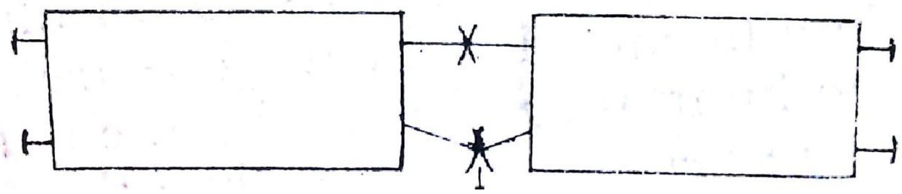


Fig. 25

3.7.3 *Dead buffers:* A dead buffer is dangerous when train is shuttling or retarding or when it travels in a sag as the compressive force due to trailing loads acts only through one buffer acting eccentrically with respect to the reactions on the other two buffers on the other side of the vehicles. Excessive flange forces are created to counteract the couple created by the eccentricity of buffing force.

3.7.4*Low buffer heights:* When the springs are weak and wagons are fully loaded the buffer heights are .less than the permissible minimum height and often go unnoticed. Latham & Isaacs have, therefore, recommended a minimum buffer height to be fixed for empty wagons which shall be higher than the minimum height permissible plus the amount of spring compression for maximum permissible load.

3.7.5*Interlocking of buffers:* 13" diameter buffers are provided in 4-wheelers while I8" dia buffers are provided in bogie stock (i. e. diameter at the plunger face). The larger diameter is to provide for the greater lateral sweep (i. e. radial) in curves. If the maximum and minimum heights are not adhered to and or the buffers are bent, there is risk, of-buffers interlocking due to relative movements of buffers horizontally and vertically interlocking of buffers are a definite predisposition to derailment of one or more vehicles.

3.7.6*Buffer springs:* Vehicles are fitted with either 20 inch ton or 40 inch ton buffers on the same head stock. These are not visible for g external examination and if there is an initial mistake in providing these and. there is a disparity in the energy absorbed these may derail during shuttling. The buffer, casings are bolted to the head stock.

Sometimes these bolts slacken and the buffers might drop off on run.
The buffer plunger is retained in position by the floating .spindle and a recoil spring is fitted to the spindle. If the security nut works out then the plunger pushes itself out from the casing and he result is that of a dead or missing buffer.
3.7.7 ***Buffer defects in general:*** All the above defects are more pronounced when there is shuttling and when the train is coasting on a down gradient.
3.8 ***Brake rigging:*** This has been responsible or derailments which are caused by incorrect centralisation and adjustment of rigging and blocks, permitting uneven application of brake power and wear in gear. Fracture in truss bars, hangers pins usually occur, which on track. Brake block deficient insecurely fastened or so thin that the flange of the wheel is 1/4" or less from the brake beam collar when the brake is applied, is rejectable {Rules 56-(39) and 57(33) IRCA part III}.
3.9 ***Short wheel base:*** From the statistical analysis of derailment on the meter gauge conducted by the efficiency bureau it is seen that the maximum tendency for derailments is in the short wheel base group for 4-wheeler wagons, the incidence being 1.9 times the rest, the ratio being even 2.9 times for 1.0 ft wheel base wagons than those over the rest.
Although the short wheel base has. less disturbing effect on excessive cant gradient, the derailments of such wagons are more due to (i) their having a tendency to hunt more freely (ii) their tendency to take up more angularity with respect to the track than the vehicles with greater wheel base, and (iii) the shorter wheel base wagons being torsionally stiffer than the ones with longer wheel base which are torsionally more flexible and the safety depth (Zpermissible) available is thus less for short wheel base wagons.

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