ज्ञान ज्योति से मार्गदर्शन

To Beam As A Beacon of Knowledge

THE INVESTIGATION OF DERAILMENTS

August 2014

INDIAN RAILWAYS INSTITUTE OF CIVIL ENGINEERING
PUNE 411001

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**INDIAN RAILWAYS INSTITUTE OF CIVIL ENGINEERING,
Pune 411001**

FOREWORD TO THE FOURTH EDITION

It has been an endeavour of IRICEN to publish books containing the latest information on subjects useful for the railway engineers. This book **'The Investigation of Derailments'** published by IRICEN has served the objective for more than three decades. Railway men of various disciplines have immensely benefited from the contents of this book. Third edition of the book was published in 2007. Since then, some new type of rolling stock have been introduced. Also, there have been changes in maintenance practices for rolling stock and track.IRICEN is bringing out this Fourth Edition of the book, incorporating these changes.

I hope that the railway men would find this revised edition a useful source of the required information for a scientific investigation of derailments.

Pune,
August, 2014

Vishwesh Chaube
Director
IRICEN, PUNE

PREFACE TO THE FOURTH EDITION

The first edition of the book 'The Investigation of Derailments' was published in May, 1981 by Indian Railways Institute of Advance Track Technology (Presently known as Indian Railways Institute of Civil Engineering), Pune. Second edition of the book was published in May, 1995, in which a new chapter, incorporating case studies was added. Third edition of the book was published in March, 2007. In this edition, information related to CASNUB bogies, Fiat Bogies and new types of locomotives was incorporated. Subsequent to the publication of the 3rd edition of the book, significant developments have taken place relating to rolling stock running on the Indian Railways network .Some new type of rolling stock have been introduced. Population of some of the newer rolling stock has significantly increased while some of the older rolling stock have significantly reduced in number, or have been totally phased out. Also, there have been changes in the maintenance practice of some of the rolling stock system. Therefore, a need has arisen for revision of the book.

In this edition, some significant additions have been made. These include, Influence of Angle of Attack on the limiting valueof Y/Q and Mechanism of re-distribution of wheel load in a bogie in Chapter 1, 'Theoretical Background'. In Chapter 3, 'Rolling Stock Features and Defects', systems of Air Spring, Hydraulic Shock Absorber, and Bogie Rotation have been added. In Chapter5,'Coaching Stock', various components of ICF All-coil Bogie have been elaborated and description of various components of LHB FIAT Bogie have been added. Chapter 6 on Locomotives has been re-written. Locomotives have been grouped based on bogie design, for the ease of comprehension at the field engineers level. New bogies, namely, High Adhesion bogie, Flexi-coil Mark I bogie, High Tensile Steel Cast (HTSC) GM bogie, Bo-Bo Flexi-coil Fabricated bogie and Co-Co Flexi-coil Fabricated bogie have been added. In Chapter 9, 'Case Studies', some good case studies related to interaction between track – rolling stock defects causing derailment as well as signaling system defects leading to derailment have been added.

I am thankful to the faculty and staff of IRICEN who have contributed immensely to the publication of this book. I am grateful to Director, IRICEN for his encouragement and guidance in bringing out this publication.

Although care has been taken to include details as per authentic and latest reference in this publication, still there may be some errors. I would be thankful to the readers for their suggestions, which may be sent to IRICEN at mail@iricen.gov.in. These would be taken into account while preparing the next version of the book.

Pune
August, 2014

Nilmani
Professor/Track – 1
IRICEN

PREFACE TO THE THIRD EDITION

Second edition of the book **"The Investigation of Derailments"** was reprinted in 2001. It was very popular among field engineers and became out of stock in due course of time.

Therefore, the third revised and enlarged edition has been brought out to fulfill the continuous demand for the book. Information pertaining to new goods and coaching stock has been added in this edition, which will be very useful to field engineers.

Although every effort has been made to bring out latest and present the book in error free manner, yet if there is any suggestion or discrepancy, kindly do write to us.

Shiv Kumar
Director, IRICEN

PREFACE TO SECOND EDITION

Investigation of the factors leading to derailment constitutes by itself a sailent facet of the accident management. The purpose of investigation is not only to pinpoint the cause of derailment but also to document the lessons learnt to avoid recurrence of lapses. This book published by IRICEN served this objective for more than a decade and railway men of all disciplines have been immensely benefited by the contents of this book. With changes in track technology and with the introduction of new types of rolling stock, need had arisen to update the information. I am glad IRICEN is accordingly bringing out the revised edition of 'The Investigation of Derailment', also incorporating case studies as a fresh chapter. I hope this enlarged edition will be even more useful to the Railwaymen of all departments, with the ultimate goal of total prevention of accidents.

Raj Kumar
Member Engineering
Railway Board
New Delhi

FOREWORD TO SECOND EDITION

A derailment costs money, anguish to users and possible loss of life. For preventing its recurrence therefore, no effort is too much.

Expressing in a nutshell, concern of a permanent way man in regard to a derailment is :

- to rescue, repair and restore
- to establish the cause by joint investigation
- to use the resultant information to guide preventive or corrective efforts.

Unless the cause is obvious, e.g. a tree or a boulder fallen on the track, breach, washaway, formation failure, etc., the task of investigating and determining the cause of a derailment is a formidable one, as a large number of variables are involved.

It is recognized that investigative competence cannot alone come from books or teaching, it is mainly derived from the practical field experience. However, the former serves as important adjunct to this total experience by enhancing our knowledge of critical factors associated with derailments. This booklet strives to help achieve this objective.

The booklet has been divided into 6 chapters:

- Theoretical background
- Site investigation
- Rolling stock suspension systems and defects
- Track defects
- Operating features.
- To sum up…

The last chapter, condensing the sailent aspects of derailment mechanics, embodies valuble case studies. This chapter has been added in the edition of 1995, while updating the contents of the other chapters.

S. GOPALAKRISHNAN
DIRECTOR IRICEN
01-05-1995

FOREWORD TO FIRST EDITION

A derailment costs money, anguish to users and possible loss of life. For preventing its recurrence therefore, no effort is too much.

Expressing in a nutshell, concern of a permanent way man in regard to a derailment is:

- to rescue, repair and restore
- to establish the cause by joint investigation
- to use the resultant information to guide preventive or corrective efforts.

Unless the cause is obvious, e.g. a tree or a boulder fallen on the track, breach, washaway, formation failure, etc., the task of investigation and determining the cause of a derailment is a formidable one, as a large number of variables are involved.

It is recognised that investigative competence cannot alone come from books or teaching - it is mainly derived from the practical field experience. However, the former serve as important adjuncts to this total experience by enhancing our knowledge of critical factors associated with derailments. This booklet strives to help achieve this objective.

The booklet has been divided into 5 chapters :

- Theoretical background
- Site investigation
- Rolling stock suspension systems and defects
- Track defects
- Operating features.

Y. G. PATWARDHAN
Principal
Indian Railways Institute of
Advance Track Technology

CONTENTS

CHAPTER 1

THEORETICAL BACKGROUND

1.1 Introduction

Motion of a rail vehicle on track is a complex phenomenon. A large number of factors are at play, having bearing on safety and stability of the movement. These factors are related to track, vehicle and dynamic interaction between them. Due to this complexity, at times, it becomes difficult to establish the cause of derailment.

The fact that a derailment has occurred implies that one or more factors crossed the safety limits during the process. What were the disposition and values of such factors are not known. Even if the same vehicle is moved over the same track, at the same speed under the same operating conditions, it may not derail, for it would be near impossible to simulate all the conditions exactly.

After identification and measurement of all track and rolling stock particulars and defects and operating features, a sound theoretical understanding of the whole phenomenon of vehicle-track interaction helps one to analyse the evidence logically and systematically and to arrive at the probable cause(s) of derailment. An investigator, therefore, has to be well acquainted with the subject of vehicle-track interaction in all its ramifications.

Some of the important theoretical aspects relevant to investigation of derailments are :

- Derailment mechanism
- Vehicle oscillations and resonance effect
- Effect of track and vehicle twist on wheel off-loading
- Lateral stability of track

Of the above, 'Derailment mechanism' forms the central theme.

1.2 Derailment Mechanism

There are two broad categories of derailment

A. *Sudden derailment,* by wheelsets jumping the rails. Such a derailment indicates that the derailing forces were high enough to suddenly force the wheel off the rail.

B. *Derailment by flange climbing* i.e. by wheel mounting the rail in a relatively gradual manner. It indicates that the derailing forces were powerful enough to overcome the normal stabilizing forces, yet not sufficient to cause a sudden derailment.

It is much easier to arrive at the probable cause if the derailment is established to be falling in category A. On the other hand, it may be more difficult to establish the cause in the event of the derailment falling in category B. *To identify the category, therefore, is the first step in derailment investigation* (See Chapter 2).

Whereas a derailment occurring by wheelset jumping hardly requires any theoretical treatment, one by flange climbing does need a deeper theoretical understanding. Discussion in this chapter relates mainly to the category of derailment by classical flange climbing.

1.3 Mechanism of flange climbing derailment and Nadal's formula.

Whatever be the track or vehicle defect or operating feature contributing to derailment, it manifests itself at the rail-wheel contact, because, finally, it is the wheel which mounts the rail.An understanding of what happens at the rail-wheel interface will, obviously, lead to a better appreciation of the manner in which vehicle and track defects and operating features contribute to derailment proneness.

Because of various reasons e.g. wheel tread conicity, track irregularities, elastic characteristics of the track, suspension characteristics of the rolling stock, vehicle loading characteristics, vehicle operation characteristics etc. the wheelset travels along the track exerting a variety of oscillations (see Vehicle oscillations).

1.3.1 Play between wheel and rail and wheel angularity

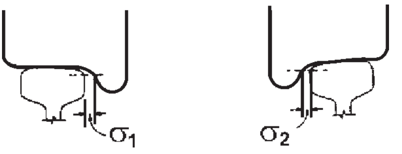
A wheel set cannot have a tight fit with the track gauge, as in such a condition the wheelset will tend to run at the flange slope rather than at the tread, increasing the derailment proneness apart from straining the track fastenings. In other words, there has to be a play between the wheelset and the track. It cannot be too small, otherwise movement of the running wheelset would not get isolated from the lateral track irregularities, resulting in generation of large lateral forces.

The designed play between a standard wheelset and standard track gauge (not Standard Gauge, which is 1435 mm) for a particular gauge, is called the standard play.

Standard play σ_S = Standard track gauge
(1676 mm for B.G.,
1000 mm for M.G.)

- (Standard wheel gauge +2 x standard flange thickness)

(1600mm for B.G.	(28.5 mm for B.G
930 mm for M.G.)	25.5 mm for M.G.)



$$\sigma_s = \sigma_1 + \sigma_2$$

Figure 1.1 Play between wheelset and track

The standard play on B.G. and M.G. on Indian Railways is 19 mm. Owing to tolerances and wear, the actual play can be different from the *standard play.*

For similar reasons, there have to be some lateral and longitudinal clearances at the axle box level.

Due to availability of such play and clearances, the wheelset is able to become angular to the rails. The wheelset, thus, rarely runs exactly parallel to the rails but moves with varying degrees of angularity. Besides, while negotiating a curve or turnout, only one axle in a rigid wheel base, if at all, may attain a radial position and the rest have necessarily to be angular to the rails (**Fig.**1.2)

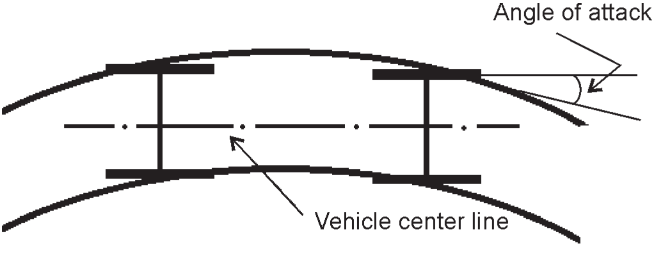


Figure 1.2 Angularity

Normally, a wheel set has a minimum of two and a maximum of three points of contact with the rails. Two of these points are between the wheel tread and rail table top on each of the rails. The third point is located between the flange and the radius of the gauge face of the rail and appears whenever one of the flanges is in action. On the straight track, flange action is intermittent, and on curves continuous.

Due to clearance between the wheel and track gauges and the axle housing and vehicle frame, an axle may assume any intermediate position between those indicated in **Fig.**1.3.

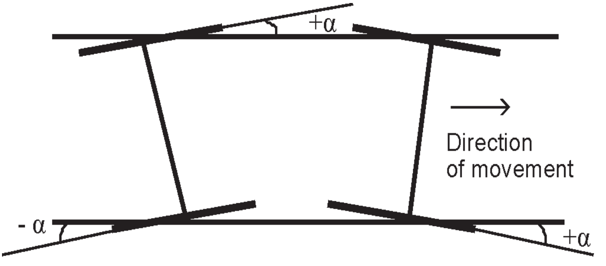


Figure 1.3 Angular position of axle

The vertical plane of the wheel thus runs obliquely to the long axis of the rail and the point of contact may lie ahead or behind the vertical projection of the wheel center on the rail table depending on whether the angularity is positive (flange moving downwards) or negative (flange moving upwards). On curves, wheel obliquity is accentuated in proportion to the ratio of the wheel base of the vehicle to the radius of curvature.

To appreciate the configuration of flange contacts in various positions of angularity, let us cut a horizontal section through the wheel flange at the level of flange contact. The sectional plan will look as shown in **Fig.**1.4.

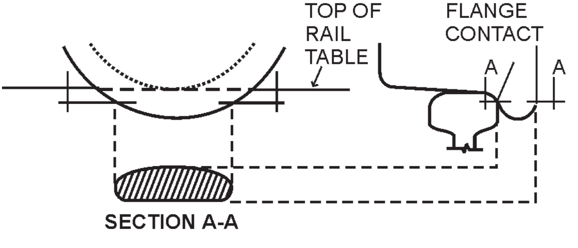


Figure 1.4 Sectional plan of wheel flange at the level of flange-to-rail contact

As the angularity of the axle changes, the flange contact can be ascertained by accordingly aligning the above sectional plan.

In general, there are 3 basic configurations of the axle:

- Zero angularity
- Positive angularity
- Negative angularity

For ascertaining whether the particular angularity of an axle is positive or negative, it is necessary to know the direction of motion and whether the flange contact is with left rail or right rail (as seen in the direction of movement).

Zero angularity

In this, the wheelset is parallel to the rails and thus angularity with the rails is zero. This configuration, along with the sectional plan of wheel flange at the level of flange contact, is shown in **Fig.**1.5.

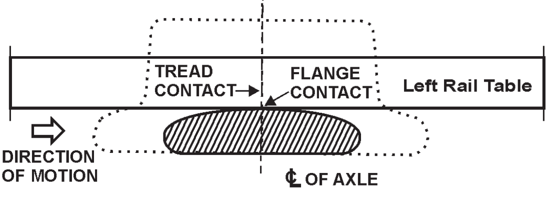


Figure 1.5 Zero angularity (Plan)

The position of tread contact and flange contact are shown. It will be seen that the longitudinal eccentricity between the two is zero.

Positive angularity

In this the wheelset is angular to the rail such that the wheel makes flange contact near its leading edge, as shown in **Fig.**1.6

The flange contact leads the tread contact. It is called a case of leading contact, the longitudinal distance between the tread and flange contacts being called positive eccentricity. Such angularity is called positive angularity, the angle between the wheel alignment and the rail in this case is called positive angle of attack (say **α**+)

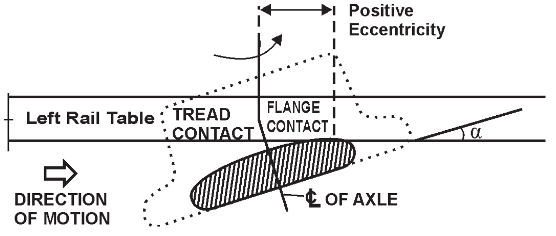


Fig.1.6 Positive angularity (Plan)

Negative angularity

In this, the wheelset makes flange contact near its trailing edge as shown in **Fig.**1.7.

The flange contact trails the tread contact. It is a case of trailing contact, the longitudinal distance between the two contacts is called negative eccentricity.

This configuration is called negative angularity and the angle between the wheel alignment and the rail in this case is called negative angle of attack (say **α**-).

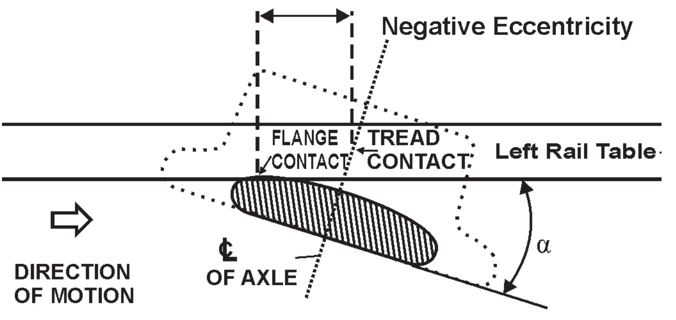


Figure 1.7 Negative angularity (Plan)

1.3.2 Safety depth

Increasing angularity of a wheel shifts the side contact of the flange with rail gauge face down over the root radius of the flange and, also, over the conical part (**Fig.** 1.8(a)). Safety depth is the amount Zc a wheel can lift off the rail table before inviting certain derailment. It is the vertical distance between the position of the point of actual contact between flange and rail gauge face of an oblique wheel corresponding to the position of Two point contact (contact both at rail table top and gauge face) , and the position of the point on the bottom of the conical part of flange, lying vertically below. (**Fig.**1.8 (b)). As seen in the figure, safety depth reduces with increasing angle of attack.

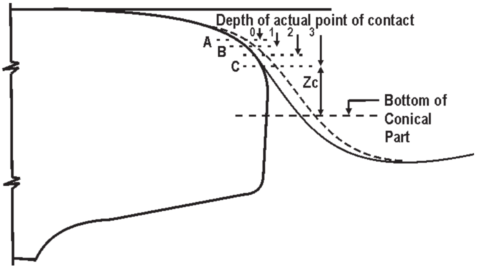


Figure 1.8 (a) Shift in flange contact with angularity 2+

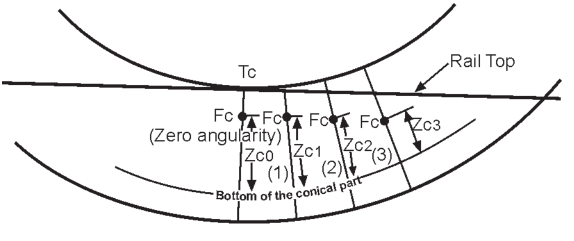


Figure 1.8 (b) Safety depth Zc reduces with increase in angularity ($Zc_3 < Zc_2 < Zc_1 < Zc_0$)

1.3.3 Frictional force

Frictional force is generated when two surfaces pressed against each other (i,e, there is a normal reaction between them), slide or have a tendency to slide. Direction of this force is opposite the relative movement and its magnitude is given by

$$f = \mu * N$$

where N is the normal reaction and μ is the co-efficient of friction. Let us find out the direction of the frictional force when wheel flange comes in contact with the rail gauge face. First , consider the case of positive angularity. A side elevation of the wheel is shown in **Fig.**1.9

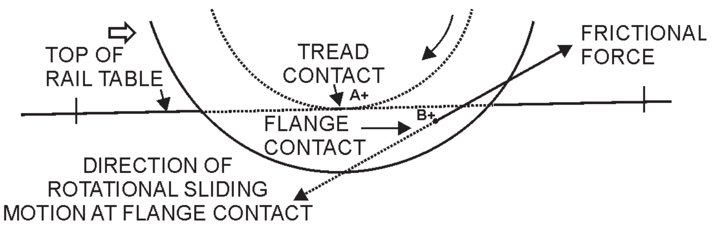


Figure 1.9 Positive angularity (elevation)

Point A+ is the tread contact and B+ is the flange contact. As frictional force acts opposite to direction of motion at the surface of contact, the direction of motion of the wheel at the flange contact has to be established first.

The wheel is undergoing translatory movement, which is predominantly in the longitudinal direction, as well as rotational movement. The resultant velocity at any point on the wheel would be a resultant of the velocities due to these two motions. This can be determined by vector addition of the two velocities.

Under normal run, there is practically no slip at the contact between rail table top and wheel tread. Therefore,

$$v = \omega * r$$

Where, v is the longitudinal speed of wheel (centre), ω is the rotational speed, r is the wheel radius at the tread – rail top contact. With v and ω related as above, the resultant linear velocity, both in magnitude and direction, can be determined at any othet point on the wheel.

As there is no sliding movement at wheel tread – rail top contact point, this point is stationary, though momentarily. Hence, another approach could be to analyse the motion of wheel considering this point as the instantaneous centre of rotation. Now the direction of motion of the wheel at point B+ will be along a line tangential to the circle drawn through B+, with A+ as the centre. The tangent is directed downward as shown by the dotted arrow in **Fig.**1.9. Consequently, the frictional force at B+ acts upwards (as shown by full line arrows) opposite to the direction of motion.

Putting it simply, in configuration of positive angularity, the wheel flange rubs against the rail head edge in a downward arcing motion (also referred to as the downward biting action of the wheel flange), resulting in frictional force acting upward.

Even though the wheel as a whole may be moving upward in the process of derailment, frictional force also acts upwards. Paradoxical though this may seem; it would be clear, if one imagines as to how a car wheel goes over a small obstruction, say, a speed braker across the path.

The wheel rotation (or tendency thereof) causes a downward biting action into thebraker, resulting in frictional force acting upwards, which assists take the wheel over the obstruction. If this friction were reduced by, say, excessive oil spill over the obstruction, one would agree, it would be more difficult for the wheel to go over it.

Another example is a person walking forward. The frictional force between the foot and the ground acts forward as the direction of motion of the foot at its contact with the ground tends to be backward. In fact it is this frictional force which enables a person to walk.

Proceeding in a similar manner, it will be clear that in the case of negative angularity, or trailing contact, the frictional force will be directed downwards as shown in **Fig.**1.10.

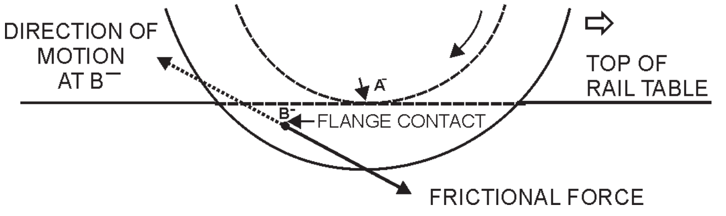


Figure 1.10 Negative angularity (elevation)

In the case of zero angularity, the frictional force acts horizontally as shown in **Fig.**1.11. This frictional force apparently has no component in the vertical direction.

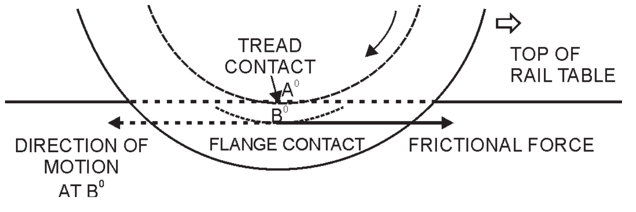


Figure 1.11 Zero angularity (elevation)

From the foregoing, it is seen that the frictional force acts upwards and, hence, functions as one of the derailing forces only in case of the positive angularity configuration. In other words, derailments proneness is higher when the wheel makes flange contact with positive angle of attack. Positive angularity is, therefore, the most critical of the three conditions of wheel angularity. Study of forces in this condition only is, therefore, relevant. On a straight track, positive angularity configuration may occur only during certain periods of the motion whereas on sharp curves, it may occur more or less throughout the period of curve negotiation. On a straight track all the wheelset positions as shown in **Fig.**1.12. fall under the category of positive angularity.

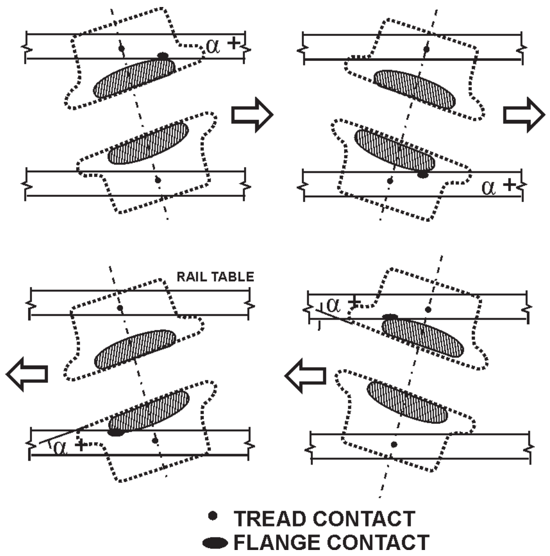


Figure 1.12 Wheelset configurations with positive angularity.

1.3.4 Mechanism of Wheel Climbing Derailment

The classical case of derailment is by wheel flange gradually mounting the rail and derailing. Let us first analyse the process qualitatively. The process is illustrated in **Fig.** 1.13.

In this figure, stage I shows the normal (ideal) position, where a wheel runs on its tread, with some clearance between the rail gauge face and wheel flange. Under the influence of features such as sharp curve, track irregularity, bogie defects, uneven loading etc., a large lateral force (Y) may be generated, which may bring wheel flange in contact with rail gauge face. This condition is shown as stage II in the figure.

In case the dispensation of forces is adverse, such as a very large lateral force or a very small vertical load, wheel may start

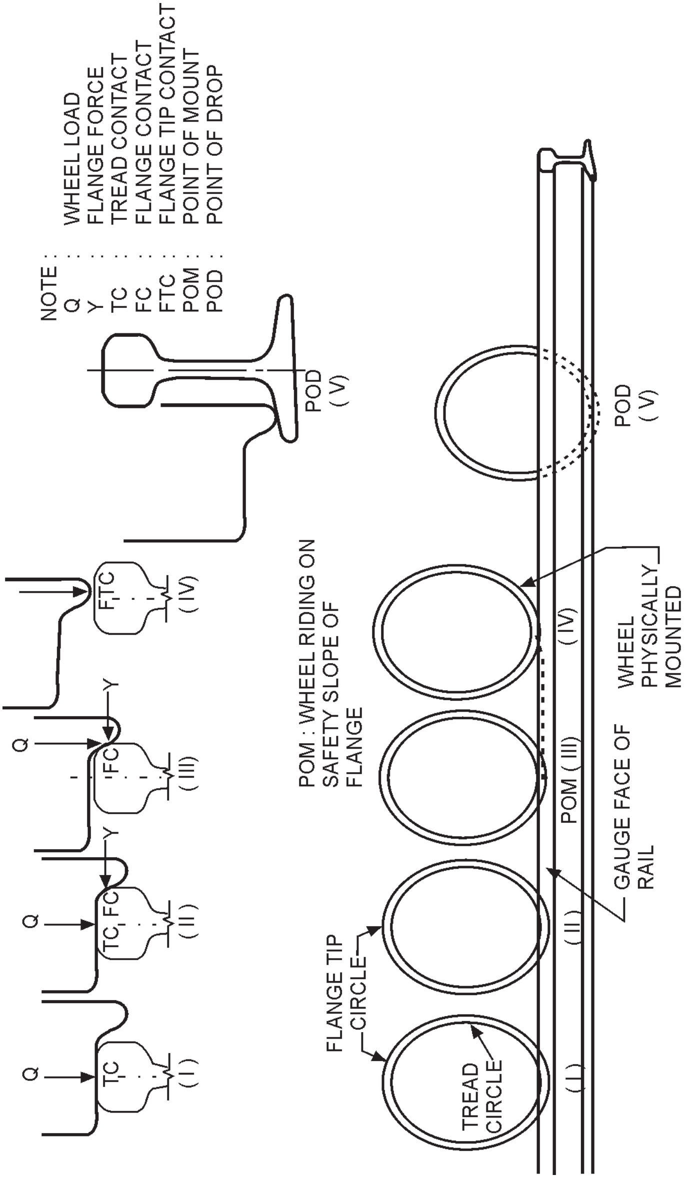


Figure 1.13 Derailment by Gradual mounting of wheel

moving in an upward direction, resulting into loss of contact between wheel tread and rail table top, exhibited as stage III. With loss of contact at rail table top, only contact left is that between wheel flange and rail gauge face. In this condition, derailment is incipient. The location where this condition gets initiated is called Point of Mount (POM). A very large stress gets generated at the point of contact, resulting into formation of a scratch mark along its path of travel.

If the adverse disposition of forces continues for a sufficiently long duration, wheel flange may reach top of rail, shown as stage IV in the figure. In this position, derailment is almost certain, as there is very little resistance to further lateral movement of wheel. Location where wheel finally drops off the rail is called Point of Drop (POD), shown as stage V in the figure.

1.3.5 NADAL'S Formula

Let us now do the quantitative analysis of the process of derailment by gradual climbing of wheel. Stability of a single wheel is analysed. An assumption is made that track is rigid, implying that it does not move. The disposition of forces on the single wheel corresponding to the configuration of positive wheel angularity and under the condition of incipient derailment is shown in Figure 1.14.

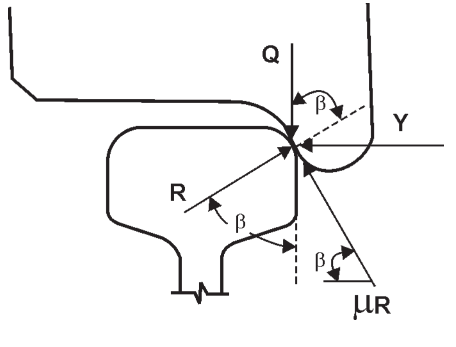


Fig.1.14. Forces at rail-wheel contact with positive angularity

Forces acting on the wheel are Vertical instantaneous wheel load, Q, and Lateral force Y. There are two other reactive forces viz. normal reaction R from the rail, acting along the normal to flange slope, and frictional force between wheel flange and rail, acting along the flange slope.

Considering that no movement is possible in the direction normal to flange slope,

R = Normal reaction from rail

= (Q cos β + Y sin β)

Frictional force, acting upward = μR

(μ being the coefficient of friction

between wheel flange and rail gauge face)

N.B. : It may be noted that the frictional force,μR to be considered is only the component, in the vertical plane across the track, of the 3 dimensional frictional force shown in **Fig.**1.9.

β = flange angle

Condition for safety against derailment:

Derailing forces < Stabilizing forces.

Resolving the forces along the flange slope (i.e. at angle β with the horizontal), the above condition implies that

$$Y \cos \beta + \mu (Q \cos \beta + Y \sin \beta) < Q \sin \beta$$

i.e. for safety against derailment:

$$\frac{Y}{Q} \ngtr \frac{\tan \beta - \mu}{1 + \mu \tan\beta}$$

\This is the well known Nadal's formula, which he enunciated in the year 1908. Y/Q is called derailment coefficient. Though many later authors have attempted to refine Nadals formula, due to its simplicity, it is the most widely used formula in investigation of derailments. The Formula provides a very conservative value for Y/Q ratio and, hence, is appropriate given the significance of safety.

Nadals formula also provides an important criterion for assessment of stability of rolling stock. However, the difference between the two applications of the formula, viz.

- in investigation of derailments, and
- as one of the criteria for assessment of stability of rolling stock

should be clearly understood.

1.3.6 (Y/Q) ratio and shift in Point of Contact

The moment a wheel attempts to part company with the rail the contact between it and the head is first broken and the remaining point begins to travel along the profile of the flange provided the lateral force to vertical force ratio is high enough to maintain the progressive lift.

The motion proceeds in the following three phases and the lateral force Y required (for a particular value of Q) is depicted diagrammatically in **Fig.** 1.15.

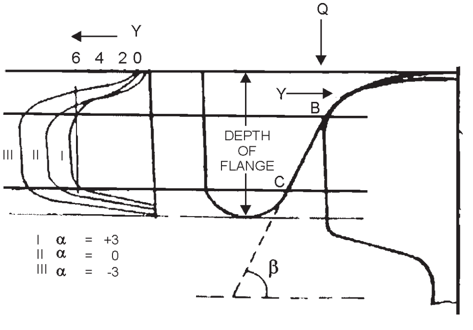


Figure 1.15 Variation of Y

1st Phase : The point of contact travels down the root radius of the profile to its friction with the conical point at B, the initial lateral force is small but rises quite steeply.

2nd Phase : The point of contact moves down the conical part of the profile from B to C, the lateral force reaches its maximum value and remains more or less constant.

3rd Phase : The point of contact rounds the radius of the crown of the flange from C onwards, the lateral force drops to zero and derailment is certain.

The zone of maximum flange force is thus the conical part of the flange (B to C) and the safe limit of shift of the point of contact is at C.

1.4 APPLICATION OF NADAL'S FORMULA IN DERAILMENT INVESTIGATION.

It will be seen from Nadals formula that, for safety to be ensured, either the left hand side i.e. $\frac{Y}{Q}$ should be smaller or the right hand side should be larger. This will happen when, in a comparative manner,

- Y is low
- Q is high
- μ is low.

With regard to the flange angle β, the issue is more complex. How the right hand side viz. the expression

$$\frac{\tan \beta - \mu}{1 + \mu \tan \beta}$$

varies with change in μ and β will be clear from **Fig.**1.16

The formula thus indicates that for safety, β should be large. Its maximum value could be 90^0. But in such a case, even with a minute angularity of the axle, the flange contact would shift to near the tip of the flange, the safety depth of flange would be almost zero and derailment proneness would increase. **Fig.**1.17 illustrates the point.

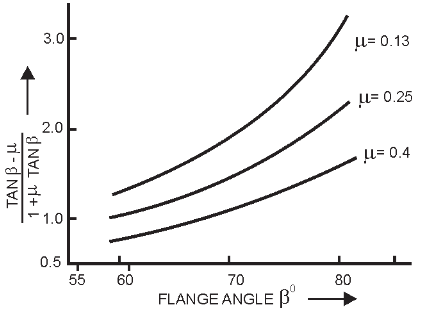


Figure 1.16

For this very reason, a flange slope is provided such that within the possible range of angularity of axle, the flange contact remains away from the flange tip. In fact, greater the angularity occurring in vehicle movement, less should be the flange angle. But there is a limit to it, as this criterion runs counter to what is indicated by Nadals formula. On the Indian Railways, a flange slope of 2.5 in 1 (tan **β** = 2.5 or **β** = 68^0-12') is adopted for a new wheel profile of carriage and wagon. For wheels of diesel and electric locomotives as well as some of the wheels of steam locomotives, the flange angle earlier provided was 60^0. However, the flange angle now provided is 70^0 on all BG diesel and electric locomotives operating Mail/Express trains with a maximum speed of 110 km/h, based on oscillation trials and service trials. But, for speeds higher than 110 km/h, the earlier flange angle of 60^0 is retained as oscillation trials and service trials with a flange angle of 70^0 for speeds higher than 110 km/h are yet to be conducted.

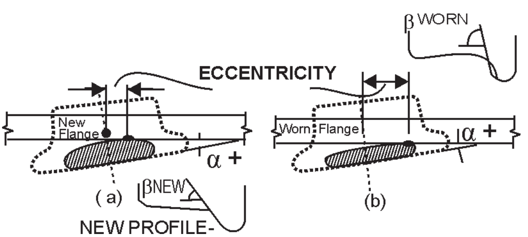


Figure 1.17 Increase in eccentricity with increase in β

With wear, β only increases and never reduces. But at the same time, wear causes the flange to become sharper which increases the value of μ (coefficient of friction) through change in the geometry of the surfaces in contact and greater biting action and, thus, reduces safety. Besides, as discussed, larger β causes the eccentricity to increase and the safety depth to reduce with even minute values of angularity. Thus, whatever advantage is indicated with increase in value of β in Nadals formula, is more or less off-set by the adverse effects of such increase. The factor β, therefore, is left out of further discussion.

A larger positive eccentricity implies that flange contact is nearer to the flange tip. It increases derailment proneness on two accounts – first, direction of frictional force is such that its vertical component (upward) is larger, second, flange safety depth is smaller.

On this account, effective μ (i.e.μ corresponding to component of frictional force in the vertical plane) increases with increasing angle of attack up to a certain limit. According to Prof. Heumann, for angle of attack

$\alpha = 0$, effective $\mu = 0$

and for $\alpha > 0.02$ radians, effective $\mu = 0.27$.

For α between 0 and 0.02 radians, there is gradual increase in value of effective μ from 0 to 0.27.

Since derailment proneness becomes appreciable only with positive angle of attack, a condition in which axle moves with a persistently positive angularity is more prone to derailment than that in which positive angularity of the axle is only intermittent.

It is of importance to note that during derailment investigation, there are no means what-so-ever of knowing the values of Y,Q, μ, eccentricity and the angle of attack at the instant of derailment. No attempt should, therefore, be made to apply Nadals formula in terms of quantitative values of Y, Q, μ, β, ratio Y/Q, eccentricity, or angle of attack. The contribution of track and vehicle defects and operating features, towards derailment should be assessed only qualitatively, by studying the magnitude of the defects and analyzing the relative extents to which nature of such defects contribute towards derailment proneness.

Thus, while investigating into a derailment, all track and vehicle defects and features and operational aspects, which can be shown to cause one or more of the following to occur, viz. :

- Y to increase
- Q to decrease
- μ to increase
- positive angle of attack to increase
- positive eccentricity to increase
- persistent angular running of the axle

should be listed as possible contributory causes. The list of such contributory defects and features thus arrived at, should then be arranged in a descending order of their assessed contribution. This will enable one, finally, to arrive at the most probable cause or causes of the derailment.

How various rolling stock and track defects and operating features contribute towards derailment proneness through one or more of the above six effects, has been detailed in Chapter 3 to 8. However, some general defects and features are discussed below (Defects and features affecting Y and Q are large in number. They have been separately discussed at appropriate places).

1.4.1 DEFECTS AND FEATURES THAT AFFECT μ

Following contribute towards increasing μ and, hence, the derailment proneness

1.4.1.1 Rusted rail

This feature becomes particularly critical if occurring on curves or turnouts. For instance, if new rails received for rail renewal on a curve have been lying on the cess for quite some time, they would get somewhat rusted. When put in the track, such rails would cause high value of μ to occur at the wheel flange-rail contact.

Another example is emergency crossover. Because of infrequent use, such crossover gets rusted, increasing the derailment proneness at the time of vehicle movement.

It is a good practice, therefore, to lubricate the gauge faces of the rail in curves and turnouts as a regular maintenance practice. Care should be taken not to lubricate the rail table as this would cause adhesion to drop and the wheels to slip or skid.

1.4.1.2 Newly turned wheel

When a worn wheel is reprofiled, the wheel and flange carry the tool marks, which get polished off only after some period of running. But during the initial stages such tool marks cause a high coefficient of friction to occur, thereby increasing the derailment proneness. Coupled with persistent angularity on curves it can reduce safety considerably. There have been practical instances of derailments, the cause for which could be traced to this aspect.

1.4.1.3 Sanding of rails

On steep gradients sanding is at times resorted to, to develop the requisite adhesion. Normally, steep gradients exist in situations involving sharp curves, for example, ghat sections. As such the increased coefficient of friction due to possibility of sand particles coming between the flange and the rail, coupled with the positive angularity on curves, increases the derailment proneness appreciably.

1.4.1.4 Sharp flange

When the radius of flange tip is reduced to less than 5 mm, the condition is called a sharp flange (as illustrated in **Fig.**1.18).

In the condition of positive angularity, it is the sharp tip which rubs against the rail head edge. The sharpness of the tip increases the biting action against the rail and, thus, increases the effective μ, thereby, increasing the derailment proneness.

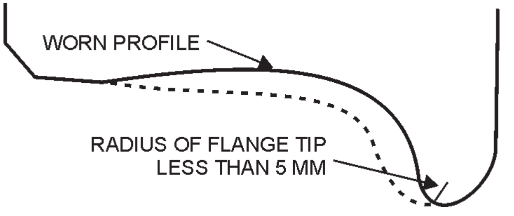


Figure 1.18 Sharp flange

1.4.1.5 Increased positive angle of attack

As discussed earlier, up to a certain limit, effective **μ** increases with positive angularity. Conditions under which positive angularity increases are :

(a) Increased play between wheelset and track, which may be owing to one or more of the following

(i) Excessive slack gauge

(ii) Thin flange (i.e.a flange worn such that the thickness is reduced to less than 16mm) [see chapter 3]

(iii) Excessive clearances between axle box and bogie frame [see Chapter 3]

(b) Sharp curves and turnouts

(c) On a given curve, the outer axles of a multiaxle rigid wheel base experience greater angularity as compared to intermediate axle.

Item (i) and (ii) above determine the maximum possible angularity of a wheelset with respect to track, whereas item (iii) decides its maximum angular position compared to the rigid wheel base.

1.4.2 DEFECTS AND FEATURES WHICH INCREASE POSITIVE ECCENTRICITY

As brought out earlier, the positive eccentricity increases primarily through the wheel flange slope becoming steeper. In this situation the flange contact, even with the same angle of attack, moves nearer to the flange tip as compared to the condition in which the flange slope is shallower. This condition is caused by wheel profile defect of 'Worn Root' .

1.4.3 DEFECTS AND FEATURES WHICH CAUSE PERSISTENT ANGULAR RUNNING OF AXLE

1.4.3.1 Difference in wheel diameters on the same axle

A difference of 0.5 mm only is permitted between the wheel diameters on the same axle for new/ turned wheel(wheel diameter is measured at a location 63.5 mm from the back of the wheel in the case of B.G. wheel and at 57mm in the case of M.G. wheel; (see **Fig.**1.19)

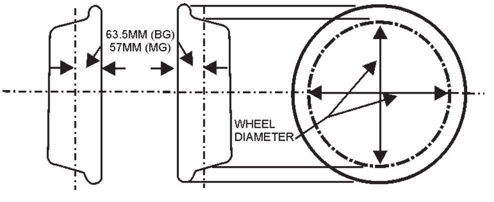


Figure1.19. Measurement of wheel diameters on the same axle

If the difference in wheel diameters is more, it will cause the axle to run eccentric, and for larger difference, persistently angular to track as the larger diameter will always try to travel a longer distance than the smaller one in a given number of revolutions.

1.4.3.2 Incorrect centralization and adjustment of brake rigging and brake blocks

The above defects in the brake system result in unequal brake pressure on the two wheels of same axle. Thus the motion of one wheel is impeded compared to the other and, as a result, the axle turns and continues with a persistently angular motion. Obviously, this will occur only during the braking operation unless the brakes have jammed, when this condition would occur throughout the vehicle run.

1.4.3.3 Hot axle

Hot axle is a condition in which owing to defects of bearing, friction between bearing and journal increases considerably producing appreciable heat, causing the axle to become hot and even to soften and fail structurally.

These conditions result in one end of the axle being restrained more than other in its rotation and, as a result thereof, the axle runs in a persistently angular motion.

1.4.3.4 Longitudinal twist of bogie frame

A bogie frame twisted in plan would result in loss of parallelism between wheel and rail. This would cause persistent angular run.

1.4.3.5 Other rolling stock defects

Rolling stock defects which result in eccentric application of longitudinal forces on vehicle body/ bogie would result in generation of turning moment (in plan). This would cause persistent angular run.

These defects are of buffer, anchor link, traction rod etc., which are discussed in subsequent chapters.

1.4.3.6 Sharp curves

Sharp curves result in higher angularity by virtue of its geometry. Magnitude of angularity is larger for sharper curves and higher rigid wheel base. System of bogie rotation is provided to reduce this angularity. Defects therein would result in persistent angular run.

1.5 STABILITY ANALYSIS BY RAIL-WHEEL INTERACTION FORCES

It is possible to analyse the critical condition of derailment, by considering the forces at rail-wheel level. Formulae evolved by the researchers Nadal and Chartet come under this category.

For assessing the stability of a particular rolling stock, in relation to above criterion, we can instrument the vehicle to measure and record continuously the values of flange force Y and wheel load Q during the test run.

N.B. Strictly speaking, Y and Q have to be measured at the rail wheel contact. This is possible only with a measuring wheel i.e. a specially designed wheel suitably strain-gauged and fitted to the vehicle being tested. A measuring wheel has since been developed on the Indian Railways.

Presently, lateral force Hy at axle box level and load Q at bearing spring level are measured instead of Y and Q at rail wheel contact. Hy is measured by placing a load cell between the axle end and the axle box adapter. Q is measured by measuring the spring deflections (by means of LVDTs viz. linear variable differential transducers) which, when multiplied by the spring constant (spring constant is load per unit deflection of the spring), gives the load Q. Since in this case, Y (or Hy) and Q can be measured, Nadals formula can be applied quantitatively.

Coming back to the Nadals formula, for safety against derailment,

$$\frac{Y}{Q} \ngtr \frac{\tan\beta - \mu}{1 + \mu\tan\beta}$$

For majority of wheels on Indian Railways, flange slope, β is approximately 68 degree.

Value of co-efficient of friction, μ depends on geometry of the surfaces in contact, condition and roughness of the surfaces, etc.

Following are the typical values of μ :

Dry rail	0.33
Wet rail	0.25
Lubricated rail	0.13
Rusted rail	0.6, or even higher

For analytical treatment of such problems involving steel to steel contact, 0.25 or 0.27 is adopted universally as the value of μ. On Indian Railways, μ in general, is taken as 0.25.

Thus, for $\beta = 68^0$ and $\mu = 0.25$,

Right Hand Side of Nadal equation works out to 1.4. In other words, for safety against derailment, ratio Y/Q should not exceed 1.4. This is the threshold value. To allow for certain margin or factor of safety, a limiting value of 1.0 for ratio Y/Q has been laid down on the Indian Railways, as one of the criteria for assessment of rolling stock stability, i.e.

$$\frac{Y}{Q} \text{ or } \frac{Hy}{Q} < 1.0$$

This criterion, however, has to be qualified by a time factor. As stated in the beginning, there are two broad categories of derailments:

- Sudden derailment by wheelsets jumping off the rails,

- Gradual derailment by flange climbing.

Nadal's formula deals with the second category viz. derailment by gradual flange climbing. Although Nadal didn't specify the time required for gradual climbing of rail to be complete, ratio Y/Q has to remain above the critical value for a certain minimum duration of time, for a flange climbing derailment to occur. Further, a higher Y/Q ratio would be needed to cause a derailment if the duration for which it acts is less than the mentioned minimum. Research on the world railways has shown 0.05 sec as the time duration, which approximately delineates the boundary between the two categories.

On the Japanese National Railways, limiting values for Y/Q ratio are :

(i) 0.8 - when duration t, for which Y/Q ratio acts is 0.05 sec or more (1-4 a)

(ii) 0.4/t - when duration t, in sec, for which Y/Q ratio acts is less than 0.05 sec (1-4 b)

The above expressions are illustrated (approximately) in **Fig.**1.20.

Based on research and considerable experience in on-track testing of freight vehicles, a 0.05 sec time duration has been adopted by the American Association of Railroads (AAR) for certification testing of new freight vehicles.

Final form of the criterion adopted on the Indian Railways is :
'the *derailment coefficient* Y/Q or Hy/Q

should not exceed 1.0, the said coefficient being measured over a duration of 0.05 second.

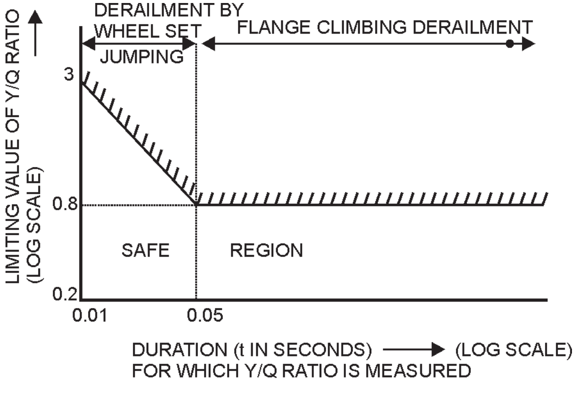


Figure 1.20 Limiting values of Y/Q ratio vs time duration

1.5.1 CHARTET'S FORMULA

While analyzing stability, Nadal did not consider the influence of the other wheel of the wheel set on the subject wheel. This influence was considered by Chartet, resulting into refinement in Nadal's formula.

For analyzing the equilibrium of forces at the moment of incipient derailment, Chartet considered the force pattern at the non-derailing wheel of the same axle, in addition to that at the derailing wheel. When the derailing wheel moves laterally, the non derailing wheel too will move so, both being integrally connected. This would cause the wheel tread of the non-derailing wheel to slide over the rail head, thus generating frictional force which would act in a direction opposite to the direction of movement of the wheel.

The direction of this frictional force, too, would appear to be a paradox. To establish this we have to ascertain the direction of liding movement (**Fig.**1.21). Since critical condition is created only with positive angle of attack, only this configuration has been considered.

With positive angularity, left hand wheel (wheel 1) would have travelled in the direction along A1B1 due to rolling. However, as explained in earlier paras, sliding takes place at the wheel flange – rail gauge face contact, and as a result, resultant lateral movement of the wheel may be zero (wheel not mounting) or very small (wheel mounting). As the right wheel (wheel 2) is integrally connected, it also undergoes the same amount of resultant lateral movement. As wheel 2 travels along A2B2 due to rolling of wheel tread on rail top, it is concluded that rolling is accompanied by lateral sliding (along B2C2) on the rail table. This lateral sliding is also called as lateral creep.

As a result of the sliding movement, the tread frictional force at the non-derailing wheel acts in a direction opposite to this movement as shown in **Fig.**1.21, Thus, this force acts in the same direction as lateral force Y does on the derailing wheel, increasing the derailing forces in the process. The stabilizing force viz. Q has to be correspondingly increased, if safety is to be ensured.

It implies that apart from ensuring that the derailment coefficient Y/Q does not exceed the limiting value, Q itself should not drop below a certain minimum value.

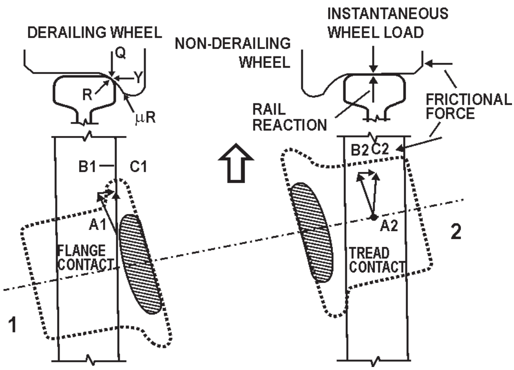


Figure 1.21 Direction of sliding friction at tread of non-derailing wheel.

Note :

Due to vertical oscillations and track and wheel defects, such as track twist and spring defect, wheel load keeps changing. Different wheel loads are defined as below

Nominal wheel load: It is half the axle load as obtained when defect-free vehicle with non-eccentric loading stands on level track with perfect geometry.

Instantaneous wheel load: It is the actual wheel load at any given instant of time during the motion of a wheelset. It constantly varies with time.

On-loading of wheel: When the instantaneous wheel load is greater than the nominal wheel load.

Off-loading of wheel: When the instantaneous wheel load is less than the nominal wheel load.

Chartet has given the following expression, to ensure safety:

$$\frac{Y}{Q} \ngtr K_1 - K_2 \frac{Q_0}{Q} \qquad (1\text{-}5)$$

Where,

Y = flange force at the derailing wheel

Q = instantaneous wheel load at the derailing wheel

Q_0 = nominal wheel load

$$K_1 = \frac{\tan\beta - \mu}{1 + \mu\tan\beta} + \mu' + \gamma \qquad (1\text{-}5\ a)$$

$$K_2 = 2(\mu' + \gamma) \qquad (1\text{-}5\ b)$$

μ' = Coefficient of tread friction at the rail-wheel contact of non-derailing wheel.

(**N.B**. : Due to different geometry of surfaces in contact, μ' is different from flange friction μ, $\mu' \cong \sqrt{2}\,\mu$)

γ = tangent of the angle of coning of wheel tread.

(for a new wheel profile with 1 in 20 coning, $\gamma = 1/20 = 0.05$)

For general values of μ and μ' (for $\mu = 0.2$)

$$K_1 \cong 2$$

$$K_2 \cong 0.7$$

For safety,

$(Y/Q) < 2 - 0.7\ Q_0/Q$

Or, $2Q > Y + 0.7\ Q_0$

A careful examination of the above expression would indicate, that even when Y becomes very small ($Y \cong 0$, a condition which would prevail at low speed), the condition for safety would be :

$$2Q > 0.7Q_0$$

$$\text{Or, } Q > 0.35Q_0 \qquad (1\text{-}6)$$

i.e. the instantaneous wheel load should not drop below a value equal to 35% of the nominal wheel load, or the wheel offloading should not be more than 65%.

The foregoing analysis in quantitative terms has application, as one of the criteria for assessment of stability of rolling stock. Allowing a margin of safety, the instrumented value of Q should, in general, be not less than 60% of the nominal wheel load i.e.

$Q > 0.60\ Q_0$ (against minimum value of $0.35\ Q_0$) (1-6a)

Or, wheel offloading < 40% (against maximum value of 65%) (1-6b)

UIC (ORE) B-55 Specialist Committee studied prevention of derailment of goods stock on distorted track. The Committee carried out derailment tests on curved tracks (R=150m) with 10mm/m cant gradient provided at the exit of the curve, combined with other adverse conditions for safety, at slow speed. Measurement of wheel off-loading corresponding to this derailment condition was analysed and based on the results, the limiting value of wheel off-loading was fixed as 60% of the wheel load.

1.5.2 FURTHER DEVELOPMENTS

As discussed above, angle of attack (α) contributes to lateral creepage through a component of the wheelset's rotational velocity. If the wheelset has a lateral velocity, in addition to the component of lateral velocity due to its rotation, the net lateral velocity of the wheelset at the contact zone would be resultant of the two. Thus, if the wheelset is moving towards flange contact with a positive angle of attack, the lateral velocity tends to reduce the 'Effective Angle of Attack'. Further, the force associated with lateral creepage are proportional to the Effective Angle of Attack.

A wheelset may have a large lateral velocity under the influence of the derailing lateral forces. At small angle of attack, say upto 5 mrad, the Effective Angle of Attack would be negative. This would result in generation of a lateral creep force opposing the derailment on the non-flanging wheel.

Therefore, criteria as per Nadal's equation is conservative at small Angles of Attack.

A TTCI flange climb criterion has been developed for North American freight cars using AARIB wheel profile with a 75^0 flange angle at speeds below 80 km/h in curving. This criterion encompasses two limits , the single wheel L/V limit and L/V distance limit (here, Y is denoted as L and Q as V). The distance limit is the maximum distance that the single wheel L/V limit can be exceeded without risk of flange climb derailment. It is possibly the first time that the wheelset angle of attack has explicitly been included in the flange climb criterion. **Fig.** 1.22. shows the simulation results of the L/V distance limits under different wheelset angle of attack. Test and simulation results show that the distance limit is a function of wheelset angle of attack.

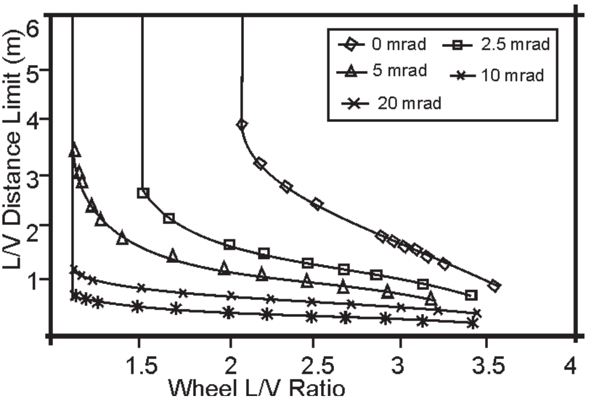


Figure 1.22

1.6 TRACK-TRAIN DYNAMICS AND ITS RELATION TO RAIL-WHEEL INTERACTION

Normally, it is the consist of train, and not an individual rolling stock, which runs on track. Features of this consist of train, such as their coupling arrangement, system of traction and braking etc., and the operating features would have a very prominent effect on rail wheel interaction. Therefore, these features of rolling stock and their interaction with features of track (geometry and alignment) must be studied for a comprehensive analysis of the theory of derailment.

Now, we may go into certain specific theoretical aspects of interaction between vehicle and track. These are

- Vehicle oscillations
- Self-excited oscillations effect of wheel conicity; critical speed
- Effect of cyclic track irregularities and resonance.
- Effect of track or vehicle twist on wheel offloading and related factors
- Lateral stability of track

1.7 VEHICLE OSCILLATIONS

A vehicle, while travelling over track, does not move smoothly but, due to various reasons, executes a variety of oscillations. These are called parasitic oscillations, as, like parasites, they feed themselves on the forward motion of the vehicle on track i.e. their energy is derived from the energy of forward motion of the vehicle. The moment the vehicle stops, the oscillations also cease. These oscillations affect Y and Q values.

For convenience of reference and analysis, it is customary to classify these oscillations according to the 3-axis coordinate system. In reference to any axis, an oscillation can be linear or rotational.

Considering the 3 axes, viz.

X axis : along the track

Y axis : lateral to the track

Z axis : vertical direction

and that there are 2 modes pertaining to each axis viz. linear and rotational, there are in all 6 modes of oscillation, as illustrated in **Fig.** 1.23.

Axis	Mode of oscilliation	
	Linear	Rotational
X	Shuttling	Rolling
Y	Lurching	Pitching
Z	Bouncing	Nosing (also called 'yaw')

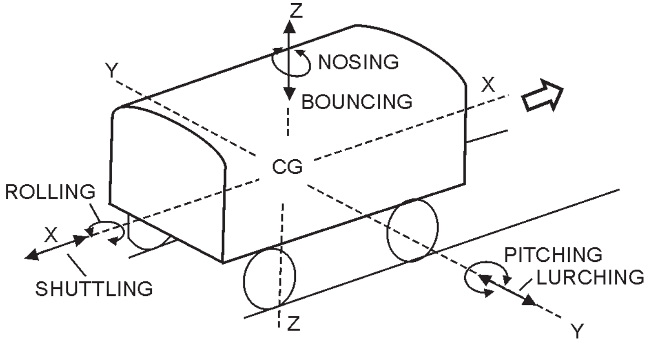


Figure 1.23 The six modes of oscillation

1.8 SELF-EXCITED OSCILLATIONS AND EFFECT OF WHEEL CONICITY

Since the inception of railway, concept of coning had been introduced to provide a simple guidance system for wheelset, requiring a minimum use of flanges.

The wheel tread conocity results in change in rolling diameter of wheel in contact with rail table whenever the wheel moves laterally.

Due to symmetry of the wheel set with respect to track, the rolling diameter of both left and right wheel are equal when wheel set is central to track. Now, let us consider a case where the wheel set has been displaced to the right of the central position. Here, right wheel rolling diameter (D_R) would be larger than left wheel rolling diameter (D_L). As the railway wheel set consists of two wheels rigidly connected, both the wheel undergo equal rotation. Therefore, the right wheel would travel a longer distance than the left wheel, due to pure rolling movement. Differential movement of the wheels results in the wheel set (axle) acquiring an angular configuration with respect to track. This results in movement of the wheel set along a curved path, and towards the center of track.

Due to the angular configuration, the wheel set continues moving to the left even after it becomes central to track (and rolling diameters become equal). As the wheel set moves further to the left of the central position , a similar mechanism sets in, whereby, due to the difference in rolling diameter of the wheels, the wheel set is gradually brought back towards the central position.Thus, this kinematical mechanism results in an oscillating lateral motion (**Fig.** 1.24).

Hence, wheel conicity makes the wheelset to come back whenever it tries to shift from the track center, thus providing a built in guidance system. Only when the lateral shift is excessive, will the flanges come into play and prevent the wheelset from derailing. On the other hand, with cylindrical wheels, even with minor disturbing forces, such as cross level difference, wheel flange would hug the lower rail and continue to do so with either the left rail or the right one, whichever happened to be lower. This would wear out the rail and wheel flange much sooner.

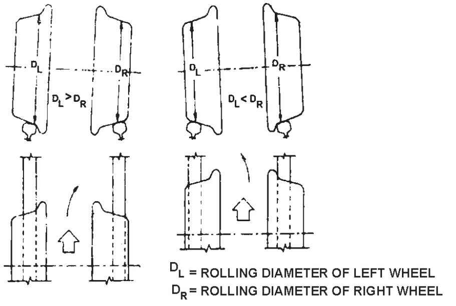


Figure 1.24 wheelset guidance by tread conicity

Thus, with conical wheel, the resulting motion of wheel set is lateral oscillatory. Mathematically, this motion is represented by a sine wave and, hence, it is called as lateral sinusoidal oscillation (**Fig.**1.25).

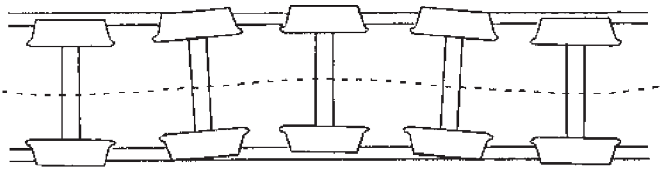


Figure 1.25 Sinusoidal motion of coned wheelset.

This is called the 'self excited oscillation' of the wheelset.

At this stage, we may define the parameters of a sinusoidal motion. Center of axle follows the path shown by wavy line in **Fig.**1.26.

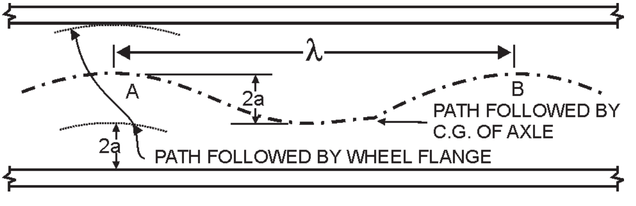


Figure 1.26 Sinusoidal motion of center of coned wheelset

This motion is a type of periodic motion. A periodic motion is one which repeats itself in all its particulars after a certain interval of time, called the period of oscillation, usually denoted by T. The distance pertaining to this time is called the wave length, denoted by Greek alphabet λ. In **Fig.**1.26, AB is the wave length λ.

If v is the speed of the vehicle, obviously,

$$\lambda = vT \qquad (1\text{-}7a)$$

$$\text{Or,}\ T = \lambda/v \qquad (1\text{-}7b)$$

1.8.1 AMPLITUDE OF OSCILLATION (a)

The maximum lateral displacement of an oscillating body from its position of equilibrium is called the amplitude of oscillation.

In **Fig.** 1.26, 'a' is the amplitude of lateral oscillation, 2a being the total range of lateral movement.

1.8.2 FREQUENCY OF OSCILLATION (f)

The number of times a periodic motion repeats itself in all its particulars in one unit of time, (usually one second) is known as the frequency of oscillation, designated as f, it is expressed as cycles per second (cps in short) or Hertz (Hz in short).

As T is the time of one cycle, number of cycles per second

$$f = \frac{1}{T} \qquad (1\text{-}8a)$$

$$\therefore\ f = \frac{1}{T} = \frac{v}{\lambda} \qquad (1\text{-}8b)$$

1.8.3 REPRESENTATION OF A WAVE MOTION BY A POINT REVOLVING IN CIRCLE

For analytical purposes, it is more convenient to represent a sinusoidal motion by revolving motion of a point around a circle. (In vibration analysis, a sinusoidal motion is more aptly referred to as a harmonic motion). The similarity between the two can be understood from **Fig.** 1.27.

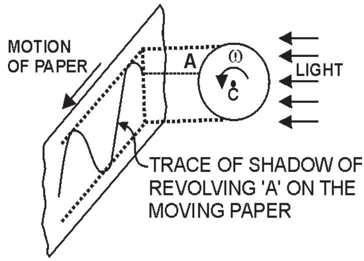


Fig.1.27 Representation of sinusoidal motion

Say, a point A is revolving in a circle around center C. Let a continuous strip of paper move in a direction perpendicular to the plane of the circle. If a light source is held on one side and a trace of the shadow of the point A on the moving paper is drawn on the paper, such a trace, as can easily be visualized, would be sinusoidal in shape, its amplitude being equal to the radius of the circle.

In this representation, the linear speed v of the wave is releted to angular frequancy ω (radians per second) of the point A.

$$v = a\,\omega \qquad (1\text{-}9)$$

Period T i.e. the time taken for a vehicle to travel over a distance of one wavelength would thus be the time which the point A takes to complete one revolution.

$$\therefore \text{Angular frequency } \omega = \frac{2\pi}{T}$$

As $$\frac{1}{T} = f = \frac{v}{\lambda} \qquad (1\text{-}10a)$$

As $$\omega = \frac{2\pi}{T} = 2\pi\frac{v}{\lambda} \qquad (1\text{-}10b)$$

1.8.4 LATERAL DISPLACEMENT

Lateral displacement from the equilibrium position at any instant of time t (measured in reference to the beginning of a cycle of oscillation) during an oscillation can be evaluated from the representative circular revolution (**Fig.**1.28).

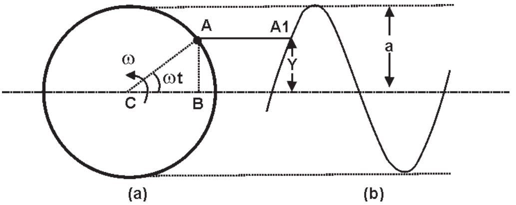


Fig. 1.28 (a) Representative circular motion,
(b) Actual sinusoidal (harmonic) motion

Say, the wheel tread contact is at position A during the oscillatory movement, its lateral displacement at a given time t is y.

Now referring to **Fig.**1.28, the corresponding position of the revolving point is A.

Say CA = a (a is the amplitude which is equal to radius of the circle)

Angle ACB = ω t (viz, angular movement in time t = angular speed x time)

It is clear that lateral displacement at any time t,

y = AB

= AC sin ωt

= a sin ωt (1-11)

We shall be referring to this expression later in the discussion.

1.8.5 LATERAL OSCILLATION AND WHEEL TREAD CONICITY

The wavelength λ_0 of the sinusoidal motion which a single wheelset would execute, is given by the formula (Klingel):

$$\lambda_o = 2\pi\sqrt{\frac{rG}{2\gamma}} \qquad (1\text{-}12)$$

Where r = wheel radius measured at a distance of 63.5 mm from inner face of wheel in case of B.G. wheel, and at a distance of 57mm in case of MG wheel (**Fig.**1.29)

G = Dynamic gauge

Dynamic gauge is the distance between the two wheel tread-railtop contact points of a wheelset. It is approximately taken as the distance c/c of rail heads. The values of G adopted on Indian Railways are :

Broad Gauge :	G	= 1750 mm
Metre Gauge :	G	= 1058 mm (or 1060 mm)

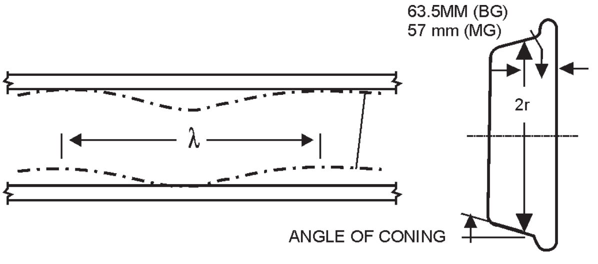


Fig.1.29 : Kinematic oscillation of a wheelset

γ = tangent of the angle of coning (e.g. for 1 in 20 coning of a new wheel profile,

$$\gamma = \frac{1}{20} = 0.05$$

As the wheel wears out, the effective conicity always increases; it may become, as steep as 1 in 10 ($\gamma = 0.1$) or even steeper i.e. the value of λ increases. For a given guage system, r and G are more or less constants. Thus λ_0 is mainly dependent on γ .

It will be evident from the expression (1-12) that with increase in value of γ , λ_0, reduces. The frequency of oscillation f , being equal to v/ λ , increases with a reduction in value of λ . This increases the wheel instability. **Fig.**1.30 illustrates the above point.

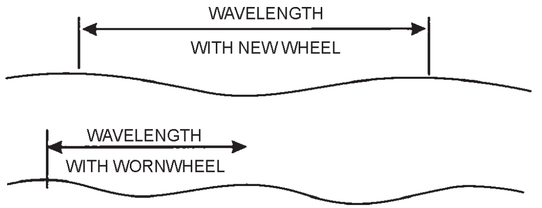


Fig. 1.30 Sinusoidal motion (a) with new wheel, (b) with worn wheel (i.e. with increased conicity, γ)

It will be clear from **Fig.**1.30 that the sinusoidal motion with worn wheel has higher frequency of oscillation as compared to sinusoidal motion with new wheel. In other words, an excessively worn wheel causes the wavelength of its sinusoidal motion to reduce, and frequency of oscillation to increase, thus, causing an adverse effect on vehicle stability. It follows, with other parameters remaining constant, the speed at which vehicle instability occurs has relation to effective conicity, γ . Thus for higher speeds, it becomes necessary to reduce conicity.

On most of the world railways, 1 in 40 conicity is provided for new wheel profile for speeds of 160 km/h and higher. Further, effective conicity increases with wheel wear, adversely affecting lateral acceleration (discussed in subsequent paras). On this account, effective conicity is monitored and maintained within specified operating limits on high speed rails.

Straight coning of a new wheel changes to curved one when worn. Question arises where and how to measure the conicity of a worn profile? In such a case, what we have to measure is the effective conicity γ_e

$$\gamma_e = \frac{\Delta r}{\Delta y} \qquad (1 - 13)$$

Δ r change in the rolling radius with lateral displacement Δ y of the wheel (**Fig.** 1.31)

For ascertaining the effective conicity, the profiles of the two wheels of the concerned wheelset should be recorded accurately with the help of a lead strip or preferably by a profilemeter. These profiles should then be aligned over the head profiles of standard rail (of section in use on the particular stretch) drawn to appropriate inclination viz. 1/20 (**Fig.**1.32)

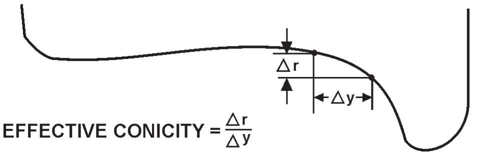


Fig. 1.31 Effective conicity

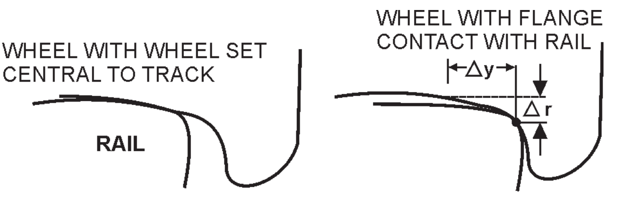


Fig. 1.32 Measurement of effective concity

The respective positions of the wheel and rail should be such as would be obtained when wheelset is central to the track gauge. Carefully identify the point of tread contact in this position, at the wheel for which effective conicity is to be measured.

Shift the wheel profile till the flange touches the rail gauge face. Note the tread contact on the wheel in this position. The distance between the two tread contacts is Δy. Measure the difference in wheel radii at these two points. This is Δr. Then, effective conicity,

$$\gamma_e = \Delta r / \Delta y$$

1.8.6 EFFECT OF RIGID WHEEL BASE ON SELF-EXCITED OSCILLATIONS

A wheelset rarely moves independently; generally it moves as part of a rigid wheel base. Wheelsets forming a rigid wheel base are held parallel to one another in vehicle or bogie frame and cannot become angular relative to one another while negotiating a curve or turnout. A rigid wheel base may have 2,3, 4 or even more axles. An example of a common bogie (CASNUB) is illustrated in **Fig.**1.33.

When held in a rigid wheel base, a wheelset has less freedom to oscillate sideways, as the other wheelset (s) impose a restraint thought the vehicle or bogie frame.

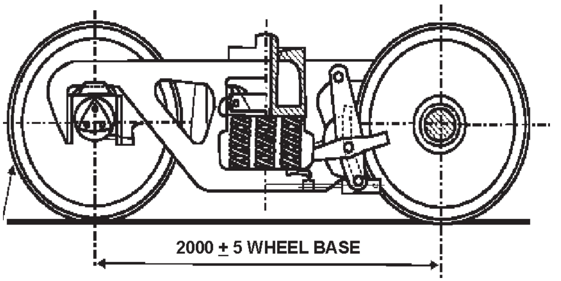


Fig.1.33 : Rigid wheel base, l, in case of a bogie stock

The wavelength of sinusoidal motion of wheelset held in a rigid wheel base of length l is given by the expression:

$$\lambda = \lambda_0 \sqrt{1 + \left(\frac{l}{G}\right)^2} \qquad (1\text{-}14)$$

λ_0 = Wavelength of wheelset when running independently (refer expression 1-12)

G = Dynamic Gauge

NB : Due to clearances at the axle box level, actual value of λ in practice is less than what is given by expression (1-14).

We have seen that longer wavelength implies greater stability. Expression (1-14) conveys that λ will be more if l is more. Hence, a longer wheel base is better from the point of view of running stability on straight.

The wheelbase, however, is limited from other considerations e.g. negotiation around curves and turnouts, wheel offloading on transition portion of curves, maximum axle load etc.

1.8.7 LATERAL ACCELERATION AND EFFECT OF PLAY BETWEEN WHEELSET AND TRACK

A wheelset can oscillate laterally only to the extent of play available between wheelset and the track. In other words, the maximum amplitude of oscillation can be only equal to half the amount of play, i.e. maximum amplitude

$$a = \frac{\sigma}{2}$$

Where σ = play

Now let us refer back to **Fig.** 1.28. We have seen that the lateral displacement y, at any instant of time t, is given by the expression :

$$y = a \sin \omega t,$$

If we evaluate the rate of change of lateral displacement i.e. we get the lateral velocity at time t i.e.

$$\frac{dy}{dt} = a \omega \cos \omega t$$

Further, rate of change of velocity would give the value of lateral acceleration i.e.

Acceleration = $-a\,\omega^2 \sin \omega t$

It will be clear that maximum value of lateral acceleration would occur when $\sin \omega t = 1$, i.e. when $\omega t = 90^0$, in which case peak value of lateral acceleration = $a.\,\omega^2$

We know that $\omega = 2\pi f$

$$= 2\pi \frac{v}{\lambda}$$

Peak value of lateral acceleration = $-a\dfrac{4\pi^2 v^2}{\lambda^2}$ (1-15)

Thus, for a given speed, lateral acceleration varies inversely as square of wavelength of oscillation e.g. if the wavelength reduces to half, the lateral acceleration would increase 4 times.

We know that for a given gauge system and wheel radius

$$\lambda \propto \frac{1}{\sqrt{\gamma}}$$ (refer expression 1-12)

or $\lambda^2 \propto \dfrac{1}{\gamma}$

Lateral acceleration $\propto \dfrac{1}{\lambda^2} \propto \gamma$ (γ is the tangent of angle of coning).

This further clarifies the adverse effect of increase in effective conicity viz. increased frequency of oscillation, increased lateral accelerations and, hence, increased lateral force Y, adversely affecting stability and safety.

The other parameter is 'a' viz. amplitude. Lateral acceleration is directly proportional to 'a' i.e. directly proportional to $\frac{\sigma}{2}$ and hence to σ i.e. play. Greater the play between the wheelset and track, greater is the possibility of the increased amplitude of oscillation , implying increased lateral accelerations and hence increased lateral forces.

An increased lateral oscillation also results in increased vertical oscillation through the effect of coning. As the wheel shifts to one side,the rolling diameters change, resulting in one side of the vehicle rising (on the side where the rolling diameter is greater) and the other side falling. As the wheel oscillates to the other side, the vertical movements also reverse. This gives rise to vertical oscillation, specifically, rolling oscillation. Thus, both Y and Q (to some extent) are adversely affected due to lateral oscillation. However, decision regarding optimum value of clearance is not simple. A certain minimum play is necessary. Otherwise, due to constricted guideway, wheels would impose undue strain on rail to sleeper fastenings and may cause overturning of rail. Also, smaller clearance would result in rail-wheel contact being nearer to flange. In case of severe lateral oscillation (may be caused by disturbances such as alignment/ cross level defect), the contact may even reach the flange root. Effective conicity for such contact locations would be very large, resulting in large lateral acceleration. In extreme cases, even flange may come in contact with gauge face of rail, resulting in 'hunting', which is explained in the following para. In addition to instability of vehicle, these conditions are likely to increase maintenance problems of gauge corner fatigue defect of rail, lateral rail wear and root wear of wheel.

Standard play on the Indian Railways is 19 mm for BG and MG. This play can increase considerably during service due to wheel flange wear and slack gauge.

For instance, a maximum wear of 12.5 mm is permissible in case of BG wheel (9.5 mm in case of MG wheel) in the thickness of wheel flange. On B.G. with this flange wear, the play would increase by an amount(= 2 x 12.5 mm) 25 mm. The total play would become (= 19 + 25 mm) 44 mm.The amplitude of oscillation can build up to a value of ½ the above amount viz. 22 mm, as compared to 19/2 = 9.5 mm, when the track gauge is exact and wheels are new. Lateral accelerations and, hence, lateral force, Y may increase in direct proportion to these values. With slack gauge, effect would be further enhanced.

Standard play on Standard Gauge on many of the Continental Railways was 10 mm, which was reduced to 7 mm by adopting 3 mm tight gauge. In view of this, standard play on the Indian Railways was reduced to 16 mm on the B.G. by adopting 3 mm tight gauge viz 1673 mm gauge instead of 1676, on straights and curves upto 4^0(radius = 440 m) on Group A, B, C & D routes. (This was in pursuance of the recommendations of Committee of Directors, Chief Engineers and ACRSs in their Report on Review of Track Standards, B.G.)

Simulation studies carried out by RDSO with the help of NUCARS (New and Untried Cars Analysis Regime Simulation) software have shown that effective conicity increases with reduction in standard play between wheel and rail. Also, lateral acceleration and ride index have been found to increase with reduction in standard clearance. UIC have carried out survey of wheel and rail interface design and maintenance practices, as a part of the Joint Research Project (JRP-2) on wheel and rail interface optimization, in the year 2005. As per the survey data, standard clearance prevailing on world Railways, as compared to Indian Railways, are lower for only passenger Railways, higher for only freight Railways and comparable for mixed traffic Railways.

During operation, following factors increase play between wheel and rail, increasing lateral oscillations and lateral acceleration.

- slack gauge,
- thin flange,
- excessive play between bearing and journal,

lateral play between axle box and pedestal/ bogie frame , and lateral play between bogie frame and bolster

Apart from resulting in increased oscillations, the above factors contribute towards increasing the derailment proneness through increased angularity of the axle.

1.9 CRITICAL SPEED

As the speed of vehicle picks up, it may encounter two types of hunting oscillations :

- primary hunting
- secondary hunting

(N.B. : A violent oscillation behaviour of a vehicle normally in a combined rolling and nosing mode is called hunting).

1.9.1 Primary hunting

This type of hunting normally occurs at relatively low speeds. In this the body oscillations are high but the bogie is relatively stable. Primary hunting mainly affects riding comfort and can be controlled by suitable damping.

1.9.2 Secondary hunting

In this, the vehicle body is relatively stable but the bogie oscillations are very high. This occurs at higher speeds. Generally speaking, amplitude of lateral oscillation increases with speed until it is equal to half of the flange way clearance. Beyond this limit, flanging occurs and , as a result, the axle rebounds. Lateral movement does not remain harmonic (sinusoidal) anymore, and becomes zig-zag (**Fig.** 1.34). Wavelength becomes shorter and frequency increases rapidly until it is in the critical range for the rolling stock and resonance occurs.

The vehicle speed at the boundary of the stable and unstable bogie running conditions i.e. the speed at which an initial oscillation just maintains its amplitude, is known as the **critical speed** of the vehicle.

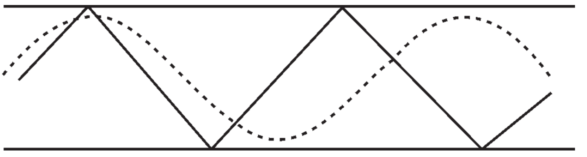


Fig. 1.34 Effect of flanging

Critical speed depends upon a variety of factors. The single most important factor is effective conicity of the wheel, the critical speed being inversely proportional to the effective conicity, other parameters remaining constant.

Some of the other factors are :

- Clearances at the axle box level
- Rotational stiffness of the bogie
- Total mass and its distribution
- stiffness values of the suspension system in various modes of oscillations, etc.

An awareness of phenomenon of critical speed is important during investigation of derailments.

The speed which a rolling stock is cleared for, is normally kept about 10 to 15% below test speeds. Since speed test is normally conducted to ascertain the maximum speed upto which the particular rolling stock would be stable, test speed is indicative, in general of critical speed of the vehicle, which may be taken to be slightly higher than the test speed.

1.10 CYCLIC TRACK IRREGULARITIES AND RESONANCE

So far we have discussed the oscillating motion of a wheelset arising out of effect of conicity, the track having been considered straight with more or less perfect geometry, i.e., oscillation consequent to an initial excitation. Now let us consider the interaction between the track irregularities and vehicle suspension system. The following example would illustrate the problem.

Consider a BG fish plated track with standard 13 m rails having a series of low joints. When a vehicle travels over such track, the vehicle will be subjected to an excitation every time it passes over a low joint, viz at the rate of 1 excitation every 13 m. The resulting oscillation of the vehicle would be in bouncing mode or pitching mode.

For a vehicle speed of 13 m/sec,frequency of excitation would be 1 cycle per second (cps) or 1 Hz (Hertz). This frequency of excitation is called forcing frequency, as it is an oscillation frequency which is forced on the vehicle just by virtue of its travel over a cyclic irregularity in the track.

If λ is the wavelength of particular track irregularity (in the above example, λ_t = 13 m) and v the speed, then, Forcing frequency

$$f = \frac{v}{\lambda_t} \qquad (1\text{-}16)$$

A vehicle suspension comprises a system of springs. Like any other body, it has various natural frequencies of oscillation in each of the six oscillation modes. Natural frequency is an inherent property of a body or a suspension system. When a suspension system is given an initial excitation in a particular mode by an external force, and that force is removed, the frequency with which the system oscillates freely (without any further external excitation) is called its natural frequency in that mode.

Natural frequency depends on stiffness of the spring system and the masses. For instance, for a simple spring having stiffness k (load per unit deflection) and mass m attached to it, natural frequency

$$f = \frac{1}{2\pi}\sqrt{\frac{k}{m}} \qquad (1\text{-}17)$$

(**N.B.** For a two stage suspension system there would be two natural frequencies.)

If forcing frequency in a particular mode of oscillation equals natural frequency in that mode, a phenomenon called **resonance** occurs. If there is no damping whatsoever in the system, the amplitude of oscillation at resonance tends to build with time and become infinite. Build up of oscillation amplitude at resonance can be controlled only by suitable damping.

Depending on the extent of damping, a system may be :

- Underdamped
- Critically damped
- Overdamped

Fig 1.35 shows all the four type of free oscillations for the sake of comparison. It will be seen that after initial excitation to an under- damped system, a few cycles of oscillation take place before the system comes to rest at the equilibrium position.

In an over-damped system, the system after being initially excited take a long time i.e. time more than one time period, to come back to the equilibrium position and no cycle of oscillation takes place.

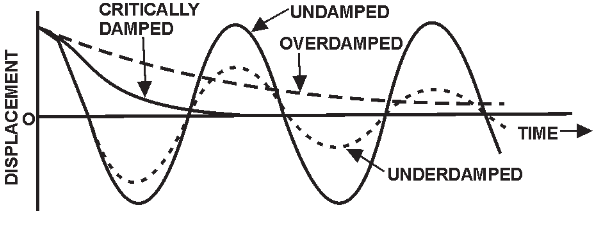


Fig. 1.35 Type of Free Oscillations

An intermediate extent of damping, called the critical damping, is the minimum damping that can be used to just prevent the system from overshooting the equilibrium position, when displaced from it and released. The ratio of extent of damping (C) actually present in a system to the value of critical damping (C_c) for the particular spring-mass system, is called the damping factor (denoted by Greek letter, zeta ζ).

$$\xi = \frac{C}{C_c} \tag{1-18}$$

Now, say the amplitude of track irregularity is X_0 and, as a result, the vehicle passing over it oscillates with an amplitude X,

then, the ratio $\dfrac{X}{X_0}$ is called the magnification factor. (1-19)

If we plot the magnification factor obtained under different values of damping factors for different ratios of forcing frequency to natural frequency, for a simple spring-mass-damper system shown in **Fig.** 1.36, we would get the curves, as shown in the Figure.

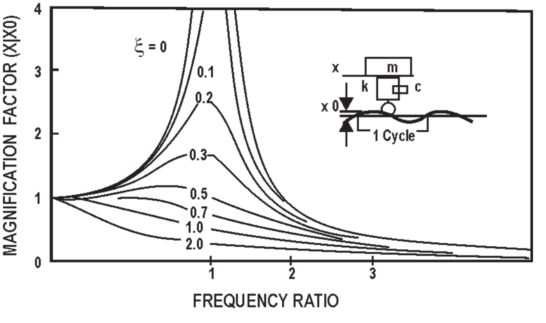


Figure 1.36 Magnification factor for a simple spring-mass damper system

It will be seen from above that at resonance (i.e.when f/f_n = 1) if ξ = 0, magnification factor is infinite i.e. at resonance, in absence of damping even a small irregularity would generate an infinite value of amplitude of oscillation in a vehicle.

But if ξ is adequate, say 0.7 or so, the magnification factor does not exceed one i.e. vehicle will not oscillate with an amplitude greater than the amplitude of the irregularity, even at resonance.

The foregoing clearly brings out the crucial importance of damping in a vehicle suspension system.

It is impossible to have a track without cyclic irregularities. Considering the varying speeds with which vehicles travel over such irregularities, the possibility of forcing frequency equaling the natural frequency and, hence, of resonance is a very real one and, in fact, does occur at some occasion or the other. The only way to safeguard against the adverse effects of resonance is to ensure adequate damping at all times.

For reference, the following irregularities would, generally, generate the oscillations indicated against each.

Track irregularity	Mode of oscillation	Affects
Low joints, unevenness, loose packing etc.	Bouncing, pitching	Q
Alignment or gauge faults	Lurching Or nosing Or rolling	Y Y Q
Twist	Rolling	Q

The above track irregularities when occurring in a cyclic form would cause resonance in the relevant oscillation modes indicated (at a particular vehicle speed).If damping is inadequate, it would have adverse effect on Y or Q, as the case may be, through build up of oscillation amplitude.

Thus, in derailment investigation, it is important to check the condition of damping system in the suspension of vehicles involved in reference to what is required to be provided and maintained as per design.

1.11 EFFECT OF TRACK AND VEHICLE TWIST ON WHEEL OFF-LOADING

Another parameter having crucial impact on safety aspects of vehicle track interaction is the track and vehicle twist.

1.11.1 Track twist

In general, twist between any four points is defined as the normal distance by which any one of the points lies outside the plane formed by the other three points. If all the four points lie in a plane, twist is zero.

When referring to track twist, it becomes convenient to define it as rate of change of cross level.

Say, twist is to be measured over 2 locations A and B, which are L metres apart (**Fig.**1.37)

Then, track twist (mm/m) =
(Algebraic difference of cross level at A and B) / L (1-20)

To illustrate the above, say

Cross level at location A = +5 mm (i.e. left rail is higher than right rail by 5 mm),

And at location B = - 7 mm (i.e. left rail is lower than

right rail by 7 mm)

A and B being 3 m apart

Then, track twist = $\frac{+5-(-7)}{3} = \frac{12}{3} = 4mm/m$ or more appropriately, 12 mm over a base of 3 m.

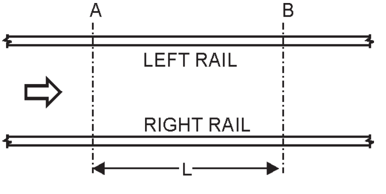


Fig. 1.37 Calculation of track twist

Pondering over above, it will be clear that twist is independent of gradient. Does the circular portion of a curve on a gradient of say 1 in 70 with, say, 75 mm cant have twist (assuming no irregularity), as compared to the circular portion of a curve with same cant on a level track ?

1.11.2 Effect of track twist

As an illustration, let us consider a stationary vehicle. If a springless vehicle stands on a track with twist, the situation is the same as that of a wooden table standing on an uneven ground such that it keeps on rocking about a diagonal (**Fig.** 1.38). In this situation,the legs which are in rocking position would be having zero load (completely off-loaded). We have seen above (Chartet's Formula) that wheel load becoming less than 35 % of nominal load would result in an unstable run. A springless vehicle or a vehicle with springs which are infinitely stiff would behave in the same manner. Here comes the role of spring in safe run of railway vehicle.

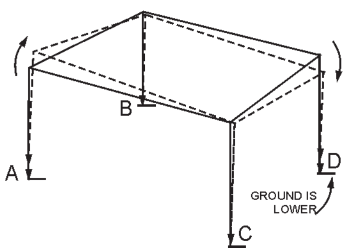


Fig. 1.38. Table on ground

1.11.2.1 Mechanism of re-distribution of forces among spring support

A loaded spring has a capacity to compress further or elongate. This feature of spring provided in suspension of railway vehicle enables it to negotiate track twist without excessive off-loading. The spring over the wheel which encounters a depression in track elongates and, thus, helps maintain some load on the wheel.

Let us consider a 4-wheeler under frame (or bogie frame) supported on four springs of equal stiffness and length, which are, in turn, supported on four points in one horizontal plane. Let us assume that the frame is loaded with a vertical load of 40 t , applied symmetric to the frame, i.e. at its centroid. Considering symmetry and force and moment balance, reactions applied by the four spring would each be 10 t. Therefore, compressive loads, and hence compression, in all the springs would also be equal (**Fig.** 1.39).

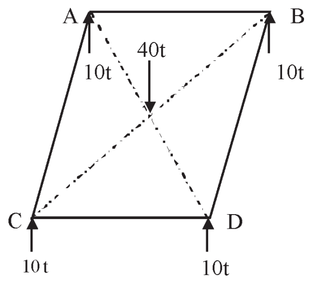


Fig. 1.39

Let us find out the effect of introduction of a depression in the support point of spring B (twist), of an amount equal to the spring compression. Here support points of other springs remain undisturbed. Assuming that the frame remains in its original position, compression (and reaction force) of spring at B would become zero (**Fig.** 1.40).

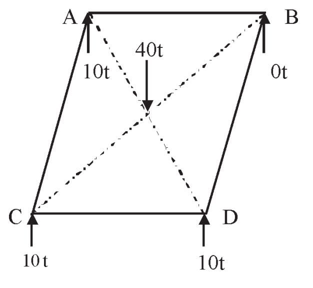


Fig.1.40

However, the frame is not in a balanced condition under application of these forces. This would result in movement (translation/ rotation) of the frame. This change can be considered to be consisting of the following two stages.

There is an unbalanced moment about diagonal AD. This would result in rocking of the frame about this diagonal. As per geometry, any tilting of the frame about this diagonal would result in increase in compression of spring B equal to decrease in compression of spring C, and vice-versa. The frame would finally tilt such that, for moment balance about AD, reactions in spring at B and C would each become 5t (**Fig.** 1.41).

Although moment equilibrium about diagonals AD and BC is achieved in this position, there is an unbalanced vertical downward load of 10t on the frame. Considering the symmetry, the entire frame would move downward in such a manner (equal incremental

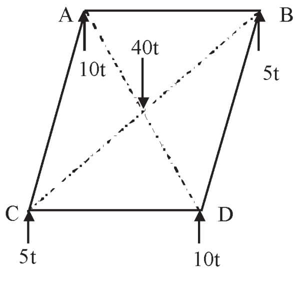


Fig. 1.41

compression of all springs) that reaction force in all the springs would increase equally, by 2.5t (**Fig.** 1.42).

From the above analysis, it is seen that if one of the spring force becomes zero due to a defect of twist, the frame both tilts and moves vertically, finally restoring (3/4 th) of the spring force. A similar analysis would reveal that this result is valid for a general case i.e. final off-loading of a spring would be only (1/4 th) of the initial off-loading. In a general analysis, all the four springs would be under equal compression (equal to $\delta_0 = f*Q_0$, where f is spring flexibility and Q_0 is the spring load corresponding to no twist). In order to find out the magnitude of depression below a spring support which would finally make the spring load zero, a series off-loading by the above mechanism would need to be considered.

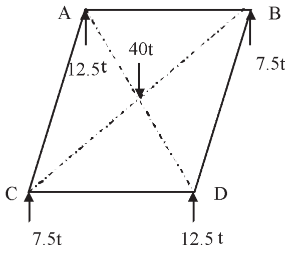


Fig. 1.42

In the Ist cycle, a depression of δ_0 is required to make the spring load zero. As 3/4th of the spring load is restored back, in the 2nd cycle a further depression equal to 3/4th of δ_0 would be required to make the spring load zero. Similarly, an infinite number of off-loading effort (depression) would be required. Hence, the total depression required to make the spring fully off-loaded would be

$$\delta = \delta_0 + (3/4)\,\delta_0 + (3/4)^2\,\delta_0 + \ldots\ldots\ )$$
$$= \delta_0\,(1+3/4+(3/4)^2 + \ldots\ldots\ldots)$$
$$= \delta_0 \quad (1/(1-3/4))$$
$$= 4\,\delta_0$$

Thus, a total depression of $4\delta_0$ at the support of spring B would make its reaction force zero. A corresponding analysis of increase in the compression of Spring A would conclude that the force of this spring would become $2Q_0$. This can also be concluded considering that the frame is loaded at its centroid, and hence the sum of reaction forces of spring A and B should remain unchanged ($2Q_0$) from equilibrium considerations.

1.11.2.2 Track defect and magnitude of wheel off-loading

Let us now consider the influence of a depression in track below one of the wheels (twist) on wheel off-loading. Consider a

4-wheeler (/ bogie) standing on a level track. Let T be the total weight of the 4-wheeler (/bogie). Here, equilibrium of the wheelset which is above the track location with a depression is considered.

Referring to **Fig.** 1.43,

a = Distance between centers of the spring A and B

G = Dynamic gauge

e = Overhang of spring center

beyond the wheel-rail contact point (taken equal on both sides)

P_A = Load in spring A

P_B = Load in spring B

R_1 = Vertical wheel load under wheel – 1

R_2 = Vertical wheel load under wheel – 2

Let the rail under wheel-2 be depressed by an amount (Z_0) which just makes the wheel load of wheel-2 drop to zero (R_2 =0).

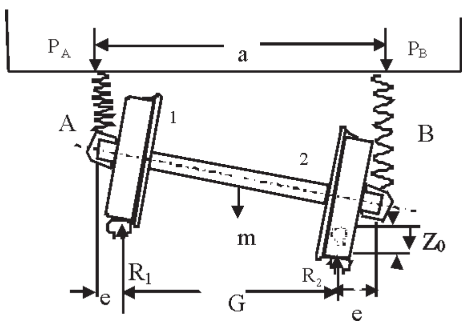


Fig. 1.43

Under the effect of this lowering of rail, the wheel set would assume an inclined position, with wheel 2 maintaining contact with rail, and spring B would elongate, resulting in drop in spring reaction P_B.

Taking moments about spring B :

$P_A a + m\,(G/2 + e)$

$= R_1\,(G+e) + R_2 e$

$= R_1\,(G+e)$ (since $R_2 = 0$) ……(1)

Taking moments about spring A :

$P_B a + m(G/2 + e)$

$= R_1 e + R2(G+e)$

$= R_1 e$ (since $R_2 = 0$) …….(2)

Subtracting (2) from (1)

$(P_A - P_B)a = R_1 G$

$(P_A - P_B) = R_1 G/a$

As the 4-wheeler (/bogie) is symmetrically loaded,

$R_1 + R_2 = T/2 \Rightarrow R_1 = T/2$

(since $R_2 = 0$)

$P_A - P_B = (T/2)*(G/a)$ …….(3)

As discussed earlier, a depression of $4\delta_0$(where $\delta_0 = fQ_0$) at the support of spring B results in a difference of $2Q_0$ in spring reaction forces of A and B ($P_A - P_B$).

Therefore, for complete off-loading of wheel 2, when $P_A - P_B = (T/2)*(G/a)$ from equation (3) above, depression of support of spring B (with respect to support of spring A) should be,

$4\,fQ_0 *(1/2Q_0)*(T/2)*(G/a)$

$= fT(G/a)$

Further, this relative depression of spring supports is related to the relative depression of the rails Z_0. From geometry of **Fig.** 1.43,

Z_0 = inclination of wheel set * G

$= (fT(G/a)/a)*G$

$Z_0 = fT(G/a)^2$ (1-21)

Z_0 is the track twist which would fully off-load the corresponding wheel. This equation is called Kerestzy Equation. However, as discussed earlier, the vehicle movement becomes unstable beyond 65% off-loading, even for very small flange forces. Hence, theoretically, 0.65 Z_0 may be considered the wheel safety depth beyond which a track depression may result in unstable run.

Referring to the problem which was outlined in the beginning, it would be clear that higher the value of Z_0 better it is, because it means that for wheel of such vehicle to offload completely, a greater track twist would be needed, a condition which is certainly more desirable. It brings out that a vehicle with better suspension system has more capability of negotiating safely even greater track irregularities in twist parameter.

In general, greater the value of Z_0 greater is the capacity of the 4-wheeler bogie to negotiate a given track twist with the residual wheel load (after offloading) remaining higher than that if Z_0 were smaller.

This, in reference to expression (1-21), implies that f, T and the ratio G/a should be as high as possible (subject, however, to other technical and practical constraints).

These three factors are now discussed below.

1.11.2.2.1 STIFFNESS OF SPRING

We have seen earlier that a springless vehicle i.e. a vehicle with no capacity to deflect under load, would have one of its wheels completely off-loaded even with a minor twist. On the other hand, springs provide the ability to deflect under load. The amount of deflection should be such that even after encountering twist, it does not elongate back fully but some residual deflection remains (deflection indicates load and residual deflection means there is still some load on the wheel). **Fig.**1.44 illustrates this.

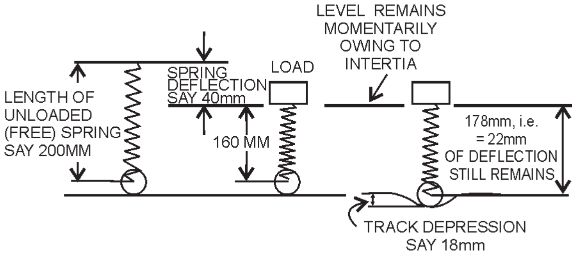


Fig. 1.44 Elongation of spring

This means, greater the amount of deflection per unit load, i.e. softer the spring, safer it is from twist point of view. Keresztys theory, too, brings us to the same conclusion i.e. f, the specific deflection of springs should be as high as possible. From practical considerations, e.g. buffer heights etc; too soft a spring cannot be provided. There is an optimum value for specific deflection of springs in a particular rolling stock.

If for any reason, the spring becomes stiffer than its design value, deflection per unit load decreases, consequently, reducing the extent of twist which such vehicle can negotiate safely. For instance, in a laminated spring (also called leaf spring), deflection is accompanied with relative longitudinal sliding between the leaves (**Fig.**1.45). If there is undue resistance to longitudinal sliding e.g. owing to rusting of leaves, there will be corresponding resistance to deflection, which would increase the derailment proneness.

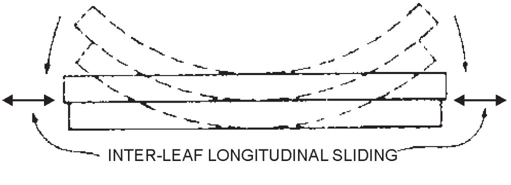


Fig.1.45 Inter-leaf sliding in laminated spring

1.11.2.2.2 Vehicle load

It has been a well known fact that an empty vehicle is more prone to derailment than a loaded one, on twisted track. Why? As seen from Keresztys theory, for Z_0 to be higher T viz. the total weight of the wagon should be higher. Greater load produces greater deflection of the springs and, hence, affords greater ability for the vehicle to negotiate a given track twist with the residual wheel load (after offloading) remaining higher than that obtaining if T were less.

1.11.2.2.3 G/a Ratio

If overhang is greater in one type of vehicle in relation to gauge, i.e. if G/a is less, as compared to another type, a given twist will produce grater elongation of spring in the former case and, hence, greater off-loadi; Thereby, increasing derailment proneness.

G/a values for common B.G. and M.G. 4-wheelers are :

B.G.	:	0.78	$(G/a)^2 = 0.6084$
M.G.	:	0.71	$(G/a)^2 = 0.5041$

The above partly explains, why M.G. 4-wheelers are more prone to derailment as compared to B.G. 4-wheelers.

Apart from the above three factors, the following factors affect the extent of twist that can safely be negotiated from consideration of wheel offloading.

1.11.3 Spring defect

Spring deflection in different springs under the same load per spring would rarely be the same. Such a variation has exactly the same effect on wheel offloading as that due to track twist.

Let us consider the variation in specific deflections in reference to.

- one spring, or.
- two diagonally opposite springs.

Going back to our example of the table, say a table with legs A,B, C and D is placed on a perfectly level ground. If the leg D is shorter than the rest, say by 10 mm, the extent of rocking of table will be through a range of 10 mm. Now, say the diagonally opposite leg A is also shorter than those at B and C by 10 mm. It will be quite clear that the extent of rocking will be through a range of (-) 10 mm to 10 mm i.e. 20 mm. In other words, the effect on wheel offloading in a vehicle having spring deflect in two diagonally opposite springs will be twice that which will occur if the spring defect is in only one spring.

Referring to Keresztys theory, if X is the defect in a spring in reference to its specific deflection, the effective defect would be X/2, as the diagonally opposite spring would share the defect through rocking and adjustment of deflections of the other 2 springs. The defect expressed as equivalent track twist would then be:

$$Z_s = \frac{X}{2}\frac{G}{a} \qquad (1\text{-}22\text{ a})$$

This will have the effect of reducing the permissible track twist by an amount Zs.

If the defect X is in two diagonally opposite springs, no further adjustment is possible and the defect expressed as equivalent track twist would be

$$Z_s = X\frac{G}{a} = 2Z_s \qquad (1\text{-}22\text{b})$$

and the permissible track twist would get reduced by this amount.

For example, a 13 mm difference is permitted in the working camber of any two of the four springs of a unit under load. Say, in two cases the working camber values are as shown in **Fig.** 1.46. Both cases are within the permissible limits. But as explained, the effect on offloading in (ii) will be twice that in (i) i.e. case (ii) is more prone to derailment than (i).

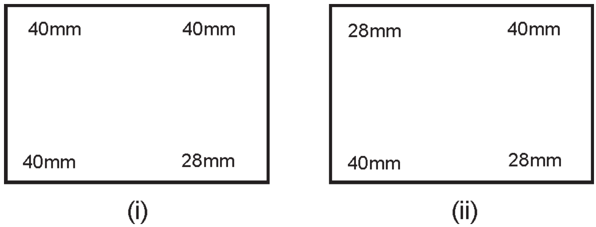


Fig.1.46 Example showing working camber values of springs of 4-wheeler

Variations in free spring height (including wheel diameter, and axle box and packing height) as well as spring stiffness are possible. Unequal loading of the left and right axle box springs, which affects wheel off-loading, can generally be detected by difference in spring heights measured in a loaded vehicle. However, under certain conditions there may be unequal loading but the same may not get reflected in loaded spring heights. Consider a case where the diagonally opposite springs have a similar defect of equal magnitude (spring height or stiffness). In this situation, there would not be any difference in loaded spring heights, although both the wheelsets would be unevenly loaded. In view of this, measurement of both loaded as well as free spring heights should be done for finding out spring defects.

In addition, for a comprehensive analysis, measurement of free spring height and load-deflection relation is required to be done in a workshop.

Considering that spring defects cause wheel off-loading in the same manner as track twist, in derailment investigation it is as important to measure these spring defects as track twist.

1.11.4 Torsional stiffness of vehicle under frame / bogie

Let us imagine a table (say 2 m x 1m) with its top only of cardboard but legs of wood. If such a table is placed on an uneven ground, will it rock? No, because, the flexible card board deflects and the legs stabilize at different level with no rocking taking place.

In other words, softer the underframe torsionally, better it is from safety consideration in reference to offloading. Deflection of an underframe is similar to deflection of a spring. If ϕ is the specific deflection of a corner of vehicle underframe, its effect expressed as equivalent track twist would be

$$Z_u = \phi \frac{T}{4}\left(\frac{G}{a}\right)^2 \qquad (1\text{-}23)$$

The permissible track twist gets increased by the above amount. Hence, greater the value of φ i.e.more flexible the vehicle underframe torsionally, (within practical limits), greater would be Zu and greater would be the permissible track twist, a feature which is desirable. On the other hand, a vehicle with a torsionally stiffer underframe has less value of φ with consequent less value of permissible track twist and, hence, is more derailment prone.

Flexibility of bogie frame in bogie vehicles has exactly similar role in reducing wheel offloading as the flexibility of vehicle underframe in a four wheeler.

In general, Zu for a B.G. 4-wheeler is about 13 mm and for M.G., 5 mm.

A rivetted underframe (or bogie frame) would be more flexible than a welded one. Open wagons will have more flexibility than covered ones.

A very good example is the large flexibility of CASNUB (BOXN) bogie on account of its construction (we will study this in subsequent chapters). It enables this bogie negotiate large track twist safely.

1.11.5 Transition portion of a curve

The transition by design has a certain twist. For instance, 1 in 720 cant gradient on the transition would mean a design twist of 1.4 mm/m.

There are two aspects that need to be studied in this regard, as follows

1.11.5.1 Track

Track can not be maintained exactly to a design value. Tolerances would always have to be permitted. Let us assume, the maximum twist we want to allow is 4 mm/m. On a straight track where design twist is zero, full 4 mm/m are available as a tolerance i.e. the irregularity permitted is 4 mm/m. Whereas on a transition, if the design twist is 1.4 mm/m, margin available for irregularity to occur is only 4-1.4 = 2.6 mm/m. If the design twist were, say, 1 in 1000 i.e. 1 mm/m, the margin available for irregularity to occur would be 4-1 =3 mm/m. But if the design twist were 1 in 360 i.e. 2.8 mm/m, tolerance for irregularity would be only 4-2.8=1.2 mm/m.

In other words,flatter the cant gradient on the transition, greater is the tolerance available in twist parameter and hence greater the safety margin. Thus, wherever space permits, cant gradient should preferably be kept as flat as possible. For the same reason, greater attention should be given to maintenance of transition portions of curves, particularly where the cant gradient is steep.

1.11.5.2 Vehicle

If a vehicle with rigid wheel-base, l_1 stands on transition curve, the vehicle suffers a twist which is equivalent to track depression say Zb1 under one wheel (**Fig.** 1.47). It will be clear that a vehicle with a longer wheel-base l_2 will encounter a value of Zb2 which will be greater than Zb1 (This situation will hold irrespective of position of the wheel-base so long as both the axles remain over the transition)

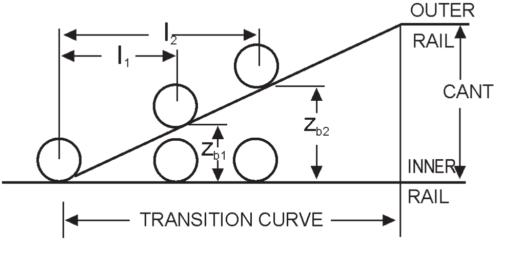


Fig.1.47 Effect of rigid wheel-base on transition curve

$Zb = i\ l$

where i = cant gradient of transition curve (e.g. for cant gradient of 1 in 720, i = 1/720)

l = rigid wheel-base

The permissible track twist has to be reduced by the above value of Zb. Hence greater the value of Zb i.e. greater the cant gradient i or longer the wheel-base l, more undesirable it is from consideration of wheel offloading. However, generally a longer wheel-base vehicle is also more flexible. Hence some of the adverse effect is offset. But if a longer wheel-base wagon is also torsionally stiff, it will be more accident-prone, particularly on transition portion of curves.

We have also seen earlier that a longer wheel-base wagon is more stable from oscillations point of view. It will thus be seen that effect of wheel base depends upon the particular situation and context i.e. from the view point of running stability, in general, longer wheel base is better, whereas over a transition curve, a vehicle with longer wheel- base, if torsionally stiff, is more prone to derailment.

This discussion refers only to a 4 wheeler vehicle. In a bogie vehicle, each bogie forms a rigid wheel-base which is small in comparison to that of 4 wheeler, and thus suffers comparatively less offloading over track twist and transitions. At the same time, the rotational friction at the center pivot and /or side bearers of the bogie, increases the effective wheel-base providing greater stability from consideration of vehicle oscillations on straight. A bogie vehicle, therefore, meets various situations with advantage.

1.11.6 Safe limit of track twist

Summing up quantitatively, the effect of various factors in Keresztys theory, the net track twist Zperm that can be permitted from safety consideration in reference to wheel off-loading, would be

$$Zperm = 0.65\ Zo - Zs + Zu - Zb \qquad (1\text{-}24a)$$

(if one spring is defective)

$$= 0.65\, Z_0 - 2Z_s + Zu - Zb \qquad (1\text{-}24\ b)$$

(if two diagonally opposite springs are defective)

Z_0 = track twist that would cause a wheel to off-load completely

Zs = spring defect expressed in terms of equivalent track twist

Zu = torsional flexibility of vehicle/ bogie expressed as additional track twist which the vehicle can negotiate before it offloads to a given extent

Zb = twist owing to cant gradient on a transition curve

(**N.B**. : If the depth (call it Zf) of the straight portion of wheel flange viz 9 mm for B.G. wheel and 8 mm for M.G. wheel is also considered, the above Zperm may be increased by the said values of Zf, **Fig.**1.48)

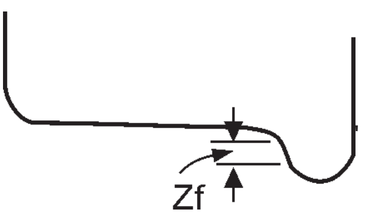


Fig. 1.48 Safety depth of flange

Values of Zperm for certain common types of B.G. and M.G. 4- wheelers have been calculated and tabulated in Efficiency Bureau, Railway Boards Study No. 4/64 on Mid-Section derailments on Central Railway (MG), and Study No.1/1966 Mid-section derailments on Southern Railway (MG). For ready reference, the values of Zperm from the second report are reproduced in **Table** 1.1

Table 1.1

Net value of Z_{perm}

S No.	Cant grade	Z perm (mm) for empty wagons					
		Meter Gauge			Broad Gauge		
		CA	C	KC	C(SWB)	C	KC
1	1 in 720 *	12.4	13.8	13.3	13.6	37.0	38.2
2	**	7.8	9.2	8.7	8.5	31.8	33.0
3	1 in 360 *	7.3	9.6	8.2	9.5	30.6	33.3
4	**	2.7	5.0	3.6	4.6	25.4	28.1
5	1 in 240 *	2.2	5.4	3.1	5.6	24.2	28.4
6	**	(-)2.4	0.8	(-)1.5	0.7	19.0	23.2

* : 13 mm defect in one spring

** : 13 mm defect in each of two diagonally opposite springs

From the above it will be seen that safety margin available from M.G. empty 4-wheelers is considerably lower than that for B.G.. However, while investigating into derailments, Keresztys theory, as in the case of Nadals formula, should be applied in qualitative terms only. Quantitative analysis can be applied only to serve as an illustration in support of relevant discussion.

1.12 LATERAL STABILITY OF TRACK

So far we have been discussing derailments involving phenomenon of flange climbing, assuming the track to be stable. There can be a situation where, under the action of relatively higher lateral force Hy, the track moves laterally before the flange climbs the rail. This is called track distortion, which might become the cause of a derailment.

It will be obvious that a track would distort if, any or both of the following occur :

- lateral strength of track reduces,
- lateral forces exerted by vehicle are excessive.

In either case, track distortion occurs when lateral forces exceed the lateral strength of track. Study of this phenomenon is important, both :

- for affording one of the criteria for assessment of stability of rolling stock, and
- for investigation of derailments.

Studies in this field have been done mainly on the SNCF (French National Railways). The aim was to determine the lateral strength of a track that could be taken as representative of the whole railway system of particular gauge, and to accordingly lay down a limit for the lateral forces that a vehicle could exert during its running. It would, thus, serve as a design criterion for assessment of stability of rolling stock.

The first question was, what should be considered as a track distortion?

Under the action of a lateral force exerted by an oscillating vehicle, track may deform slightly. Normally, such deformation is within the elastic range, i.e. as soon as the vehicle passes, the track more or less comes back to the original position. When the lateral forces reach a certain critical value, the track deforms in a manner that it does not return to the original position even after passage of the vehicle. That is, the track undergoes a permanent set.

A small permanent deformation may gradually aggravate under the repeated lateral action of passing wheelsets. When such deformation reaches 1 to 2 cm, it may be called track distortion (This track distortion is different from buckling of track which takes place under the action of thermal forces, accompanied or not by vehicle lateral forces). A lateral force, which after repeated application produces such a distortion, is said to have reached the critical limit from track stability consideration and such a force is indicative of the lateral strength of the particular track. Thus, limiting lateral force and lateral strength of track have one and the same value.

Blondel, after a number of practical tests, had proposed the following expression for the limiting lateral force or lateral strength of track

$$Hy < a + b\,P \qquad (1\text{-}25\ a)$$

Where P = axle load in tonnes.

a & b are constant

In general, a = 2 tonnes

b = 0.4

The formula then becomes

$$Hy < 0.4\,P + 2 \qquad (1\text{-}25b)$$

(**N.B** Hy has to be measured over a distance of application of 2 metres. Such a force application i.e. the one in which force acting laterally moves longitudinally, was found to be the most critical as compared to static or impact loads. Besides, it truly represents actual vehicle movement.)

The above formula, though established for Standard Gauge, is in adoption on the Indian Railways as one of the criteria for assessment of stability of rolling stock.

Later, on the SNCF, more elaborate laboratory and field tests were conducted on ordinary and LWR track, on straights and curves, also studying the effect of repeated application of each level of lateral force. As a result thereof, A. Prud' homme proposed the following expression for the limiting lateral force:

$$H_y \not> \left(1 + \frac{P}{3}\right)\rho \qquad (1.26)$$

where, ρ = a coefficient to take into account particular state of the track, with regard to its layout, structure, standard of maintenance etc. It is generally taken as 0.85.

Prud'homme formula has since been adopted by most of the World Railways, including the Indian Railways.

Since Prud'hommes formula gives a lower value than Blondels, it implies a lower limiting value for the lateral force which a vehicle may exert and, hence, is more demanding with regard to rolling stock design.

It may be mentioned that the strength of track assumed for design of vehicles is the one which an average good track will have, just after through packing.

In derailment investigation, one should identify :

- the defects in rolling stock that could have caused lateral forces Hy to increase,
- the defects in track that could have caused a reduction in its lateral strength.

  (N.B. The rolling stock and track defects are discussed in Chapters 3 to 8.)

1.13 DETERMINATION OF SAFE PERMISSIBLE MAXIMUM SPEED OF ROLLING STOCK

The Railway Board is the final authority for regulating and sanctioning speeds of all rolling stock under the Indian Railways Act. The responsibility for determining and certifying the maximum permissible speeds of the rolling stock has, however, been delegated by the Railway Board. Accordingly, RDSO in coordination with Zonal Railways, deals with all cases of introduction of new design of rolling stock, increase in speed of operation of existing rolling stock and increase in maximum permissible speed of rolling stock on a particular route.

Sanctioning speeds for new designs of rolling stock is done by RDSO by initially determining provisional maximum permissible speed based on the design features and data, and where appropriate, also on a comparison of the performance of similar designs of rolling stock already in service. The provisional speed will normally be lower than the designed or projected service speed of the stock. This provisional speed, however will not be more than 80 km/h for BG passenger stock and 65 km/h for BG goods stock. However, in respect of such stock whose suspension characteristics are superior to and whose axle loads and track loading densities are not exceeding those of the stock already proved in services, the maximum limits will be 105 km/h for BG passenger stock and 75 km/h for BG goods stock. The final maximum permissible speeds of all new designs of rolling stock will be determined by Executive Director/Stds. (Motive power) in consultation with Executive Director/Stds.(Civil) (Track) & Executive Director (Bridges & Structure), after due consideration of the services to be performed in comparison with similar stock already in service, and on the basis of oscillation trials for assessing the riding quality and/or stability. In the evaluation of the data recorded by instruments in Oscillation trails, the recommendations of the Standing Criteria Committee will be the guiding factor. These recommendations are as follows

1.13.1 Diesel & Electric locomotives

a. A lateral force Hy exerted by the wheel-axle set on track lasting more than 2 metres should not exceed 0.85 (1+P/3), where P is the axle load in ton.

b. Isolated peak values exceeding the above limit are permissible, provided the record shows a stabilizing characteristic of the locomotive subsequent to the disturbances.

c. A simultaneous assessment is carried out for the lateral force exerted by the adjacent axles at a point where a high lateral force is exerted by a particular axle to ensure that the phenomenon is only of isolated nature.

d. A derailment co-efficient should be worked out in the form of ratio between the flange force Y and the instantaneous wheel load Q continuously over a period of 1/20th second;

the instantaneous value Y/Q shall not exceed 1. (Since there is instrumentation problem in measuring Y, RDSO is presently taking Hy as Y).

e. The values of acceleration recorded in the cab at a location as near as possible to the bogie pivot shall be limited to 0.3g both in vertical and lateral directions. The peak value upto 0.35g may be permitted, if the records do not indicate a resonant tendency in the region of peak value.

f. There should be general indication of stable running characteristics of the locomotive as evidenced by the rotational movement of the bogie on straight and curved track, by acceleration readings and instantaneous wheel load variation.

g. In the case of such locos where measurement of forces as envisaged under 'a' and 'd' above is not possible, evaluation shall be in terms of Ride Index which shall not be greater than 4; a value of 3.75 is preferred.

1.13.2 Carriage

a. Ride Index shall not be greater than 3.5; a value of 3.25 is preferred. For EMU and DMU type stock, Ride Index shall not be greater than 4.0.

b. Values of acceleration recorded, as near as possible to the bogie pivot, shall be limited to 0.3g both in vertical and lateral directions. A peak value upto 0.35g may be permitted if the records do not indicate a resonant tendency in the region of the peak value.

c. There should be general indication of stable running characteristics of the carriage as evidenced by the rotational movements of the bogie on straight and curved track, by the acceleration readings and instantaneous wheel load variations/spring deflections.

1.13.3 Wagons

a. A lateral force Hy exerted by the wheel-axle set on the Track lasting more than 2 metres should not exceed 0.85

(1+P/3) where P is the axle load in ton.

b. Isolated peak values exceeding the above limit are permissible, provided the record shows a stabilizing characteristic of the wagon subsequent to the disturbances.

c. A simultaneous assessment is carried out for the lateral force exerted by the adjacent axles at a point where a high lateral force is exerted by a particular axle to ensure that the phenomenon is only of isolated nature.

d. A derailment co-efficient should be worked out in the form of ratio between the flange force Y and the instantaneous wheel load Q continuously over a period of 1/20th second; the instantaneous value Y/Q shall not exceed 1. (since there is instrumentation problem in measuring Y, RDSO is presently taking Hy as Y).

e. There should be a general indication of stable running characteristics of the wagons, as evidenced by the rotational movements of the bogie on straight and curved track, by the acceleration readings and instantaneous wheel load variations/spring deflections.

f. In the case of such wagons where assessment of forces is not possible, evaluation shall be in terms of Ride Index which shall not be greater than 4.5; a limit of 4.25 is preferred.

1.13.4 Locomotive, coaching stock and wagons - some General principles of comparative performance

For evaluation of stability for the various categories of stock, a valuable inference may be drawn by comparing the recommended criteria and other parameters of the new type of vehicle with a comparator vehicle upto a speed at which the comparator is already in service with satisfactory record of stability or with the same vehicle operated at a lower speed, where its safety has been established, the track conditions for the two comparative tests being similar. The vehicle stability can be considered acceptable upto a speed at which the performance of the vehicle is adjudged as comparable.

1.13.4.1 Oscillation test, whether for proving rolling stock or route, is undertaken on normal track which is maintained to average conditions; meaning no special maintenance work on track will be undertaken for the purpose of undertaking trials. On such average conditions of the track, the above mentioned criteria regarding ride behaviour of the rolling stock should be met with through the design characteristics of the rolling stock. The rolling stock is also expected to be in the normal maintenance conditions obtainable in service and not as of new conditions, with the wheel tyres of average wear.

1.13.4.2 From the foregoing paragraphs, it is obvious that the performance of all the vehicles at their respective maximum permissible speeds should be compatible to the average standard of track obtained in the field and these vehicles can, therefore, be considered to be compatible to each other among themselves. This is an important requirement on Indian Railways where goods trains as well as high speed passenger trains run on the same track.

1.13.4.3 Whenever a modification/improvement is made to the design features of any existing rolling stock or if the maintenance practices are relaxed resulting in variation in the average condition of maintenance, the worthiness of the rolling stock with changed characteristics has to be proved by oscillation trials, irrespective of whether the intention of changes is to increase the permissible speed or not.

1.13.4.4 The procedure for certification of maximum permissible speed for rolling stock, either for deciding the speed potential of any particular type of rolling stock, or for deciding the maximum permissible speed of a nominated train on any particular route, has been laid down in Policy Circular No.6, issued under Railway Boards letter no. 92/CEDO/SR/4/0 Pt. dt. 23.12.99 and modifications issued from time to time. This will have to be referred for detailed information. However for at a glance understanding, the flow chart at Annexure A at the end of the book may be referred to.

■ ■ ■

CHAPTER-2

SITE INVESTIGATION

2.1 GENERAL

Importance of a systematic and thorough investigation at the site of a derailment cannot be over emphasized, as, on it depends the ability to arrive at the most probable cause or causes of derailment.

It is necessary to proceed to the site quickly, notonly for reasons of protection of track, rescue, first aid, and restoration, but also for collection of all possible evidence before it is tampered with or destroyed willfully or otherwise.

On reaching a site of derailment, one normally encounters a scene of damage and disarray-damaged track, vehicles derailed and lying around in various positions, broken components of vehicles and track, etc. The question arises how and where to begin the investigation? The first important thing is to locate and examine the wheel mounting mark or marks at the initial point of derailment to identify the category of derailment i.e. whether the derailment is

- a sudden one, by wheel sets jumping the rails, or
- by flange climbing.

This enables the scope of investigation to be narrowed down.

Thus, precise measurements and critical and detailed examination of the wheel mounting marks should be made e.g. their length, nature, whether straight or curved or wavy, strong or faint, broken or continuous, single or multiple, etc. Preferably, photographs should be taken of such marks, not only on the rail, but also on the fastenings, sleepers and ballast.

In many cases, after the initiation of derailment, more vehicles derail before the train comes to a stop. This creates a multiplicity of wheel mounting marks and it becomes necessary to find out as to which wheelset derailed first. This needs a bit of 'Sherlock Holmes' approach involving matching of the lengths and nature of the various wheel trail marks with the position and orientation of the derailed vehicles, marks on the wheel treads and flanges damage to vehicles and wheel sets and corresponding tell-tale signs on the track e.g. spill of wagon contents, paint marks etc.

The wheel mark at the initial point of derailment should be examined to establish the category of the derailment.

2.2 SUDDEN DERAILMENT

This is characterized by a short mark, typically less than 60 cm, across the rail table (or just a scratch on the rail head edge or no mark at all but drop mark on the sleeper or ballast). It indicates that the derailing forces were high enough to suddenly force the wheel off the rail. One should, therefore, look for features that would cause sudden development of high flange forces. As mentioned in the beginning of this booklet, it is relatively much easier to arrive at the probable cause of this category of derailment.

Possible causes-acting alone or in combination-may be :

- sudden shifting of load
- resonant rolling, nosing or hunting
- sudden variation in draw bar forces induced by possible improper train operation; braking or acceleration
- broken wheels

- failure of vehicle or track components
- obstruction on the track
- entanglement with track of hanging parts of rolling stock.

(**N.B.** Above are only quoted as examples and the list is not exahaustive)

2.3 GRADUAL DERAILMENT BY FLANGE CLIMBING

A long mark, typically longer than 60 cm (in some cases it may be very long) on the rail table, indicates a gradual derailment by classical flange climbing. It implies that the derailing forces overcame the stabilizing forces and that this adverse situation persisted till the wheel climbed the rail, rode the rail head and slipped off. The derailing forces, however, were not high enough to cause a sudden derailment.

These derailments may be caused by a combination of very high flange force with heavy wheel load (in this case the wheel mark on rail would be very prominent) or a high flange force with lighter wheel load (wheel mark would be less prominent, and smaller in length).

Sometimes, derailment may be caused by wheel load becoming very small (e.g. due to severe track twist or spring defect), in combination with flange forces, which may be large or small. In such cases, wheel marks on rail may be short or even absent.

In these cases, it may be relatively more difficult to establish the cause. Yet, it is this category which Nadals formula finds its application in affording a means of going about the task of investigation.

The standard method of site investigation (in the context of arriving at the probable causes of derailment) should, thus, cover the following:

- Identification and detailed and critical examination of the wheel marks on the rail head, fastenings, sleepers and ballast, the wheel trail marks and the corresponding marks on the wheelset which derailed and other vehicles to identify the wheelset which derailed first and to establish the initial point of derailment.

- Detailed examination of wheel marks on the rail head, fastenings, sleepers and ballast at the initial point of derailment to establish the category of derailment.
- Investigation and recording of defects, condition and history of travel of the concerned rolling stock.
- Dynamic behaviour of such rolling stock on above track geometry.
- Investigation and recording of the operating features, and analysis of the dynamic behaviour of the rolling stock in reference to the same.

Discussion of rolling stock and track defects and operating features and analysis of their contribution towards increasing derailment proneness have been detailed in Chapters 3 to 8.

For the action to be taken at an accident site, reference is invited to the relevant chapters of the Permanent Way Manual, Accident Manual and Railway Boards letter no. 70-Safety (A & R) 21/4 dated 29-06-1971. Among other particulars and documents, the following need special mention.

2.4 PRESERVATION OF CLUES

Any component or feature, which bears evidence to the accident occurrence, has to be preserved at all costs.

2.5 ACCIDENT SKETCH

This should be prepared with great care so as to show all the relevant details of an accident site. A yardstick to judge whether the sketch is satisfactory or not is to imagine oneself in the position of a person who has not visited the site at all and whether one would then be able to get the complete picture of the accident from the sketch itself or not.

An example of an accident sketch is shown in **Fig**.2.1.

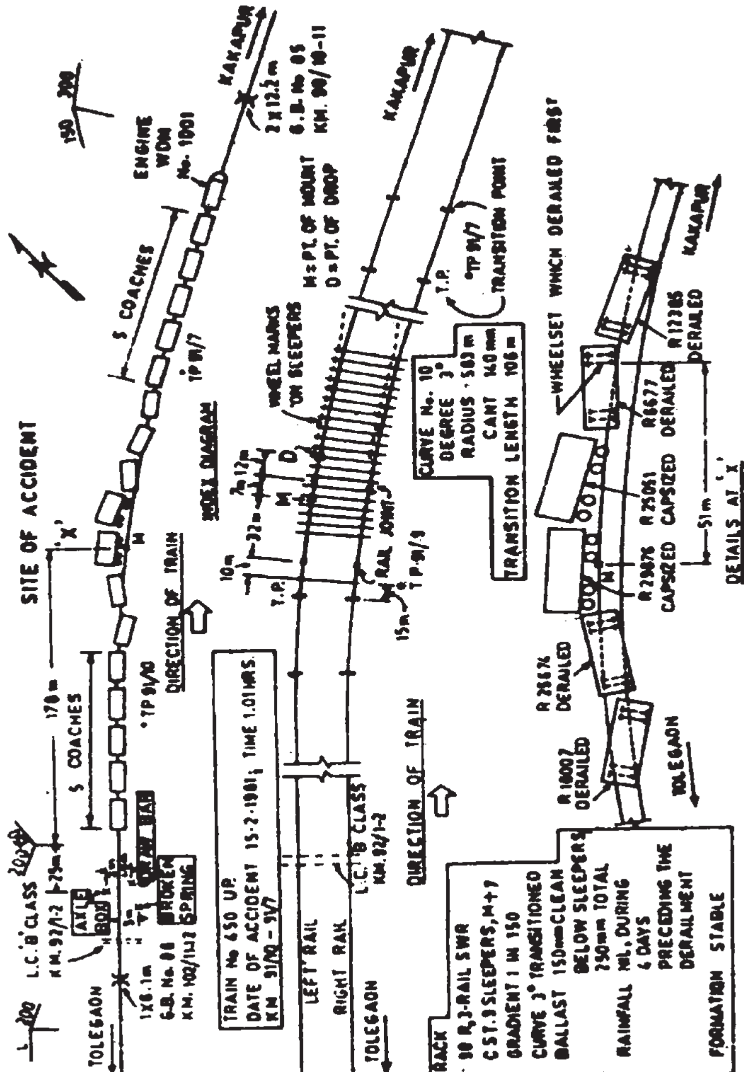


Fig. 2.1 Typical sketch of accident site

■ ■ ■

CHAPTER 3

ROLLING STOCK FEATURES AND DEFECTS

An understanding of the basic features and functions of various components of rolling stock and its suspension system enables a more rational identification of defects there in. Rolling stocks have evolved over Indian Railways over a very long period of time. Significant improvements have been made in the systems and components of rolling stock. It is an on-going process and improved rolling stock keep getting introduced regularly on the Indian Railways.

There are three distinct portions of rolling stock:

- wheelsets
- suspension system
- vehicle body

3.1 WHEELSET

Two wheels and an axle form a wheelset. A wheelset should have running stability and have safety against derailment by mounting. It should negotiate various track features viz. points and crossings, curves etc. safely and without damaging the track components.

3.2 SUSPENSION SYSTEM

Suspension system has two basic components, spring and damper.

3.2.1 SPRING

It is well known that springs improve riding comfort. A spring less vehicle would give a jarring ride to a passenger, whereas springs by absorbing shocks render the ride smoother. However, in railway vehicle, springs have a more important function which has direct bearing on safety. As it has been brought out in chapter 1, springs provide a vehicle with the ability to negotiate track twist without the wheel off-loading to too low a value. Thus, any defect which directly or indirectly affects the functioning of springs, makes the vehicle more derailment prone through the Q parameter as per Nadal's Formula.

The main bearing springs may be provided in a vehicle suspension in single stage or double stage.

3.2.1.1Single stage suspension

It has only one stage of springs between wheelset and vehicle body. The most common example is the ordinary 4 - wheeler wagon. Normally, freight stocks have single stage suspension. Some of the locomotives also have single stage suspension.

3.2.1.2 Double stage or Two-stage suspension

In this, springs are provided in two stages between wheelset and vehicle body, viz.

- Primary stage
- Secondary stage

3.2.1.2.1 Primary stage

This comprises the set of springs which bear on the axle boxes, directly or indirectly. Such springs are called primary springs and the suspension, primary suspension.

3.2.1.2.2 Secondary stage

This comprises the set of springs which, through a bolster, bear directly the weight of the vehicle body and transmit it to a bogie frame, which further rests on the primary springs. Such springs are called secondary springs and the suspension, secondary suspension.

Coaching stock normally has two-stage suspension. Some of the locomotives also have two-stage suspension.

3.2.2 DAMPING SYSTEM

Damping, too, has a comfort function, and a safety function. A spring when subjected to an excitation would continue to oscillate for a long time unless the oscillations are damped out. At relatively smaller amplitude of oscillation, mainly comfort level is adversely affected. Thus, damping system has an important role in ensuring comfortable ride.

In addition, as detailed in Chapter 1, damping has an important safety function. In case of vertical oscillations, without adequate damping in a suspension system, the amplitude of oscillations would tend to become very large, leading to dangerous off-loading during resonance. As discussed earlier, phenomenon of resonance is likely during actual run, depending on vehicle speed, wavelength of track irregularity and natural frequency of oscillation of the vehicle.

Thus, a defect in the vertical damping system would increase the incidence and extent of off-loading and, hence, the derailment proneness of the vehicle.

Similarly, lateral dampers have a role in controlling lateral acceleration and lateral (Y) forces. As discussed in chapter 1, lateral oscillations are significant at higher speeds. Therefore, lateral dampers are provided in rolling stock designed to operate at higher speeds. Defect in lateral dampers would increase the probability of derailment at higher speeds.

With springs and damping devices, a suspension system has to perform, in general, the following functions :-

- Hold the wheelsets (forming a rigid wheel-base) laterally and longitudinally, parallel to one another,
- Transmit the vertical load from vehicle body to the wheels, at the same time permitting unhindered relative vertical movement between the suspension system and the wheelsets to enable the springs and dampers to function freely.
- Transmit the longitudinal tractive and braking forces from vehicle body equally on the two wheels of a wheelset.
- Transmit lateral force from vehicle body to the wheel sets i.e. provide lateral restraint
- Permit bogie (or truck) to undergo unhindered but damped rotation and lateral sliding for negotiation around curves and turnouts.

N.B. Rolling stock can be categorised into the following:

Four-wheeled units

Vehicle body is supported on two axles which form the rigid wheel-base.

Six-wheeled units

Vehicle body is supported on three-axled rigid wheel base.

Bogie stock

Vehicle body of such stock is supported on two bogies (or called trollies), each of which may comprise two-axled rigid wheel-base or three axled rigid wheel-base.

3.3 VEHICLE BODY

Various designs of vehicle body, depending on functional requirement are available. In general, vehicle body should always meet with the requirements of Schedule of Dimension. Vehicle under frame should not have excessive vertical twist. Besides, the rectangular under frame should not be distorted into a parallelogram (as seen in plan).

We shall now discuss various rolling stock defects, analyzing their effect on derailment proneness. First, general defects which are more or less common to various types of rolling stock, will be dealt with. Defects particular to specific rolling stock will then be taken up.

N.B. It may however be mentioned that it is not possible to include all defects, tolerances, condemning limits, repair & maintenance practices, etc. in this booklet. Though many of the provisions have been included, stress has been laid more on explaining the manner in which rolling stock defects affect safety. For a complete description of defects, tolerances, condemning limits, repair & maintenance practices, etc., reference should be made to IRCA Conference Rules (referred to as IRCA CR hereafter), Part III for goods stock and part IV for coaching stock, Schedule of Dimensions B.G. & M.G., Manuals for Coaches and Wagons maintenance and related Instructions and circulars.

Under the chapter ‘Rejections’ of the IRCA Conference Rules, such of the clauses which stipulate the rejection limits for rolling stock parameters have been prefixed with the letter 'S'. Some of these have been reproduced under the relevant items of rolling stock in this booklet.

Besides, reports of Mechanical or Civil-and-Mechanical wings of RDSO, which detail the results of oscillation trials and other tests of respective rolling stock, are important references.

We start with wheelsets, taking up next suspension system and vehicle body.

3.4 DEFECTS OF WHEELSET

There are two methods of installation of wheel to the axle shaft – Shrink fitting and Cold press-on. Normally the Cold press-on method is adopted in manufacturing and Shrink fitting is recommended for maintenance.

Rejections (prefixed by ‘S’)

Goods and Coaching stock (B.G. & M.G.)

* Wheel shifted on axle
* Any prescribed tyre defect namely reduced tyre thickness, flange thickness, worn tread profile causing hollow tyre and deep flange, reduction of flange root radius, formation of sharp edge on flange and flat places on tread as shown in plates 59 to 66, and 40 to 46 of IRCA CR, Pt.III and IV respectively.
* Tyre loose, cracked or broken

Let us now take the wheel profile first. Earlier, IRS wheel profile was used on the Indian Railways as shown in **Fig.** 3.1. However, it was found that rail interaction with IRS wheel profile resulted in rapid wear of the flange and root of the flange during initial stages till a wear adapted/worn wheel profile was obtained. As wheel wear subsequent to the worn wheel profile obtained in service was considerably less, worn wheel profile, as shown in **Fig.**3.1(c), has since been adopted even for new wheels. Condemnation limits for these wheels remain similar to those of IRS wheel profile according to IRCA Part III & IV.

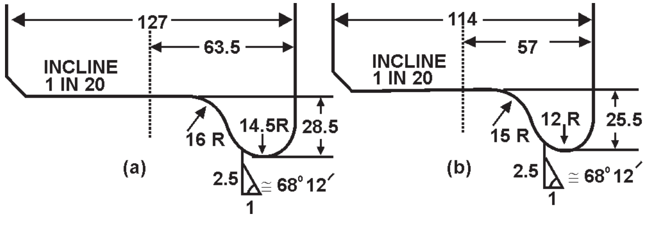


Fig. 3.1 IRS wheel profile (a) B.G. Wheel.(b) M.G. wheel

As wheel gets worn, they are profiled in workshop to standard intermediate profiles. There are a total of 9 such profiles for coaching stock, having flange thickness of 28 mm to 20 mm, with all the integral values in between. Out of these, for coaches of passenger trains with velocity 110 kmph and above, profiling is to be done as per profiles of 25 mm flange thickness and above only, Whereas other coaches can be profiled as per any one of these 9 profiles.

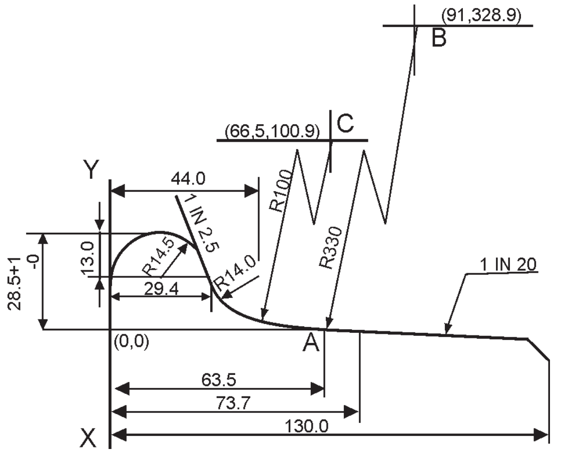


Fig.3.1 (c) Wear adapted or Worn wheel profile (B.G.)
(Sketch – 91146, for Main line, EMU, DMU, MEMU coaches)

When wheel profile gets worn, it may reach condemning limits in reference to any one or more of the following conditions:

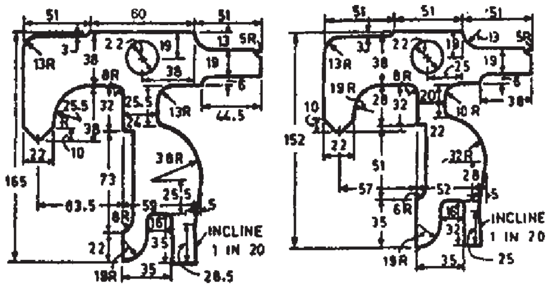
i) thin flange ii) sharp flange iii) worn root

iv) deep flange v) hollow tyre or false flange

vi) flat tyre

The above terms denote that the wheel has reached the condemning limit in the particular parameter.

The dimensions which delineate the condemning limits have been given in the following paras. The condemning limits are checked by a tyre defect gauge illustrated in **Fig.**3.2. Each type of wheel profile has its own tyre defect gauge. Tyre defect gauge are of 'Go- No-go' type, as they only indicate whether a particular limit has been exceeded or otherwise, and not the actual measurement.



(a) (b)

Fig.3.2 Tyres defect gauge (a) B.G. (b) M.G.

3.4.1 THIN FLANGE

When the flange thickness (of B.G. or M.G.wheel) reduces to less than 16 mm, the condition is called thin flange (**Fig.**3.3) Thickness of a flange is normally reckoned at a distance of approximately 13 mm from the flange tip for B.G. or M.G. wheel. For coaches of mail/express (speed 110 kmph and above) this limit is 22 mm instead of 16 mm.

3.4.1.1 Effect on Safety

A thin flange implies greater play between the wheelset and the track which increases derailment-proneness as under:

- Lateral oscillations increase due to greater play, adversally affecting Y and Q
- Angularity of axle increases

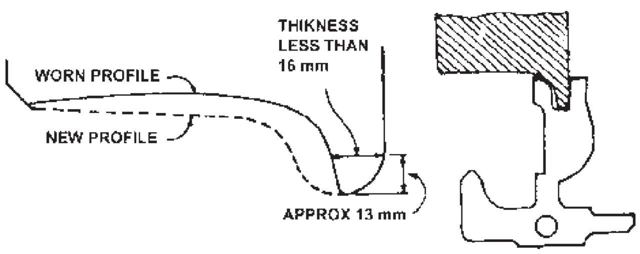


Fig. 3.3 Thin Flange

Besides, if the flange is too thin, the back of the wheel may damage the tongue rail while passing through the switch flange way gap of points and crossing particularly with curved switches. Minimum switch flange-way gap is calculated from the formula :

Switch flange-way gap = Track gauge - (minimum wheel gauge + minimum flange thickness) (3-1)

If the flange is thinner than 16mm, it would require a correspondingly greater flange-way gap which may not be actually available at site, resulting in possibility of damage to the tongue rail.

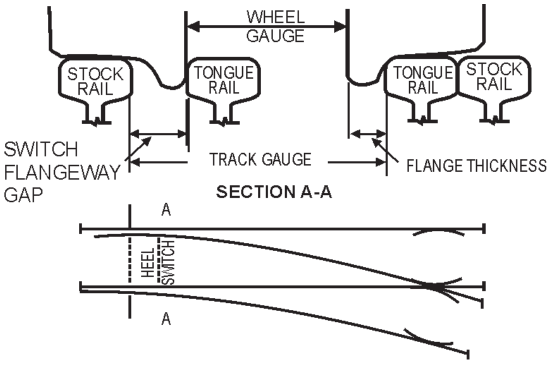


Fig. 3.4 Switch Flange-way gap

3.4.2 SHARP FLANGE

When the radius of flange tip reduces to less than 5 mm (B.G. or M.G.), the condition is called sharp flange (**Fig.**3.5)

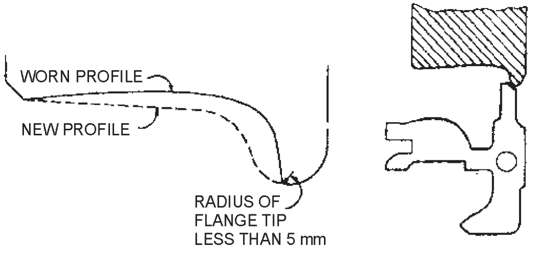


Fig.3.5 Sharp flange

3.4.2.1. Effect on safety

Derailment proneness increases through the following:-

- μ increases due to change in geometry of the wheel flange, resulting in greater biting action of the wheel flange along the rail head edge;
- Positive eccentricity increases even with the same value of axle angularity;
- a sharp flange may split open slightly gaping points while traveling in facing direction, or may mount over a slightly chipped tongue rail which presents a square surface to an approaching wheel.

3.4.3 WORN ROOT

When the radius of the root curve reduces to less than 13 mm (B.G. or M.G.) the condition is called worn root (**Fig.**3.6). It is attendant with increase in the flange slope angle i.e. increase in value of β.

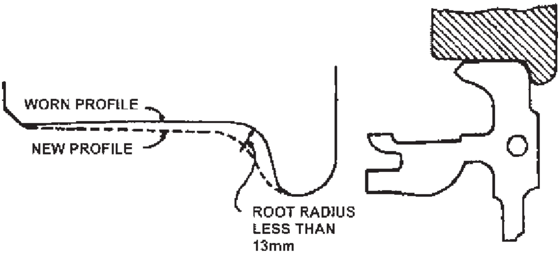


Fig.3.6 Worn root

3.4.3.1 Effect on Safety

- Positive eccentricity increases even with the same value of axle angularity.
- Effective μ increases owing to reduction in the taper of the wheel flange i.e. owing to increase in value of ß.

3.4.4 DEEP FLANGE

When the depth of the flange, as measured from theflange tip to a point on the wheel tread (63.5 mm away fromthe back of B.G. wheel or 57 mm away from back of M.G.wheel) becomes greater than 35 mm (B.G.) or 32 mm (M.G.)the condition is called deep flange (**Fig.**3.7)

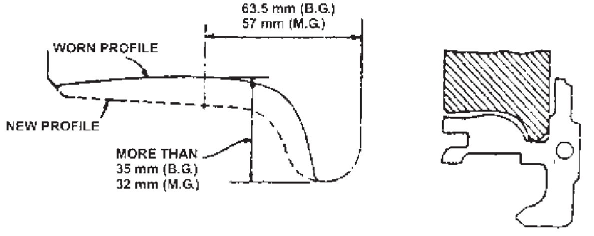


Fig. 3.7 Deep flange

3.4.4.1 Effect on Safety

In this condition, the wheel flange coupled with vertical wear of the rail head would tend to ride on the fish plates and check/ distance blocks and thus strain and damage the track components.

3.4.5 FALSE FLANGE/HOLLOW TYRE

When the projection of the outer edge of the wheel tread below the hollow of the tyre exceeds 5 mm then the outer edge of the wheel is called false flange, and the worn tread is called hollow tyre (**Fig.**3.8)

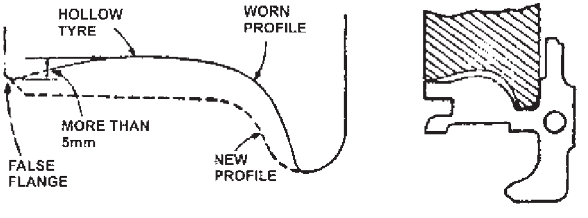


Fig.3.8 False flange / hollow tyre

3.4.5.1 Effect on Safety

A false flange may spilt open points while travelling in trailing direction, as the false flange may tend to get wedged in between the tongue rail and the stock rail as shown in **Fig.**3.9.

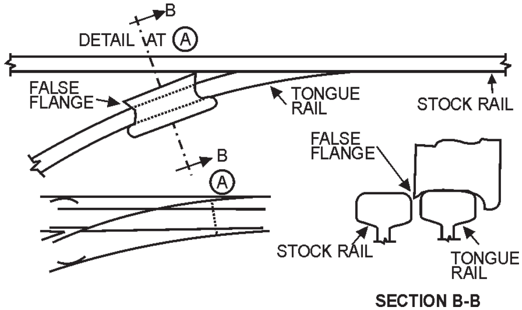


Fig.3.9 Wheelset with false flange negotiating switch portion in trailing direction

While negotiating the crossing portion, the wheel going across the wing rail would get lifted as instead of travelling on the tread portion it would travel on the false flange during the duration the wheel travels across the wing rail. This will make the wheel to suddenly lift up and drop down near nose of crossing. This may cause damage to the crossing portion (**Fig.**3.10)

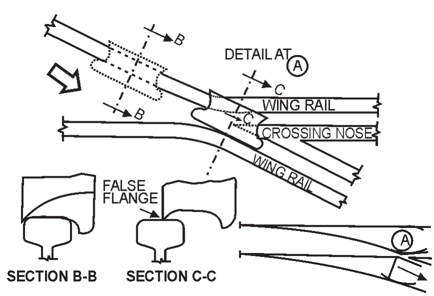


Fig.3.10 Wheelset with false flange negotiating crossing portion

Also, while approaching a crossing, false flange on field side of wing rail will have a tendency to guide the wheel, resulting in excessive strain on the opposite check rail by the other wheel of the wheelset. This would result in increase in check rail wear and may even damage the bolts, thus, increasing the chances of derailment.

Particularly, on diamond crossings, where the check rail guidance is already very small, such situation would render the wheel appreciably derailment-prone.

3.4.6 FLAT TYRE

The maximum permissible length of flat on the wheel tyre is :

Broad Gauge :

i)	Goods stock	60 mm
ii)	Coaching stock (ICF, BEML)& Locos	50 mm
	Meter Gauge (all stock)	50 mm

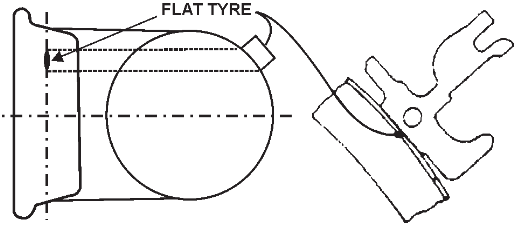


Fig. 3.11 Flat tyre

Wheel flats cause high dynamic augment of wheel load on rail. This increase mainly depends on depth/ length of flat and speed of train, and may be as high as 3.5 times the static wheel load. Such high impact loads have been found to significantly increase the probability of rail fracture, particularly in presence of crack in rail and in combination with high tensile stresses. Wheel flats are generated due to sliding of wheel and, hence, are likely to be more prevalent in rolling stock running at locations such as ghat section, sub-urban section.

Wheel Impact Load Detection (WILD) equipment has been a well established system for detection of high impact load on rail on foreign Railways running high axle load freight traffic. This equipment has also been developed on Indian Railways and its installation has been started since 2007. It has been installed at 15 locations (as of 2011) and there is a target of installing more of this equipment, so as to cover the Indian Railways network thoroughly.

WILD equipment identifies the wheels which are generating very large impact load. Based on the augment of impact load, Maintenance and Critical Alarms are generated. Railway Board have stipulated the action required to be taken on these alarms, such that undue damage is avoided to the rail. These need to be fully implemented in the field.

In addition, more awareness should be inculcated in the concerned staff such that rolling stock generating repetitive hammering sound are reported promptly.

3.4.7 Wheel profile defects of locomotives

Wheel wear of locomotives are to be measured at a frequency as specified in Inspection Schedule for locomotive. Wear required to be measured are flange wear, root wear and total tread wear (comprising of tread wear plus wear due to flat on tread). Length of wheel flat is also to be measured.

Wear measurement gauge is as shown in figure 3.12.

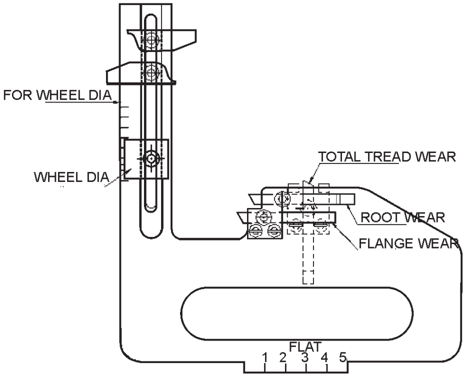


Fig. 3.12 Tyre defect gauge for locomotive

Location of measurement and limits of wear are shown in figure 3.13. Specific limits of wear for different locomotives are stipulated in RDSO Instruction Bulletin No. MP.IB.BD.02.16.01 (Rev.01) dt. 31.12.2009.

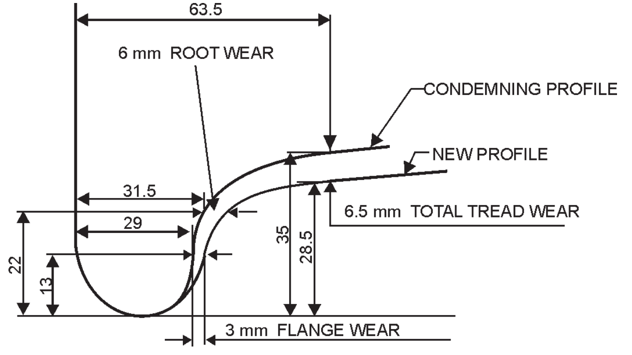


Fig. 3.13 Condemning wear limit

Effect of wear in excess of specified limits should be analysed in the same manner as done for wheel profile defects of wagon/ coach. It follows that excessive wear resulting in sharp flange or false flange should be analysed for its contribution to derailment.

3.4.8 OTHER WHEEL DEFECTS

The wheels sets shall be visually inspected and action taken as detailed in the following paras.

During investigation of derailment, physical damage to wheel and its contribution to failure in guiding of wheel along track should be analysed. Some of the obvious defects could be broken axle/ wheel, shifted wheel on axle, burnt rim, cracked/ broken flange, cracked/ broken plate, cracked hub and built-up tread.

3.4.8.1 Shattered Rim (Fig. 3.14)

A wheel with a fracture on the tread or flange must be withdrawn from service. This does not include wheels with localized pitting or flaking without presence of any rejectable condition.

Fig.3.14 Shattered Rim

3.4.8.2 Spread Rim (Fig. 3.15)

If the rim widens out for a short distance on the front face, an internal defect may be present. Spreading of the rim is usually accompanied by a flattening of the tread, which may or may not have cracks or shelling on the tread. Such wheels must be withdrawn from service.

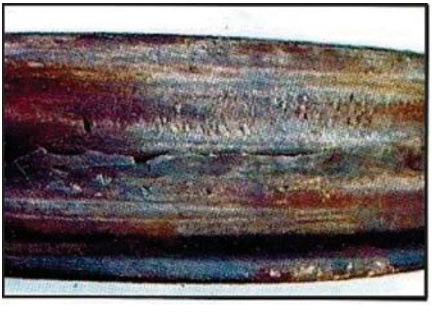


Fig.3.15 Spread Rim

3.4.8.3 Shelled Tread (Fig. 3.16)

Shelling can be identified by pieces of metal breaking out of the tread surface in several places more or less continuously around the rim. Shelling takes place when small pieces of metal break out between the fine thermal checks. These are generally associated with small skid marks or Chain Sliding.

Fig.3.16 Shelled Tread

Such wheels should be withdrawn from service and sent to workshop for reprofiling.

3.4.8.4 Thermal Cracks (Fig. 3.17)

Thermal cracks appear on a wheel tread due to intense heating of the wheel arising out of severe brake binding. Such cracks occur on the tread and generally progress across the tread in a transverse and radial direction.

Fig.3.17 Thermal Cracks

Whenever such a crack becomes visible on the outer face of the rim or tread crack has reached the outer edge, the wheel should be withdrawn from service.

3.4.8.5 Heat checks (Fig. 3.18)

Thermal cracks are deeper and need to be distinguished from superficial cracks visible on tread or adjacent to the braking surface. These are called heat checks, which are usually denser than the thermal cracks. Heat checks are caused on the tread due to heating and cooling cycles undergone by the wheel during normal braking. Such wheels do not need to be withdrawn.

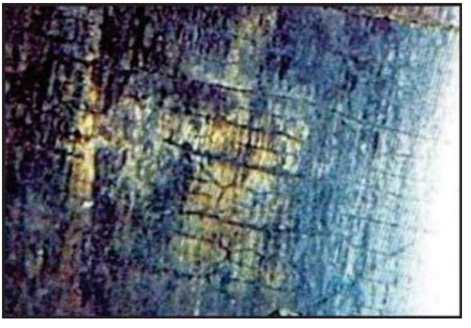


Fig.3.18 Heat Checks

3.4.9 DIFFERENCE IN WHEEL DIAMETERS

The wheel diameter is measured on the tread at a distance of 63.5 mm from the back of the wheel in the case of B.G. wheel and at a distance of 57 mm in the case of M.G.wheel. Two measurements viz. across the quarter points, should be taken for each wheel (**Fig.** 3.19)

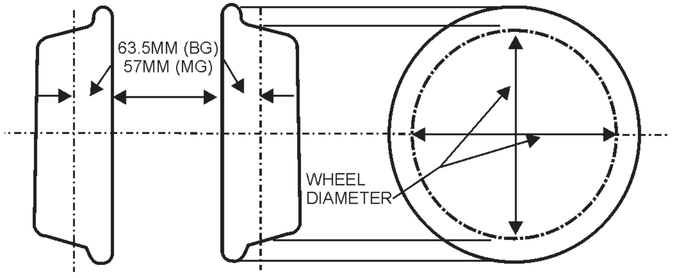


Fig. 3.19 Measurement of wheel tread diameter

When wheels are changed/turned, it should be ensured that the variation in tread diameters does not exceed the maximum permissible limits indicated in Table 3.1.

Table 3.1 Wheel diameter difference

Wagon IRCA Part III, 1982 – Rule 2.8.14.2	Same axle (mm)		Same trolley		Same wagon (mm)	
	BG	MG	BG	MG	BG	MG
4-wheel trolley	0.5	0.5	13	10	25	13
4-wheel unit	0.5	0.5	-	-	25	13
Coach (IRCA Part IV, 2003 – Rule 2.8.3						
4-wheel bogie (IRS, non-IRS)	0.5	0.5	13	10	13	10
4-wheel bogie (ICF, BEML)	0.5	0.5	5	5	13	10
6-wheel bogie	0.5	0.5	6	6	6	6
4-wheel unit	0.5	0.5	-	-	25	13

Difference in diameter on the same axle would result in longer distance being traversed by the larger diameter wheel (compared to the smaller wheel) on account of pure rolling. Thus, the axle will have a tendency of persistent angular movement. Larger the wheel diameter difference, larger will this tendency be, increasing derailment proneness.

Variation in wheel diameter on same bogie will affect distribution of load among different wheels, resulting in off-loading. Variation in wheel diameter on a coach/ wagon will affect even distribution of load and would result in off-loading.

Limits of wheel diameter difference mentioned in table 3.1 above relate to changing/ turning of wheel in workshop. Service limits of these parameters for coach and wagon have not been specified. However, service limits for wheel diameter difference have been stipulated for locomotives vide RDSO 'Instruction Bulletin No. MP.IB.BD.02.16.01 (Rev 01) dated 31.12.2009'. The service limits for WAG9 loco, for example, are 2.5 mm on same axle, 4.0 mm on same bogie and 20.0 mm on same loco. In view of the above discussion, investigation of derailment cases, involving even coach/ wagon, may be done keeping these limits and their effect on safety in mind.

3.4.10 WHEEL GAUGE

Wheel gauge is the distance back-to-back of the wheels on a wheelset. Wheel gauge should be checked at quarter points (**Fig.**3.20).

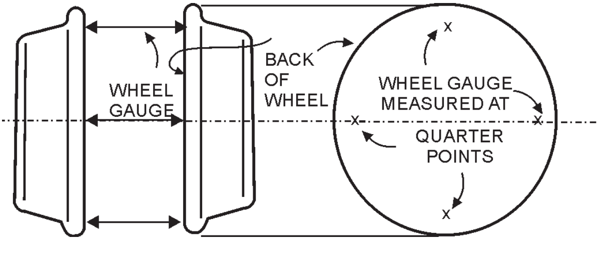


Fig. 3.20 Wheel gauge

No variation, whatsoever, is permitted among the values of wheel gauge measured at quarter points. A variation in the values of wheel gauge measured at quarter points indicates a bent axle.

A bent axle on motion will start wobbling causing severe vibrations between the bearing and the journal and, consequently, may result in damage to bearing and hot axle condition. This would result in persistently angular run and may even cause fatigue and fracture of axle.

Subject to the above condition, the actual value of wheel gauge can vary between the following tolerances given in Table 3.2 (Ref. IRCA Pt.III , 01.05.1982 – Rule 2.8.7 & IRCA Pt IV, 01.06.2003 - Rule 2.6.8)

Table 3.2 wheel gauge

Goods & coaching stock	BG (mm)	MG (mm)
Standard	1600	930
Maximum	1602	932
Minimum	1599	929

In case wheel gauge is more than the permissible tolerances, there would be a possibility of a relatively new wheel hitting the nose of crossing, as wheel gauge is one of the parameters which decide the required clearance at check rail opposite nose of the crossing (**Fig.** 3.21)

Check rail clearance < Track gauge - (Maximum wheel gauge + flange thickness of new wheel) (3-2)

In case wheel gauge is more, check rail clearance would have to be less than what is actually provided as the standard, if damage to nose is to be avoided.

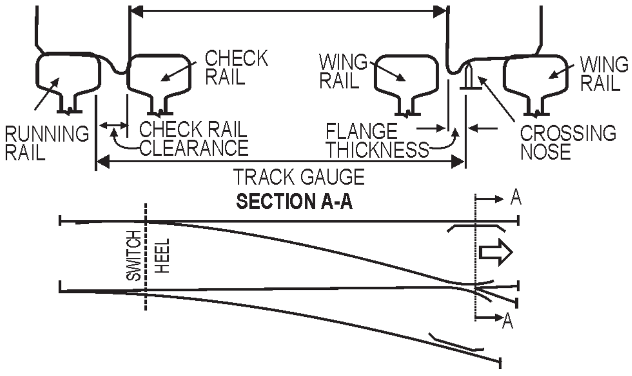


Fig. 3.21

If, however, wheel gauge is less than the minimum value, both of the wheel many hit the wing rail. Also, there would be a possibility of wheel hitting at the back of the open tongue rail while passing through the switch flange-way gap, and thus, damaging the tongue rail (see expression 3-1).

3.5 JOURNAL

The portion of axle on which the bearing rests, is called the journal . The cross-sectional centre of the journal has to be coaxial with those of the axle and wheel, otherwise journal centre itself would revolve causing severe vibrations in the bearing-journal assembly leading to hot axle.

In case of plain bearing, journal gets worn with passage of traffic. However, this is not the case with roller bearing. Minimum permissible limit and nominal size of journal diameter for plain bearing and roller bearing, respectively, are stipulated corresponding to gross axle load in IRCA Rules.

3.6 AXLE BOXES

Axle boxes are basically of two types :

- Plain bearing axle box
- Roller bearing axle box

3.6.1 PLAIN BEARING AXLE BOX

In this, a bearing brass with its bottom portion formed of white metal ensures, along with lubrication, a low coefficient of friction between the bearing and the revolving journal. The axle box top plate rests on the bearing with a slipper plate between for adjustment of levels of the axle box top in the event of journal diameters and bearing thickness varying and also for ease of removal of the bearing.The axle box contains oil-soaked packing for lubrication of the journal.

Plain bearing has been phased out from most of the rolling stock on Indian Railways. Hence, this would not be discussed further.

3.6.2 ROLLER BEARING AXLE BOX

Roller bearings are of three types - cylindrical, spherical and tapered.

3.6.2.1 Cylindrical Roller Bearing

Rollers of these bearings are cylindrical in shape. These are provided in BEML Coaches, some engines (viz. WAG7, WDM3D etc.) and some wagons viz. BOX, BCX, BRH, etc.

3.6.2.2 Spherical Roller Bearings

In this type of bearing, diameter of rollers at the ends are slightly lesser than diameter at the centre (**Fig.** 3.22).

The bearing consist of an outer ring having a continuous spherical race way within which operate two rows of barrel shaped rollers, which in turn are guided by an inner ring with two raceways separated by a centre rib. The spherical roller bearings have self aligning properties and, therefore, can automatically adjust to deviations in the centre line of the axle.

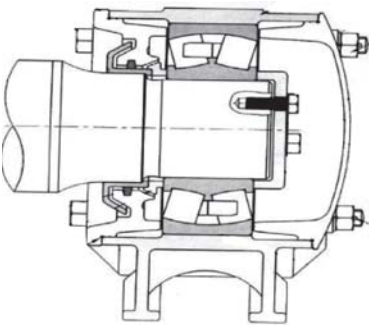


Fig. 3.22 Spherical roller bearing

The axle boxes can tilt up to 2^0. The roller bearings are so designed that they can get out of alignment by 2 to 3 mm without any detriment to their performance.

These type of bearings are provided in ICF all coil coaches, some engines and some wagons viz. BOBS.

3.6.2.3 Tapered Cartridge Roller Bearings

These bearings are designed to take both radial and lateral loads. It consists of 2 rows of tapered roller bearings kept apart by a spacer ring over the inner race (Figure 3.23). During the run the outer race is held firm in the adaptor and the inner race rotates along with the axle and the roller bearings, rotating about the axis of journal. But moment of any defect, the outer ring starts rotating under the adapter producing a loud whistling noise and thus saves the train from any serious disaster and also saves the wheel tread from skidding etc. These are provided in some engines, some wagons viz BOXN, BCN etc. and LHB coaches.

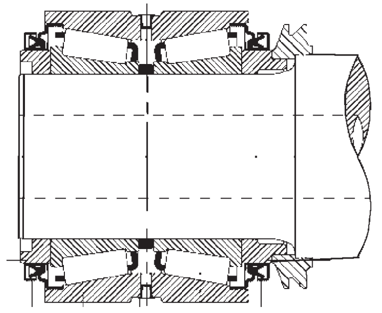


Fig. 3.23 Tapered cartridge roller bearing

3.6.3 PRO AND PR PLATES

On the left hand end of each sole bar or on the wagon body, PRO or PR plates are fitted (**Fig.**3.24) PRO plate is fitted on the stock having plain bearing axle boxes, and PR plate on the stock having roller bearing axle boxes.

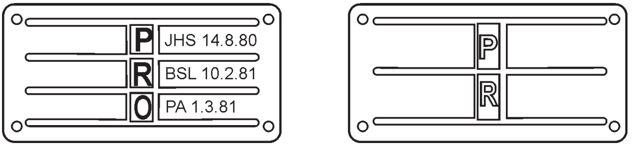


Fig. 3.24 PRO and PR plates

The letters stand for the following :
Plain bearing axle boxes :

P : Periodical overhaul of axle boxes

R : Re-packing of axle boxes

O : Oiling of axle boxes.

Roller bearing axle boxes :

P : Periodical overhaul of axle boxes

R : Re-examination and greasing of axle boxes

In case, it is not possible to attach the standard PRO or PR plates to the sole bar, stenciling of PRO and PR particulars on the sole bar is also permitted.

Against the relevant letters, the code name of the station and date on which particular attention has been given are stenciled. When periodical overhaul is carried out, date of repacking is not stenciled against R as the date and code stenciled against P serves the purpose.

3.6.4 AXLE BOX DEFECT

Roller bearings need much lesser maintenance attention as compared to plain bearing. However, following defects of roller bearing would affect its performance, necessitating its replacement

- Pitted or flaked roller tracks and rollers
- Cracked or deformed or badly worn out cage
- Cracked inner or outer ring
- Scored or damaged outer surface of the outer ring
- Indentation on rings or rollers
- Scoring of roller tracks or rollers
- Rust/corrosion, damage or excessive fretting corrosion
- Rings exhibiting deep straw or blue or purple colour indicating
- Heat effect
- Excessive or less radial clearance.

The above defects are likely to result in increased friction and hot axle condition. Hot axle is a condition under which the actual temperature of axle box becomes higher than normal operating temperature, which is about 70 deg centigrade.

Rejections of axle box as per IRCA Rules (prefixed by S) are as below

Goods stock (B.G. & M.G.) :

* Axle box cracked below the journal centre or with a crack through both lugs of an axle guard jaw or groove
* Axle box broken
* Hot box
* Axle box face plate deficient or with a rivet broken or deficient

Note : When an axle box face plate rivet is found broken or deficient the box must be examined from inside to ensure correct assembly of fittings and contents.

Coaching stock (B.G. & M.G.) :

* Axle box or lug/wing broken
* Hot box
* Plain bearing axle box face plate non-standard,deficient or gaping or if the bolt/rivet is deficient.

3.6.5 AXLE GUARD AND AXLE BOX GROOVE

Axle guard horn cheeks passing through the axle box grooves (**Fig.**3.25 and **Fig.** 3.26) perform three of the important functions referred to while discussing the functions of the suspension system at the beginning of this chapter :

- to hold the axles in a rigid wheel-base
- while performing the above function, to permit unhindered relative vertical movement between the vehicle body and the axle, to enable the springs to deflect and function freely
- to transmit the tractive and braking forces to the axle

Fig. 3.25 Suspension system of 4-wheeler.

To have maximum life out of the axle guards, a piece called horn cheek is rivetted to the axle guard and it is this horn cheek which passes through the axle box groove and rubs against the groove surfaces. On wearing out, the horn cheek can be replaced by a new one without the need of replacing the entire axle guard.

The horn cheek-axle box groove guidance, or a similar arrangement, is the most common type of guidance and is provided in all freight stock and in some locomotives (some special guidance arrangements will be discussed under the relevant rolling stock in which they have been provided).

In some of the rolling stock, e.g. 4 wheelers, the axle guards on either side of an axle box are connected together by a bridle bar below the axle box. When axle guards transmit the tractive or braking forces, only one of them can bear against the axle box groove wall. Bridle bar enables the two axle guards to share transmission of tractive and braking forces, apart from functioning as a connecting member between the two axle guards, imparting rigidity. In some rolling stocks where axle guards are quite strong, bridle bar may not be necessary.

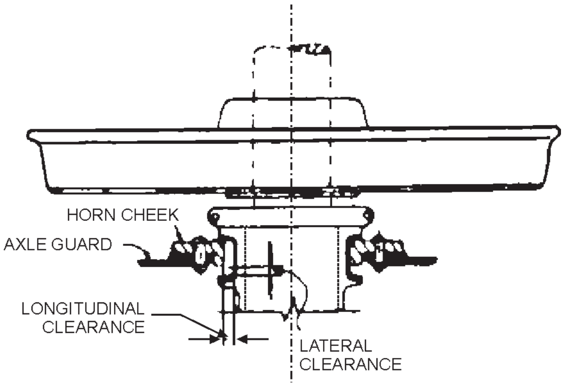


Fig. 3.26 Lateral and longitudinal clearances (plan)

3.6.5.1 Clearance between the horn cheek and axle box groove

Lateral clearance

The minimum and maximum permissible total lateral clearance between the axle guard and the axle box groove or the horn cheek and the axle box with plain bearing shall be as under :

- *Goods and coaching stock bogie & 4-wheelers (B.G.&MG)*

Minimum 6 mm	(for shops, after POH)
Maximum 10 mm	(for maintenance depots, when a coach is received in sick line)

- *BOX wagons :*

  Minimum 20 mm

  Maximum 25 mm

- *CRT Wagons (RDSOs L.No.MW/CRT dtd. 02.12.93)*

  Nominal 12 mm

Maximum 18 mm

Longitudinal clearance

The minimum and maximum permissible total longitudinal clearance between axle guard and the axle box groove or the horn cheek and axle box with plain bearing shall be as under:

- *Coaching stock bogie & 4 wheelers (B.G. & M.G.)*

  Minimum 3 mm (for shops, after POH)

  Maximum 10 mm (for maintenance depots, when coach is received in sick line)

- *Goods stock (B.G. & M.G.) :*

  Not specified, but axle box should not work out.

- *BOX wagons :*

  Minimum 12 mm

  Maximum 18 mm

If the above mentioned clearances are exceeded, their effect on safety is the same as that of increased play between wheelset and track viz.

- increased lateral oscillations, leading to adverse effect on Y & Q,
- increased angulartiy of axle.

REJECTIONS (Prefixed by S)

Goods and coaching stock (B.G. & M.G.)

* Axle guard worn or expanded sufficiently to permit either leg to work clear of its groove in the axle box
* Any portion of the axle guard horn cheek broken or deficient

* Axle guard so bent as to prevent free movement of axle box
* Axle guard cracked unless repaired according to the relevant IRCA Conference Rules, Part III or IV
* One or more rivets deficient or broken or of wrong size
* Any axle guard leg shaking due to slack rivets
* Bridle broken or deficient
* Bridle without jaws turned ends Additional for coaching stock

Additional for coaching stock

* Two or more rivets slack in any one leg of an axle guard
* Axle guard bridle without turned ends or of wrong size.

With track irregularities and vehicle oscillations, the spring undergoes deflections with consequent relative vertical movement between axle guard and axle box groove. That is, if the movement of axle guard is prevented, it in effect would prevent the deflection of the spring and render the spring ineffective, thus affecting safety (see Chapter 1, Effect of track or vehicle twist on Q).

3.7 SPRINGS

Springs commonly in use are of three types :

- leaf or laminated spring
- helical spring
- air spring

3.7.1 LEAF OR LAMINATED SPRING

A number of plates of graded lengths, held together by a spring buckle, form what is called a leaf spring or laminated spring. The ends of the longest plate are bent into eyes. (**Fig.**3.27)

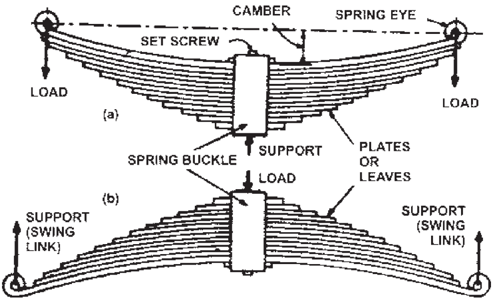


Fig. 3.27 Leaf or laminated spring (a)normal (b) inverted

3.7.1.1. Spring gear defects and tolerances

The deflection behaviour of a leaf or laminated spring is determined by checking its

- free camber, and
- working camber.

Free camber

Free camber when measured of a new spring, should in no case be less than the design value stipulated for that particular type of spring. However, the measured free camber may be more than the design value by certain tolerance. In general, tolerances of measured free camber from design camber are :

i)	Loco and carriage springs	(- 0, + 3) mm
ii)	Goods stock, BG	(-0, +6) mm
iii)	Goods stock, MG	(-0, +5) mm

Working camber

For measuring the working cambers of the springs, the vehicle should be made to stand on a level track and then loaded ensuring that the load is not eccentric. The vehicle should preferably be loaded upto the full carrying capacity.

In the case of a four wheeler wagon, the difference in working camber between any two of the 4 springs shall not exceed 13 mm. However, as already brought out in Chapter 1, *Effect of Track or Vehicle Twist on Wheel off-loading,* if the defect is in two diagonally opposite springs the contribution towards derailment-proneness is much more as compared to when the defect is in only one spring.

3.7.1.2 Shackle plates and scroll iron

The common assembly by which a vehicle e.g. four- wheeler goods wagon, is suspended from the ends of a laminated spring is what is known as shackle suspension (**Fig.** 3.28).

For such goods and coaching stock suspensions, stipulated clearances between spring eye and sole bar and lateral clearance between shackle pin and shackle plate and scroll iron/ eye of spring are to be maintained.

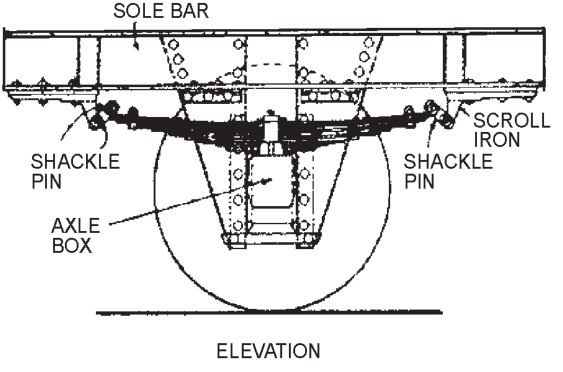


Fig. 3.28 Shackle Suspension of four-wheeler wagon

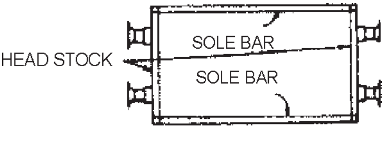


Fig. 3.29 Plan of wagon underframe

3.7.1.3 Spring and spring gear defects– rejections (prefixed by S)

Goods and coaching stock (B.G. and M.G.)

* Any plate of a laminated bearing spring or any coiled bearing spring cracked or broken

  Additional for coaching stock – Any coil spring fully compressed or broken, subject to

  i) Even one broken coil spring detected at originating/ destination station

  ii) Even one broken coil spring detected en-route with excessive tilt

  iii) If only one broken coil spring is detected en-route, without excessive tilt of axle box, then the coach may be allowed to continue journey till destination at a speed not exceeding 100 kmph

* Bearing spring buckle loose, broken, cracked and/or packing plate loose or deficient
* Any plate or buckle loose and/or displaced from its central position by 13 mm or more
* Bearing spring buckle not sitting square in the axle box housing or crown packing where fitted
* Flange of any wheel within 25 mm of the bottom of a wagon
* Incorrect type of bearing spring for the particular design of wagon
* Scroll iron fractured, deficient of a rivet or fitted with a loose or wrong size rivet
* Scroll iron shifted or out of alignment by more than 25 mm
* Shackle or pin cracked, deficient, broken or of wrong size
* Bearing spring hanger cracked or broken or nut or jib

cotter deficient or defective

* Bearing spring shackle pin not fitted with a split cotter

* On Metre Gauge wagon, bearing spring shoe fractured or with a rivet, bolt or stud broken or deficient or bolt or stud of wrong size

* Bearing spring eye or shackle plate touching the sole bar (in static or dynamic condition)

Note - This defect in dynamic condition can be confirmed by the presence of rubbing marks at the bottom surface of the sole bar. When a laminated spring deflects, it elongates thereby causing the shackle plate to move, as illustrated in **Fig.**3.30. If movement of shackle plate is prevented, in effect, it prevents deflection of spring and renders it ineffective. This increases derailment proneness..

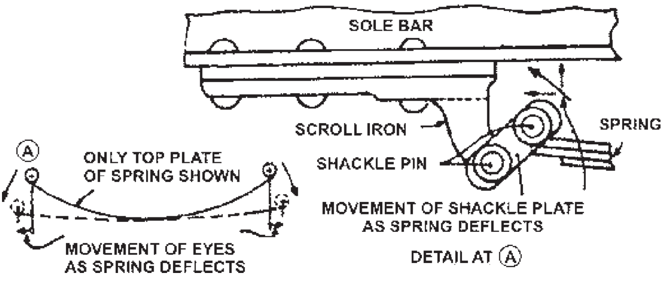


Fig. 3.30

Additional for coaching stock

* A variation in the type of bearing spring under the same vehicle, i.e. difference in the number, thickness or width of spring plate only. It is, however, permissible for the top plate of a bearing spring to be thicker by 3 mm, than the other plates of other spring and the top plates of the spring of a vehicle.

Note : This shall not apply to TLRs which are fitted with springs of different sizes on the two trolleys.

* Side control spring (leaf type) and/or drag link broken or deficient on BEML type.

3.7.1.4 Stiffness of the spring

It should be checked whether the spring leaves are so rusted as to render the spring stiff as this condition would reduce the ability of the vehicle to negotiate track twist without the residual wheel load dropping to too low a value.

3.7.2 HELICAL SPRING

A rod or wire coiled with a certain pitch, forms what is known as Helical spring. (**Fig.** 3.31).

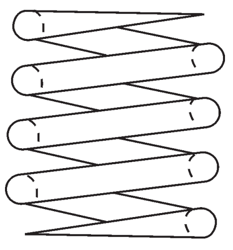


Fig. 3.31 Helical spring

When an outer coil has one or more inner concentric coils of smaller diameters, it is called a nest of springs.

3.7.2.1 COMMON DEFECTS IN HELICAL SPRINGS

Following are the common defects in helical spring

- cracked or broken spring
- shifted spring
- spring fully compressed
- length of spring beyond limit
- loss of elasticity, resulting into the spring getting fully

compressed in dynamic condition

- Rusted spring

Instructions for maintenance of helical springs of coaching stock are available in 'Handout No. C -8419 (rev. 1)' of November, 1996 by RDSO. In addition to maintenance procedure, categorization of springs based on their height under a specified load is also included in these instructions. Similar details are included in Wagon Maintenance Manual for wagons and Maintenance Instructions for various locomotives. Same category of spring is to be provided on the bogie, although category may be different for primary and secondary suspension, in order to limit wheel off-loading.

3.7.3 AIR SPRING

Air (pneumatic) suspension (**Fig.** 3.32,) is a suspension where properties of air are used for cushioning effect. Enclosed pressurised air in a pre-defined chamber called air spring, made up of rubber bellow and emergency rubber spring, provides various suspension characteristics including damping. The air spring system consisting of air reservoirs and pipes is shown in **Fig.** 3.33.

Air springs are height-controlled load leveling suspension devices. With changing loads, air spring reacts initially by changing the distance between air spring support and vehicle body. The height monitoring valve (called leveling valve) is in turn actuated, either taking the compressed air to air spring or releasing air pressure from it to the atmosphere. This process continues until the original height is restored. This mechanism ensures a constant floor height on coaches provided with air springs, irrespective of the load. This greatly reduces problems associated with low buffer and coupler heights. The mechanism is schematically represented in **Fig.** 3.34.

In case of coil spring, deflection is proportionate to the load. Therefore, under high payload, space constraint becomes critical, requiring use of stiffer springs resulting in unsatisfactory ride behavior and reduced speed potential. Air springs, through

Fig. 3.32 Air Spring in EMU Coach

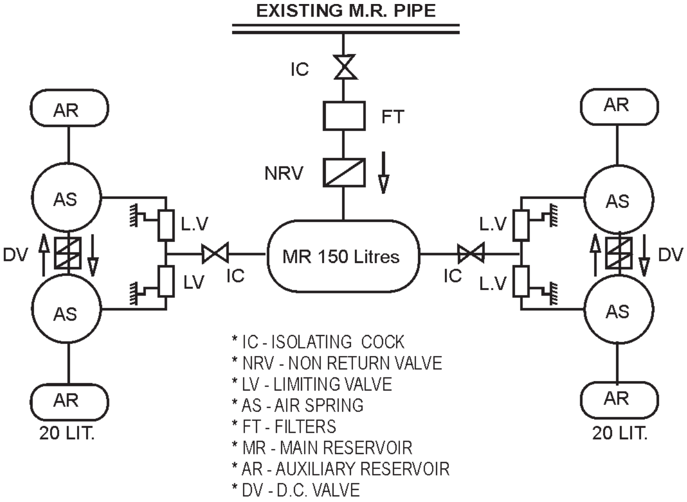


Fig. 3.33 Air spring system

their control mechanism, offer a load proportionate stiffness and, hence, prospect of better ride behavior with higher speed potential.

Over the years the number of passenger travelling in suburban area of metropolitan cities have increased manifold. Existing bogie suspension was not designed for such a heavy payload and adverse ratio between empty to full load condition, leading to maintenance problems. Air suspension at secondary stage has been taken up with optimized values of stiffness and damping characteristics to overcome the problem. It has been decided to build all AC EMU, AC-DC EMU and HHP-DMU coaches with air suspension in future. It has also been decided to provide air suspension on all stainless steel shell coaches (LHB type shell) provided with ICF bogies, for main line coaches.

RDSO have issued 'Maintenance Instructions on Air Suspension for DC, AC, AC-DC EMU/ HHP DMU Coaches', No. CMI – 9802 (Rev – 2), February, 2008. As per these instructions, air spring should be checked for any damage/ leakage and its piping should be checked for any leakage/ damage by soap test. The Instructions describe the procedure to be adopted for 'test for leakage'. Air is to be pumped into the air springs to a value of 9.0 kgf/cm2 by adjusting the horizontal lever of the leveling valve. All pipe joints are to be tested for leakages. The pressure is to be maintained for one hour. Drop in pressure should be within 1% of the test pressure.

Operating instructions have been issued by RDSO regarding deflated air spring (No. MC/CB/MM dt. 09.09.2009). As per these instructions, it should be ensured that all air spring are in inflated condition by noting that leveling valve lever are in horizontal position. If it is not possible to inflate the air springs of a particular coach, these should be isolated by air spring isolation cock and speed of train restricted to 60 kmph.

3.8 DAMPING

Damping is of two types

- Friction damping
- Hydraulic (viscous) damping

3.8.1 FRICTION DAMPING

As the name implies, the two parts oscillating relative to each other rub against each other at predetermined surfaces, causing friction, which provides damping. The commonest example of

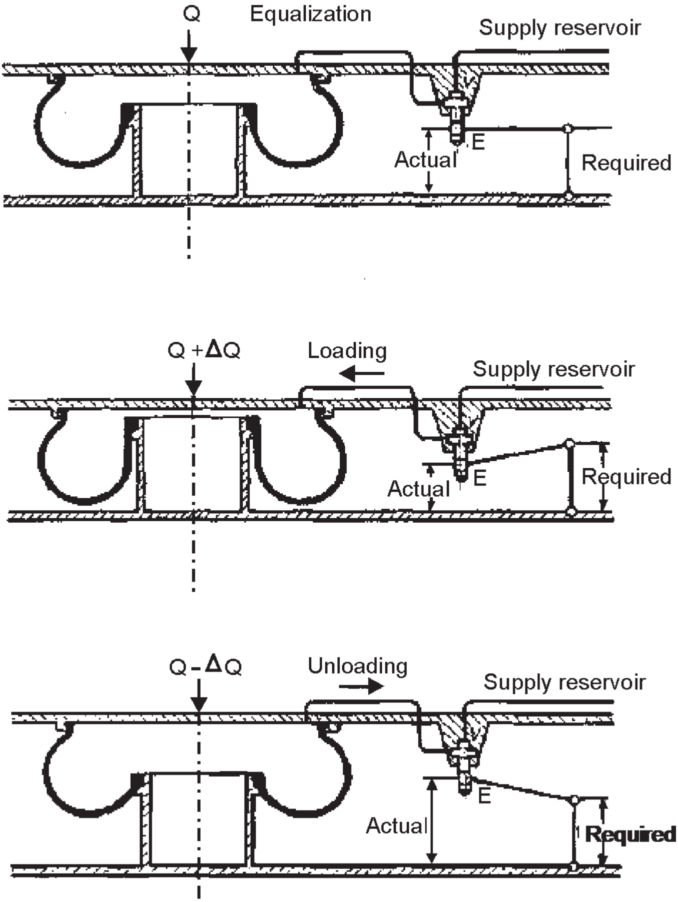


Fig. 3.34 Air Spring control mechanism

friction damping is the laminated spring itself. As a laminated spring deflects under load, relative sliding takes place at inter-surfaces of the various plates causing what is called inter leaf friction A laminated spring has, thus, built-in damping.

Another common example of friction damping is a spring loaded friction snubber, the principle of which is illustrated in **Fig.**3.35. The two rubbing components may be of any shape and the rubbing surface may be vertical or horizontal or inclined, providing damping in the corresponding direction. This type of damping is provided in CASNUB bogie and several locomotive bogies. These would be discussed in subsequent chapters.

An axle guard moving relatively up and down inside the axle box groove also provides some measure of friction damping.

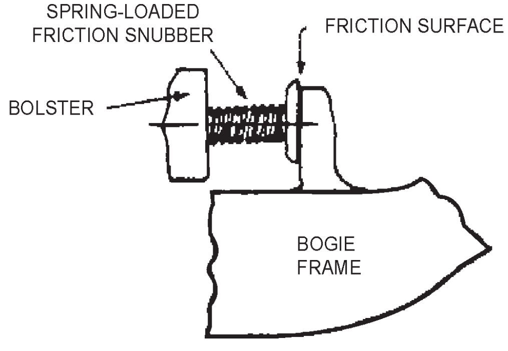


Fig. 3.35 Friction snubber arrangement

3.8.2 Hydraulic Damping

Telescopic cylindrical guides immersed in oil (commonly known as dashpot) is a common type of hydraulic damper. As the two parts of the dashpot move relative to each other, oil gets forced through small holes from one chamber to the other. Viscous resistance to this flow provides damping (**Fig.**3.36). This arrangement is discussed in detail in the chapter on coaches.

Shock absorbers (**Fig.** 3.37) are another type of hydraulic dampers, being provided in coach and locomotive of more recent origin. Although they work on the similar principle of viscous resistance to oscillation when a hydraulic oil is made to flow at a high speed, their design requires much lesser maintenance.

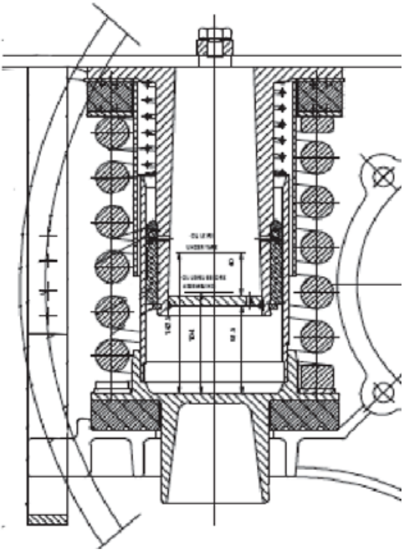


Fig. 3.36 Dashpot

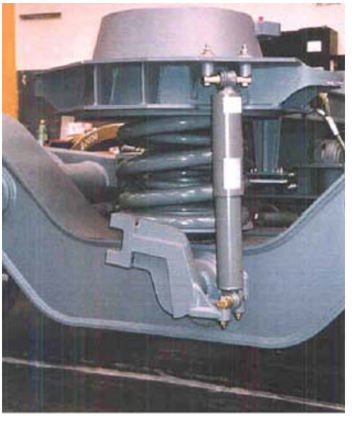


Fig. 3.37 Shock absorber

3.8.3 Defects in damping system

Any defect that tends to reduce the amount of damping in the system would contribute to derailment- proneness through adverse effect due to delayed damping of oscillations and possible resonance (see Chapter 1, Vehicle Oscillations.)

In a hydraulic damper, the level of oil has to be correctly maintained within the stipulated limits. For instance in the case of ICF coaches (ICF Laminated as well as ICF all-coil), the oil level in hydraulic dashpots is required to be checked every month, and oil replenished, if it is below 80 mm in case of B.G. and 71mm in case of M.G., ICF bogies. Over filling of oil in the dashpots shall be avoided as it renders the suspension stiff. It should not be less than 60 mm.

Hydraulic dashpots (on ICF trolley) broken, is a rejectable item (prefixed by 'S' in IRCA CR).

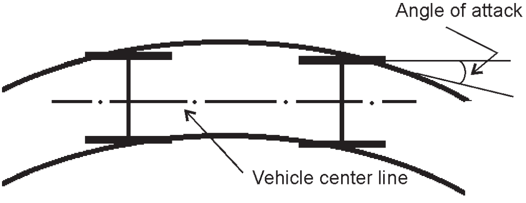
Defects in shock absorber may be indicated primarily by oil leakage or physical damage. As per RDSO specification C-8703 (Rev.1) for hydraulic shock absorbers for coaching stock, they should be given Schedule overhaul when their capacity varies beyond ±20% of their specified value, or after 4 lakh kilometers/ alternate POH, whichever is earlier. They should be given Non schedule overhaul whenever suspected to be defective.

A simple manual test may also be carried out to detect defects in shock absorbers, as included in Maintenance Instructions for some of the locomotives (eg. WDG4). This is based on the principle that the damping force generated is proportional to the velocity of oscillation (and in the opposite direction). In this test, a compressive force is applied manually, which would be quite small compared to the rating of shock absorbers, after disconnecting its one end, and rate of compression is observed. A significant rate of compression would indicate a defective shock absorber.

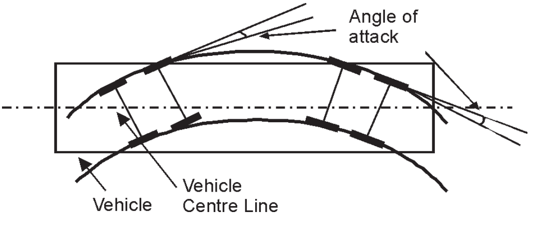
3.9 BOGIE ROTATION

3.9.1 Significance of bogie rotation

A four-wheeler cannot be made too long, as the longer rigid wheel-base will not be able to negotiate curves and turnouts without developing excessive angularity and excessively high flange forces.



(a) 4 – Wheeler stock



(b) Bogie stock

Fig. 3.38 Bogie Rotation (a) 4- Wheeler stock

(b) Bogie stock

Thus, when a longer vehicle body is needed to have a higher pay-load, it has to be supported on two bogies, each of relatively much shorter rigid wheel-base. The two bogies can undergo rotation in relation to the vehicle body as well as to each other while negotiating curves and turnouts and, thus, avoid excessive positive angularity and, hence, build up of flange forces. This is illustrated in **Fig.** 3.38.

The foregoing implies that if on a curve or turnout bogie rotation is hindered and not allowed to take place freely, it would have the effect of increasing the effective rigid wheel-base, thereby causing high positive angularity and flange forces to occur. Besides, a bogie having tendency of jamming during bogie rotation may get so jammed after some rotation on a curve or turnout that it may not return to the normal position when coming on to the straight. This will, thus, result in persistent angular running of the axles on the straight.

3.9.2 System of bogie rotation

Connection between the car body and bogie must:

* Allow the bogie to turn relative to the car body in curves
* Transmit the vertical, traction and braking forces
* Provide additional control of lateral suspension inputs
* Assist in maintaining the stability of the bogie
* Provide longitudinal stability of bogie frames and equal distribution of load over the wheelsets (for traction rolling stock)

These problems are solved differently depending on the type of the rolling stock - traction or trailing, passenger or freight, moderate or high speed.

Flat Centre Plate is the most common connection in three-piece freight bogies, secured by pin pivot at the centre. The plate transmits the majority of the car body weight and the longitudinal and lateral interaction forces. The pin pivot has large in-plane (horizontal) gaps to the car body and only provides emergency restraint. The centre plate allows the bogie to rotate in curves and generates a friction torque that imparts yaw stability.

Although simple in construction, there are several disadvantages. Clearances exist in lateral and longitudinal directions. Relative motion occurs under high contact pressure, resulting in significant wear on the surfaces. When the car body rocks on straight track, the contact surface becomes very small and high contact pressures may lead to cracks in the centre plate. In curves, the car body leans on the side bearer, creating additional friction torque that resists bogie rotation and increases wheel – rail forces. To combat these problems, modern designs use a flat centre plate combined with elastic side bearers which resist car body rock and reduce the load on the centre bowl.

In **Spherical Centre Bowl** arrangement, the car body rests on the spherical centre bowl and elastic side bearers (Figure 3.39). Advantage of this design is the lack of clearance in the horizontal plane and elimination of edge contact during car body roll.

This results in reduced levels of contact stress and increase in the centre bowl service life. A similar system is provided in CASNUB bogie on Indian Railways.

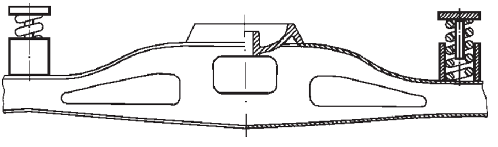


Fig. 3.39 Spherical Center Pivot

In order to exclude edge contact and increase the friction torque to resist bogie yaw, bogies with centre pivots were developed. Car body mass in this case is mainly transmitted to the side bearers and the car body can only turn relative to the bolster about the vertical axis.This design is widely used in passenger coaches (e.g. ICF All-Coil Bogie).

Connection of Car Body to Bolsterless Bogie is a modern design using either flexicoil spring or air spring. In such suspensions the springs can achieve large deflections in shear, providing sufficiently large longitudinal displacements to allow the bogie rotate in curves. In case of air spring, it is generally arranged in series with a rubber-metal spring to provide some minimum suspension in case air spring deflates. Transmission of longitudinal forces is generally done through the centre pivot and traction rod linkage. Bolsterless bogie designs achieve reduction in bogie mass.

Features of bogie rotation system used on rolling stock on the Indian Railways are discussed in more detail in subsequent chapters.

3.9.3 Defects

Hindrance to bogie rotation is an important defect to look for in the event of a bogie stock involved in derailment. This is particularly important in case of derailment in sharp curves.

Where to look for such defects? Bogie rotation system is constituted of center pivot and side bearers.

While a bogie rotates, it continues to support the vertical weight of vehicle body. Thus, relative movement takes place at surfaces where the weight of vehicle body is supported on the bogie. Such surfaces should be checked for any tendency towards jamming or undue hindrance against bogie rotation. Possible defects are :

- Uneven wear (**Fig.**3.40)
- Excessive wear
- Lack of lubrication (at such surfaces where lubrication is required e.g. at surfaces of steel-to-steel contact).
- Ingress of dirt, coal ash etc. and rusting

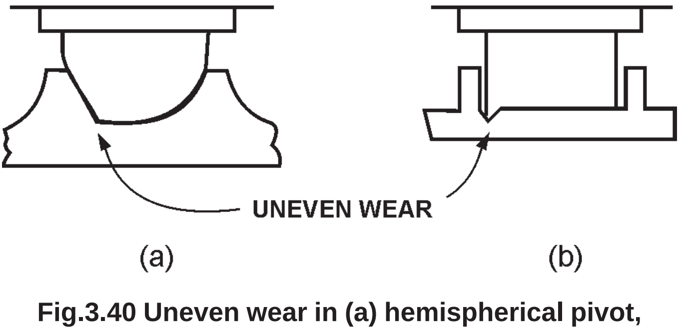


Fig.3.40 Uneven wear in (a) hemispherical pivot, (b) flat pivot

Particular locations required to be checked for some common types of rolling stock have been indicated in the respective chapters to follow.

3.10 BRAKE GEAR

The Air Brake goods stock on IR is at present fitted with single pipe graduated release air brake system. The diagram shown in **Fig.** 3.41 illustrates the schematic layout of air brake equipment on the under frame of freight stock.

During charging stage, brake pipe is charged to 5 kg/cm^2 pressure, which in turn charges control reservoir and auxiliary reservoir to 5 kg/cm^2 pressure via distributor valve. At this stage, brake cylinder gets vented to atmosphere through passage in Distributor valve.

For application of brakes, the pressure in brake pipe has to be dropped. This is done by venting air from driver's brake valve. Reduction in brake pipe pressure positions the distributor valve in such a way that the control reservoir gets disconnected from brake pipe and auxiliary reservoir gets connected to brake cylinder. This results in increase in air pressure in brake cylinder resulting in application of brakes. Magnitude of braking force is proportional to reduction in brake pipe pressure.

For releasing brakes, the brake pipe is again charged to 5 kg/cm^2 pressure by compressor through driver's brake valve. This action positions distributor valve in such a way that auxiliary reservoir gets isolated from brake cylinder and the latter is vented to atmosphere through distributor valve and, thus, brakes are released.

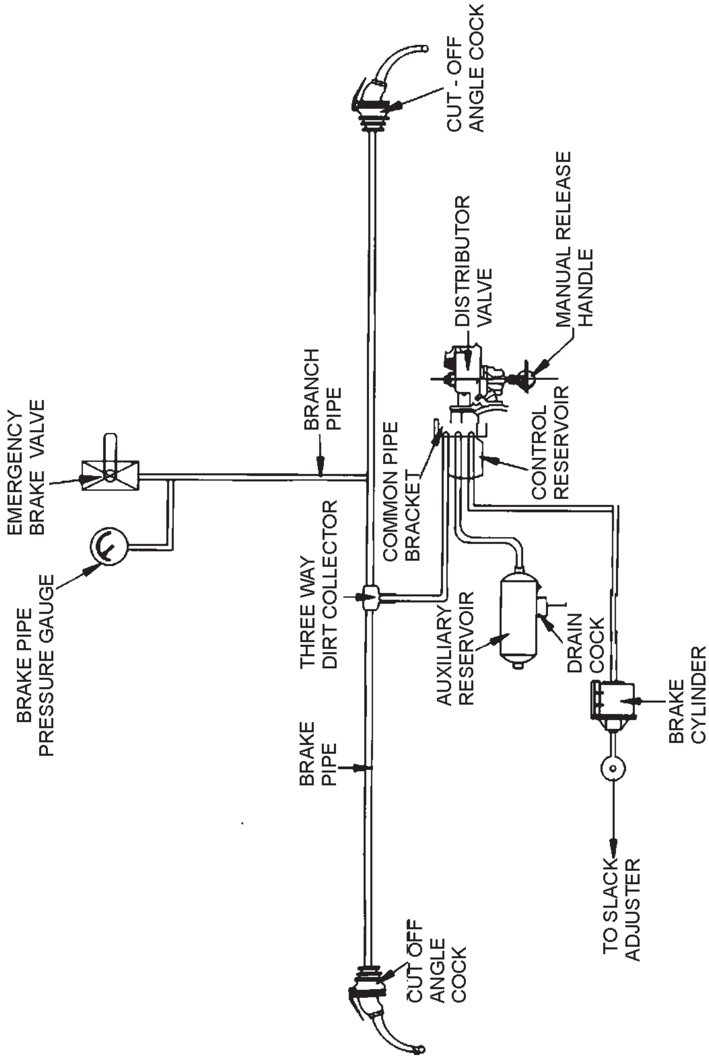


Fig. 3.41 Air Brake System (Single Pipe)

On every wagon fitted with air brake system, one brake cylinder (Figure 3.42) is provided for actuating the brake rigging for the application and release of brakes. During application stage, the brake cylinder receives pneumatic pressure from the auxiliary reservoir after being regulated by the distributor valve. Thereafter, the brake cylinder develops mechanical brake power by outward movement of its piston assembly. To transmit this power to the brake shoe, the push rod of piston assembly is connected to the brake shoe through a system of levers to amplify and transmit the brake power. During release action of brakes, the compression spring provided in the brake cylinder brings back the rigging to its original position.

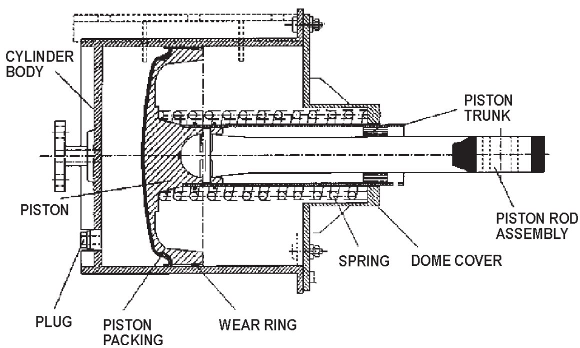


Fig. 3.42 Brake Cylinder

In Twin pipe Brake System provided on main line coaches, the locomotive compressor charges the feed pipe and the brake pipes throughout the length of the train. The feed pipe is connected to the auxiliary reservoir and the brake pipe is connected to the brake cylinder through the distributor valve.

Brake application takes place by dropping the pressure in the brake pipe.

Brake rigging is provided for transmitting the braking force from the brake cylinder to the wheel tread, with required magnification using levers (Figure 3.43).

Slack adjuster (also known as brake regulator) is a device provided in the brake rigging for automatic adjustment of clearance/slack between brake blocks and wheel. It is fitted into

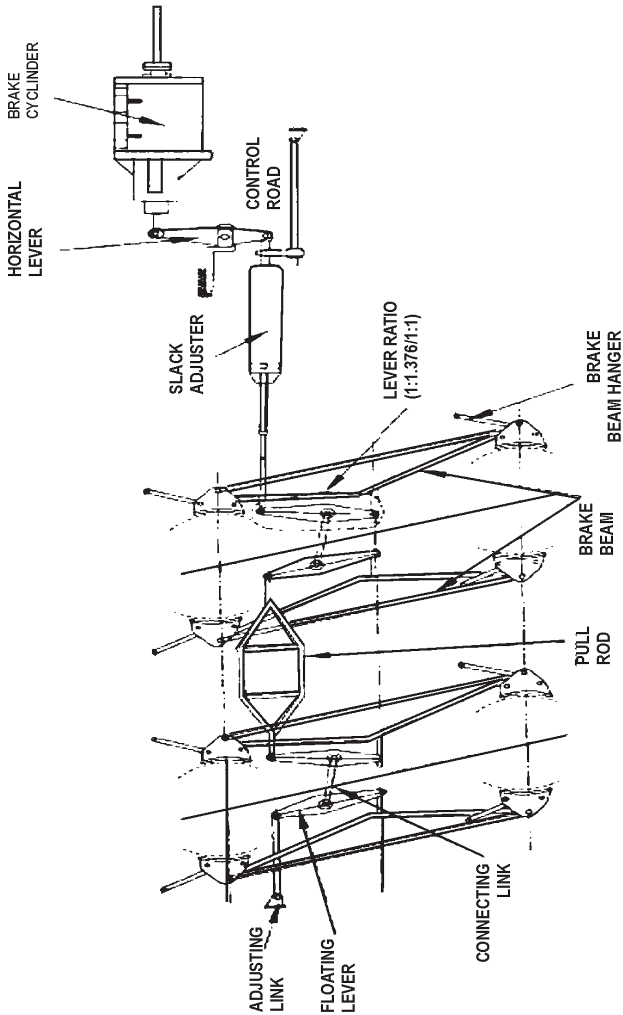


Fig. 3.43 Brake Rigging (Coach Under Frame)

the brake rigging as brake rigging as a part of mechanical pull rod. The slack adjuster is double acting and rapid working i.e. it quickly adjusts too large or too small clearance to a predetermined value.

On application of brakes, the brake pressure on the two wheels of an axle should be more or less equal, otherwise, the wheel which is braked less would tend to travel more, causing the axle to become angular. The axle would, thus, run persistently angular during the brake application. Defects which could cause the above condition to occur are:

- brake block deficient
- incorrect centralization and adjustment of brake rigging and brake blocks
- uneven application of brake power and wear in gear
- uneven wear of brake blocks on the same axle.

BRAKE GEAR - REJECTIONS (prefixed by S)

Goods and Coaching stock (B.G. & M.G.) :

§ Vacuum cylinder trunion or bracket broken or any bolt or rivet deficient, or welding failed

§ Brake block deficient, broken at the eye, not secured properly by correct size cotter/nut with split pin and/or key with split pin, or worn so thin so that the flange of the wheel is 6 mm or less from the brake beam collar on brake application

§ Any defect in brake rigging preventing application or release of brakes

§ Any brake rigging pin, split pin or cotter free to work out, deficient or broken

§ Safety hanger or bracket for brake beam or pull/push rod deficient, broken, of wrong size or not properly secured.

Additional for coaching stock :

§ Brake cylinder deficient or inoperative, subject to limits laid down in Rule 2.12.2.2 of the IRCA Conference Rules, Part IV.

2.12.2.2. – Trains with speed more than 105 kmph for BG (and 75 kmph for MG) would not be allowed to move from primary maintenance depot with less than 100 % brake cylinders operative. For other trains, this limit would be 90 %. However, no bogie vehicle would be allowed to leave primary maintenance depot with both cylinders defective.

§ Vacuum/ Auxiliary reservoir suspension bracket broken/ deficient or not secured properly.

3.11 TWIST IN UNDERFRAME

3.11.1 LONGITUDINAL TWIST

A longitudinal twist would cause the axles to remain persistently angular to the track, thus increasing the derailment-proneness. This can be detected by measuring the length of diagonals joining the four corners of the under frame (or bogie frame) . A trammel gauge is used for measuring the length. Locations between which the tramming is to be done are specified in the Maintenance Instructions of the rolling stock. These points are physically located on the under frame (or bogie frame). Tramming is normally done during POH. In case of investigation of derailment, need for tramming should be decided based on the physical condition, i.e. presence of cracks, damage, distorsion etc., of the under frame/ bogie frame.

3.11.2 VERTICAL TWIST

Vertical twist is detected by measuring the height of sole bar at the four corners of the underframe (or bogie frame) above the rail level, keeping the vehicle on a level track.

A twist in underframe is equivalent to a twist in the track and, thus, will increase derailment proneness of the vehicle. A vertical twist of the order of 20 mm can appreciably increase the derailment proneness.

3.11.3 UNDERFRAME DEFECTS - REJECTIONS (prefixed by S)

Goods and coaching stock (B.G. and M.G.)

- § Crack visible on both sides of the web or loose patch on any rolled or pressed section or built up girder forming part of the underframe. A weld unsupported by a patch on a web or a patched member which shows signs of crippling.
- § Headstock bent so that the centre of the buffer face is displaced in any direction more than 35 mm from its normal position in case of B.G. goods stock, or 38 mm in case of B.G. coaching stock.

  Note : No packing is permissible except on wagons booked for repairs.
- § Truss rod brackets deficient or fractured. Defects of bogie frame have been further discussed in subsequent chapters

3.12 BUFFERS AND DRAFT GEAR

There are two main arrangements of draft and buffing gear in use on Broad Gauge. The older arrangement, which is mainly found in coaches and on few wagons, consists of a screw coupling with side buffers (Figure 3.44). In this design the draft load is transmitted through the screw coupling, draw hook and draw hook springs while the buffing force is borne by the side buffers. The conventional screw coupling (WA/BD-125) has a working load of 22.5t.

Fig. 3.44 Screw coupling and side buffer

Restrictions of size and weight limit the extent to which the draft capacity of the above coupling can be improved. Recognizing this fact, the other arrangement on BG wagons, and on some coaches, is that of a Centre Buffer Coupler (CBC) which transmits both draft and buffing loads (Figure 3.45). The knuckle type centre buffer coupler has been adopted for BOX, BOXN and other new design of wagons. The working strength of CBC coupler is 120 tonnes. CBC also has a transition version called "Transition Coupler" which incorporates a screw coupling and a pair of side buffers to permit attachment with wagons fitted with screw coupling.

Actual coupling between the vehicles is called Draw Gear and the impact absorbing apparatus, through which the Draw Gear is attached to the vehicle, is called Draft Gear. Free Slack (or Loose Slack) is theclearance within the Draw Gear which may be Run- out or Run- in without any compression or extension of the Draft Gear. Spring slack is the additional movement which occurs when the Draft Gear is fully compressed. Its value is upto 5 inch.

Buffing gear plays a vital role in protecting the entire wagon against damages due to impacts. Buffing springs have to perform the basic function of absorbing buffing impacts received in service and to transmit these gradually to the under frame.

Hence, working capacity of these buffing springs should be adequate to meet these requirements. Buffing gears are of two types viz. "Long Case" and "Short Case".

Long case buffers are higher capacity buffers. In order to protect under frame, it was decided to replace short case buffers. Main components of the buffing gear sub-assembly are Plunger, Buffer casing, Spindle, Outer coil spring and Inner coil spring.

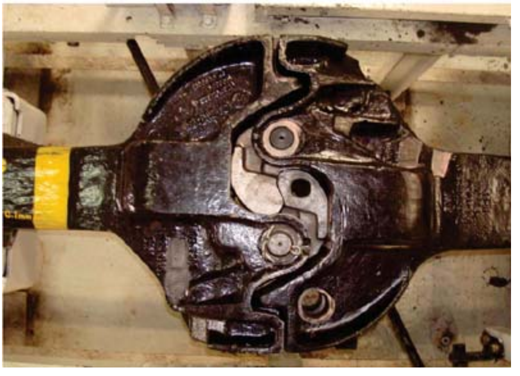


Fig. 3.45 Center Buffer Coupler

Draw gear arrangement has to be of robust construction and adequately sprung to minimize the impact loads owing to the starting jerks reaching the wagon underframe. The draw gear provides a continuous link between different vehicles comprising the train and failure of any of these can lead to train parting. Main components of BG conventional draw gear are Draw Hook, Draft link, Helical Springs and Screw coupling consisting of Shackle, Links and Screw .

Indian Railways use AAR type centre buffer couplers. The draft capacity of the AAR coupler depends on the strength of knuckle, which is the weakest link in the assembly. With CBC, coupling action between wagons is automatic . Shank/ Draw bar is cast together with the coupler head. Tail end of the draw bar is connected to draft gear through a central pin. Draft gear fits into the draft gear pocket of coach/ wagon and absorbs energy both in buff and draw modes, with a designed limit of stroke in each mode. Absorption of energy (impact) is achieved through the rubber pads provided in the draft gear. The maximum permissible free slack in the draft gear is 25 mm, beyond this it shall be removed and reclaimed or condemned.

One of the failures of the coupler system having safety implication is train parting. Prescribed precautions are to be taken at the time of coupling of CBC and it is to be ensured that the lock lift levers of both the couplers are completely dropped. Items to be observed in the event of train parting are locking status of knuckles of both the couplers, whether knuckles were found locked or open, position of lock lift lever of CBC in which knuckle was found open, whether it was fully or partially dropped, and whether manual uncoupling rod was locked.

Buffer defects have significant effect on derailment-proneness. The effect, basically, is due to eccentricity of buffing forces caused by such defects. The eccentricity in buffing forces can be in :

§ vertical direction, in which case Q values are affected

§ horizontal direction, when Y values are affected

§ inclined direction, when both Y and Q values are affected.

3.12.1 DEFECTS RESULTINGIN VERTICAL ECCENTRICITY OF BUFFING FORCES

i) **Difference in heights of buffers** of adjoining vehicles. This can happen at the junction of a set of empties and a set of loaded wagons (**Fig.**3.46)

Maximum and minimum permissible buffer heights in Broad Gauge and Metre Gauge stock (goods and coaching) are shown below:

BG	-	Maximum - 1105 mm (empty)
	-	Minimum - 1030 mm (loaded)
MG	-	Maximum - 585 mm (empty)
	-	Minimum - 535 mm (loaded)

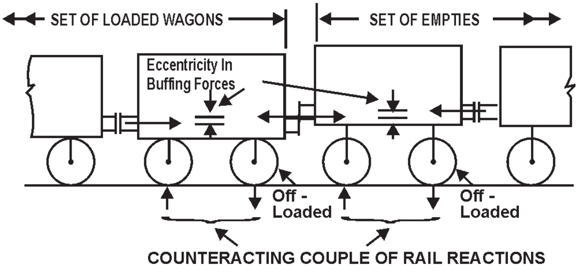


Fig.3.46 Effect of vertical eccentricity in buffing forces

Note : To make up the buffer heights to maximum permissible limits due to reduced diameter of wheels, a packing piece of required design and size may be interposed between axle box crown and spring

.

ii) Drooping buffer

iii) Buffer displaced vertically from its normal position,

due to headstock being bent. (on B.G. Maximum permitted displacement of buffer face due to head stock being bent is 35 mm in any direction from its normal position in case of wagons and 38 mm in case of coaching stock; see **Fig.** 3.47).

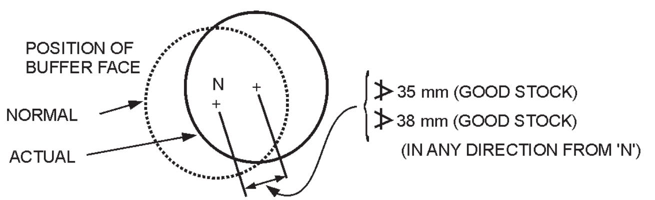


Fig.3.47 Maximum permitted displacement of buffer face from its normal position

3.12.2 DEFECTS RESULTING IN HORIZONTAL ECCENTRICITY OF BUFFING FORCES

i) Buffer deficient

ii) Dead buffer

A buffer shall be considered dead when it is ineffective or when its projection from the headstock (on B.G.) is below the prescribed minimum limits viz.

a) Goods stock:

For long case : 584 mm

(maximum limits is 635 mm)

b) Coaching stock : 584 mm

(maximum limit is 635 mm)

3.12.3 DEFECTS RESULTING IN ECCENTRICITY OF BUFFING FORCES IN BOTH DIRECTIONS

Buffer face is displaced from its normal position in an inclined direction, due to headstock being bent. On B.G. maximum permitted displacement of buffer face due to headstock being bent is 35 mm in any direction from its normal position in case of wagons and 38 mm in case of coaching stock (see **Fig.**3.47).

3.12.4 BUFFING GEAR DEFECTS

- Rejections (prefixed by S)

B.G. Goods stock :

§ Buffer deficient

§ More than one dead buffer on two consecutive wagons on a train

§ More than three rivets or countersunk bolts missing from a buffer plunger

§ Buffer spindle broken or nut deficient or of incorrect size.

B.G. coaching stock :

§ Buffer deficient or drooping

§ Any buffer dead

§ Buffer spindle broken or nut deficient.

M.G. goods and coaching stock:

§ Wagons fitted with a buffer of other than a screw coupling type

§ Buffer of screw coupling type or component parts deficient or so defective as to prevent tight couplings

§ One wing of buffer face cracked or missing if below level of the U in the buffer face

§ Buffer spring deficient, broken in more than two pieces or of incorrect size

§ Buffer coupling hook bolts, not secured according to relevant Rules of the IRCA Conference Rules

§ Wagons not fitted with buffers of a non-rotatable type or with some form of anti-turning device other than a linear

§ Buffer shank worn more than 13 mm through any section or less than 45 mm diameter over threads

§ Buffer shank with nut, check nut split pin or cotter deficient or of wrong size

Note : When split pin is used a check nut must be fitted. Buffer head cracked 19 mm or more from coupling pin hole.

Irregular loading (B.G. stock) :

§ Difference of height from rail level of more than 64 mm between any two buffers on the same vehicle measured at the head stock.

§ Flange of any wheel within 25 mm of the bottom of vehicle.

Now we will be taking up some common types of rolling stock and discussing their additional typical features and defects in subsequent chapters.

■ ■ ■

CHAPTER 4

FREIGHT STOCK

There are various types of wagons on the Indian Railways system. Their nomenclature is attached as Annexure – 1 of this chapter

4.1 ORDINARY FOUR-WHEELER GOODS WAGON

A typical suspension arrangement of above type of wagon is shown in the photograph in **Fig.** 4.1. It is a single stage suspension and is also called shackle suspension. This stock has only laminated springs, which provide in-built damping through inter-leaf friction. The springs rest on plain bearing axle boxes. The wheelsets are held in the wagon underframe through conventional axle guard horn cheek - axle box groove guidance. The wagon underframe comprises headstocks and sole bars. The buffers are connected to the headstocks. Chapter 3, **Fig.** 3.28 and 3.29 may be referred to.

The axle guards and scroll irons are connected to the sole bars. The rolling stock details and defects discussed in the preceding Chapter (except the portion relating to bogie stock e.g. bogie rotation) fully cover the above four-wheeler wagon.

Fig. 4.1 shows, at a glance, the various items and features to be checked during derailment investigation in the event of such stock being involved in the derailment.

N.B. Items and features to be checked which are more or less common in principle, to all types of rolling stock, e.g. wheel profile defects, wheel diameter, wheel gauge, clearances at axle box level, defects in axle guard and axle box assemblies (plain bearing or roller bearing), defects in spring gear, buffing gear and brake gear, defects in underframe and trolley frames, etc. have already been covered in Chapter 3,. Therefore, only additional defects particular to the relevant stock have been discussed in this Chapter.

Figure 4.1 General Suspension arrangement of 4-wheeler wagon

Particular items to be examined in derailment :

- *Axle guard, Horn cheek and Briddle bar*
- *Horn cheek – Axle box groove Guidance and Clearances*
- *Laminated bearing spring – General assembly of spring gear, Free and Working camber, scrag test*
- *Scroll iron, Shackle plate and Shackle pin - Clearances at Shackle pin whether shackle plate touching bottom of sole bar*

4.1.1 CR WAGON, MOD. I AND MOD.II (RDSOS REPORT NO. M264).

The conventional four wheeler covered wagon CR has an axle load of 16.26 t and a maximum permissible speed of 75 km/h. With a view to enhance the pay load of the covered wagon, two different modifications to the existing CR wagon were carried out. Modification-I mainly consists in the provision of 20.32 t roller bearing axles and plate type axle guards and Modification II, in addition to 20.32 t roller bearing axles, a single link suspension for lateral flexibility and auxiliary coil springs, in series with laminated spring, for improved vertical suspension.

Under empty conditions, there is a gap of 4 mm between auxiliary spring stops. When the wagon is loaded beyond 1.34 t, this gap closes on account of flexing of helical springs and the auxiliary springs get out of the load circuit and the loading is entirely through the laminated springs. Thus, adequate flexibility in empty condition and safe loading through laminated springs in loaded condition are provided. The features of these suspensions are shown in **Fig.** 4.2.

Table 4.1 shows the principal features of CR Conventional, Mod.I & Mod.II wagons.

4.2 SPECIAL BOGIE FREIGHT STOCK

There are four types of IRS bogie trucks :

- Four wheeled cast steel bogie truck : There are two cast steel side frames joined together by a spring plank, which is rivetted to them. A floating bolster rests on nests of spring on either end of the spring plank. Wagon load is transferred to the bolster through center pivot and then to two side frames through bearing springs. These bogies have only secondary spring.
- Four wheeled diamond frame bogie : It is similar to cast steel bogie but bolster and side frames are fabricated, instead of cast. Side frames are comprised of top and bottom arch bars, connected by a tie bar.

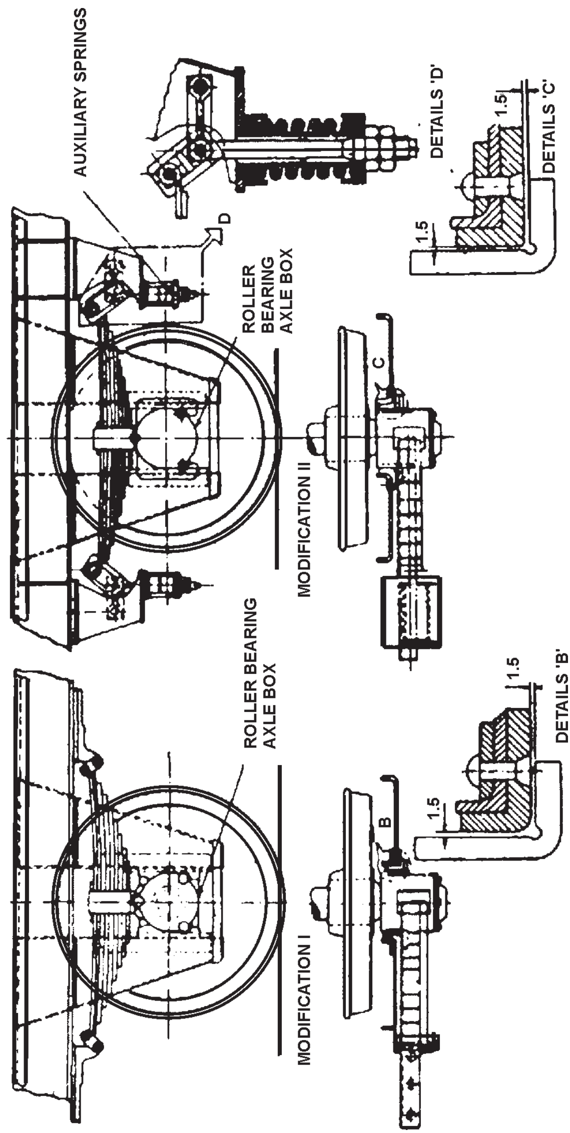


Fig. 4.2 CR Wagon Mod I and Mod II

Table 4.1
Principal features of 'CR' wagons

Particulars	Conventional	Modification I	Modification II
Length over buffer	8433mm	8433mm	8433mm
Length over headstock	7163mm	7163mm	7163mm
Wheel base	4572mm	4572mm	4572mm
Wheel diameter on tread	1092mm	1000mm	1000mm
Nominal axle load	16.26t	20.32t	20.32t
Lateral clearance (designed values)			
(a) between axle box horn guide	3.2mm	1.5mm	1.5mm
(b) between axle box and brass journal or roller bearing	0.8mm	Nil	Nil
Longitudinal clearance (designed values)			
Between axle box and horn guide	1.5mm	1.5mm	1.5mm

- Four wheeled fabricated UIC type (BOX) : This bogie has a completely fabricated rigid structure with laminated bearing spring suspension and long shackles at the primary stage i.e. directly over the axle boxes.
- Four wheeled cast steel bogie with long travel spring and snubbing device : There are two types of these bogies, one with spring plank connecting two side frames (CASNUB bogie) and the other, plankless bogie (AMSTED bogie).

N.B. : It may be mentioned that the letters e.g. BOX, BOBS, BOY, BOI etc. designate the type of wagon body, whereas the bogies have distinctive names. Only in case of BOX wagon, the bogie also is typified by the letters BOX, though such bogies can also be used under some different types of wagon bodies.

All special bogie freight stock have the following features in common :

- they have a high play load-to-tare ratio
- axle box is of roller bearing type
- suspension is primarily single stage
- body weight is transferred to the bolster primarily through the center pivot, the two side bearers providing the requisite support during vehicle motion.

Hence, in all these stocks the surface of the center pivot and side bearer should be checked for any tendency towards jamming or hindrance against bogie rotation.

4.2.1 BOX wagon (RDSO Report Nos. M-186 & M-192)

This high sided bogie open wagon has bogies based on 'UIC long link' design.

The bogie comprises an all-welded light weight plate construction bogie frame which is suspended by long links or shackles (332 mm long) from the ends of laminated bearing springs resting on roller bearing axle boxes. The axle boxes have conventional horn cheek-axle box guidance.

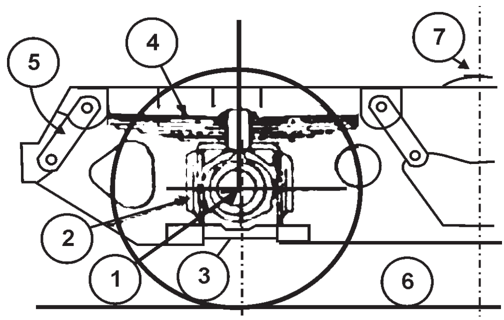
The general arrangement of bogie suspension is shown in **Fig.** 4.3.

Bolster is fixed to the bogie frame. The hemispherical center pivot is designed to take the superstructure weight with 4 mm clearance between the side bearers and the body. The maximum clearance at the side bearers arising out of wear during service should not exceed 14 mm. The laminated springs which are of ribbed and grooved type are provided with a buckle having a spigot at the bottom which engages in a bushed seat at the crown of the axle box.

Free camber of spring = 47 mm (+6, -0) mm

A maximum negative camber of 4 mm is permitted for the springs. The links or shackles should articulate freely. Clearance between the shackle pin and bush hole when worn should not exceed 3 mm.

Play between axle box lug and horn cheek when new is 20 mm. Maximum permitted in service is 25 mm.



1.Roller bearing axle box 2. Horn Cheek 3.Briddle bar 4. Laminated spring 5.Shackle (Link) 6. Bogie frame 7. Centre pivot

Fig. 4.3 Suspension arrangement of BOX bogie

Items to be checked during derailment investigation:

General – Defects in wheelset, axle box assembly, spring gear,buffing gear, brake gear, clearances and play at axle box

Particular – Surfaces of centre pivot/ side bearer for hindrance to rotation, Negative working camber of spring (max 4 mm), Free swivel of shackles, Brake gear set to empty or loaded

BOX wagon brake gear is fitted with slack adjuster. The brake rigging can be set to empty or loaded condition.

This feature should be particularly checked to have an idea comparatively of the braking distance or the possibility of hard braking (e.g. when brake is set to loaded, while wagon is empty) which could cause the wheel to jump.

Other particulars are

Bogie wheel base	2000 mm
Wheel diameter	1000 mm
Axle load	20.32 t

The wagon is provided with AAR E-type 'Centre Buffer Coupler.

This bogie has been cleared for a maximum operational speed of 75 km/h, having been tested and found stable upto 104 km/h.

The defects to be checked for are by and large similar in principle to what have been discussed in the preceding Chapter.

4.2.2 DIAMOND FRAME BOGIES

There are three common types of bogies in this category:

- Talbot bogie (used under BOBs wagon body)
  (RDSOs Report No. M.208 and M. 209)
- RDSO Soft Ride bogie (RDSOs Report No.M.261)

The general arrangement of suspensions of Talbot bogie and RDSOs soft ride bogies are shown in Figs. 4.4 and 4.5.

In a diamond frame bogie, the axles are gripped by two longitudinal cast steel (e.g. Talbot and Casnub bogies) or fabricated mild steel (e.g. RDSO Soft Ride Bogie) side frames connected together by a spring plank, on either end of which rests a cluster of helical spring, which in turn support the bolster. There are no springs at the axle box level. The bolster is guided through bolster lugs sliding along the vertical members of the side frames. This provides for location and guidance of axles in the bogie suspension.

Except Talbot bogie, other bogies have spring loaded friction snubbers (4 per bogie) which provide damping, both in lateral and vertical modes, in the suspension system. The damping force is proportional to weight on the bogie pivot.

4.2.2.1 Defects

Items and features to be checked during derailment investigation for defects in Diamond frame bogies in general:

General

* Defects in wheelsets, axle box assembly, bolster lug-side frame member guide surfaces, spring gear, buffing gear and brake gear
* Clearances at the bolster lugs slide frame member guide.

Particular

* Surface of the centre pivot for any hindrance to bogie rotation (also check the side bearers in this context)
* Spring loaded briction sunbbers (except in Talbot bogie)

Table 4.2 gives the salient particulars of Talbot and

Soft Ride bogies.

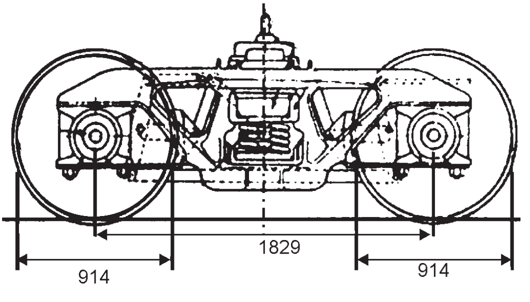


Fig. 4.4 Suspension arrangement of Talbot bogie

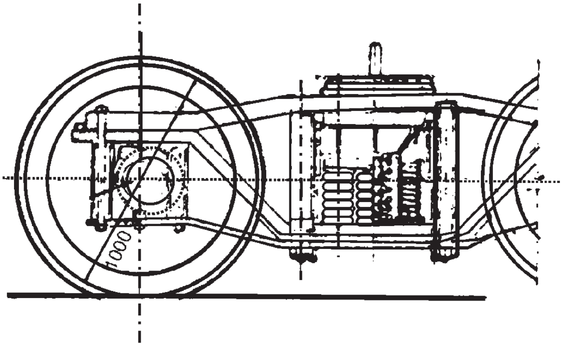


Fig. 4.5 Suspension arrangement RDSO Soft Ride bogie

Table 4.2
Salient particulars of Diamond frame bogies.

Description	Talbot	RDSO Soft Ride Bogie
1. Axle load (t)	22.86	22.86
2. Bogie rigid wheel base	1829	2000
3. Wheel diameter (mm)	914	1000
4. Type of axle bearing	SKF roller bearing	AAR standard cartridge type roller bearing
5. Bolster springs (No. per bogie)	10 nests (5 on each side)	14 nests (7 on each side)
6. Clearance between bolster and side frame		
(a) Lateral (mm)	6(total)	± 6
(b) Longitudinal (mm)	5 (total)	± 1
7. Damping	Nil	4 spring loaded friction snubbers per bogie

8.	speed upto which tested and found stable (km/h)	64 (empty) 48 (loaded)	110
9.	speed cleared for (km/h)	56 (empty) 43 (loaded)	75

4.3 CASNUB BOGIE

4.3.1 GENERAL DESCRIPTION

This bogie was first fitted in BOXN wagons and was designated as CASNUB 22W. This was later modified as CASNUB 22W(M) to take care of high wheel wear reported on earlier version. Subsequently CASNUB 22NL (Narrow jaw) and CASNUB 22 NLB (Narrow jaw with fish belly bolster) versions were introduced. The CASNUB 22 HS bogie has been developed for high- speed operation with maximum permitted speed up to 100 km/h. All CASNUB 22W bogies are to be converted to CASNUB 22W (Retrofitted) by the maintenance depots and workshops. The various bogie versions developed are as under

- CASNUB -22W
- CASNUB -22W (Retrofitted)
- CASNUB -22W(M)
- CASNUB -22NL
- CASNUB -22NLB
- CASNUB -22HS

These bogies are used in the following wagons:-

BOXN, BCN, BCNA, BRN, BTPN, BTPGLN, BOBR, BOBRN, BOBY, BOBYN, BFK

4.3.2 CONSTRUCTION DETAILS

The bogie comprises of two cast steel frames and a floating bolster. The bolster is supported on the side frame through two nests of springs. This also provides a friction damping proportional to load. A fabricated mild steel spring plank connects the side frames. A photograph of CASNUB bogie is shown in **Fig.** 4.6. **Fig.** 4.7 shows the bogie general arrangement.

The salient features of CASNUB bogie are given below.

No.	Features	Description
1	Gauge	1676mm
2	Axle load	20.3t, However all bogies except CASNUB 22 HS can be upgraded upto 22.9t
3	Wheel diameter	1000mm (New) 956mm (new) for Retrofitted CASNUB 22W
4	Wheel base	2000mm
5	Type of Axle bearing	CASNUB 22W & 22W(M) (i) Cylindrical Roller Bearing Axle Box in a limited number of CASNUB 22W Bogies only. (ii) Standard AAR Tapered Cartridge Bearing Class E suitable for 144.5 x 277.8 mm wide jaws. CASNUB 22NL, 22NLB & 22 HS standard AAR tapered Cartridge Bearing Class E suitable for 144.5x277.8 mm narrow jaw
6.	Distance between journal centres	2260mm
7.	Distance between side bearers	1474mm
8.	Type of side bearers	CASNUB 22W Roller type (clearance type) Retrofitted CASNUB 22W, CASNUB

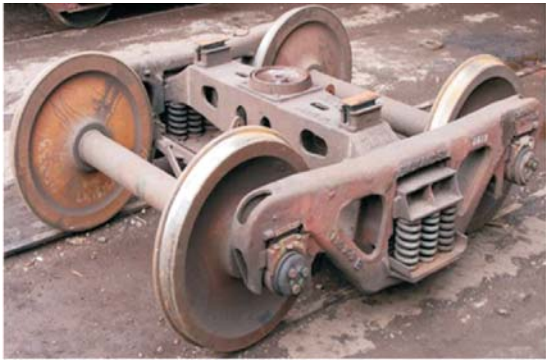
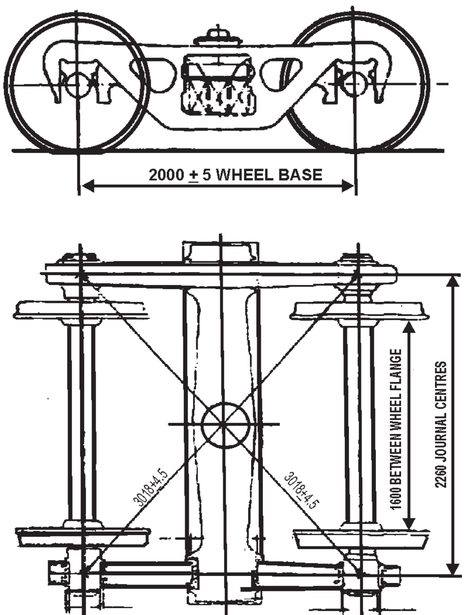


Fig. 4.6 CASNUB



CASNUB - 22NL, NLB, HS

Figure 4.7 CASNUB Bogie – General Arrangement

		22W(M), 22NL,22NLB Constant contact type (Metal bonded rubber pad, housed inside side bearer housing) CASNUB 22HS Spring loadedRoller type (clearanc constant contact type side bearer
9.	Type of Pivot	CASNUB 22W IRS type TOP Pivot – RDSO Drg.No.W/BE-601 Bottom Pivot – RDSO Drg.No.W/BE-602 or similar
10.	Anti Rotation features	Anti rotation lugs have been provided between bogie bolster and side frame.
11.	Type of brake beam	CASNUB 22W,22NL,22NLB, 22HS: Unit type fabricated brake beam supported and guided in the brake beam pockets CASNUB 22W(M): Unit type Cast Steel brake beam suspended by hangers from side frame brackets.
12	Suspension details	Long travel helical spring.
13	Elastomeric pads	On all types of bogies except CASNUB 22W.

The CASNUB bogie assembly consists of the following components:

i. Wheel set with Cartridge Bearing

ii. Axle Box/ adapter, retainer bolt & side frame key assembly

iii. Side frames with friction plates and brake wear plates

iv. Bolster with wear liners

v. Spring plank, fit bolts & rivets

vi. Load bearing springs and snubber springs

vi . Friction shoe wedges

viii. Centre pivot arrangement comprising of Centre pivot top, Centre pivot Bottom, Centre pivot pin, Centre pivot retainer & locking arrangement

ix. Side Bearers

x. Elastomeric Pad

xi. Bogie Brake Gear

xii. Brake Beam

4.3.2.1 Wheel set with cartridge bearing

The initial batch of CASNUB bogie was fitted with cylindrical roller bearing axle box and matching wheel set. However standard AAR taper cartridge bearings have been subsequently standardized for these bogies. Maintenance requirement of cartridge taper roller bearing have been issued under Instruction for inspection and maintenance of Cartridge Taper Roller Bearing fitted on Cast Steel Bogies, Technical Pamphlet No. G-81 by RDSO.

Wheel diameter for new wheel is 1000 mm. However, for CASNUB 22W (retrofitted), maximum permissible wheel diameter is 956 mm. Condemning wheel dia is 906 mm for all versions but with suitable packing.

4.3.2.2 Axle

Axles have to be subjected to ultrasonic testing during ROH/POH or whenever the wagons are sent to the shops. Wheel sets whose axles have under gone ultrasonic testing shall be stamped on the hub fillet.

Whenever axles are renewed, the workshop shall punch the following particulars in 5 mm letters on the axle end :-

i. Serial No.

ii. Workshop code where pressing has been done

iii. Date of pressing

iv. Journal centre

v. Pressing on pressure in tonnes (Both ends)

4.3.2.3 Axle box adapter, retainer bolt and side frame key assembly

CASNUB 22W

Initial lot of CASNUB 22W type bogies were provided with cylindrical roller bearing axle box on the wheel sets. However, cartridge taper roller bearing was soon standardized having adapter & adapter retainer bolt. The CASNUB 22W bogies are provided with wide jaw adapter as per RDSO sketch No. Sk-78527 but without elastomeric pads with wheel sets to Drg. No. WA/WL-4902, Sk-68512 and WD-89025-S/1 with retainer bolts to Drg No. SK-69594.

CASNUB 22W(M)

Wheel sets are with wide jaw adapter, cartridge roller bearing and adapter retainer bolt (WA/WL-4902/WD-89025-S/1 for wheel sets).

CASNUB 22NL,22NLB & 22 HS bogies

Wheel sets are provided with narrow jaw adapter, cartridge roller bearing (WD-89025-S/1 for wheel sets).

CASNUB 22W (Retrofitted)

Bogies are provided with modified wide jaw adapters but these are not interchangeable with CASNUB 22W and CASNUB 22W(M).

The wear limits are given in Table 4.3.

4.3.2.4 Adapter

When a bogie is dismantled, adapter is to be thoroughly inspected for soundness and wear. Wear at Thrust shoulder, Adapter bore (bearing seat), Adaptor crown lugs, Adapter crown seat, Adapter side lugs and Adapter sides is to be checked by corresponding gauges. Wear limits are given in Table 4.3.

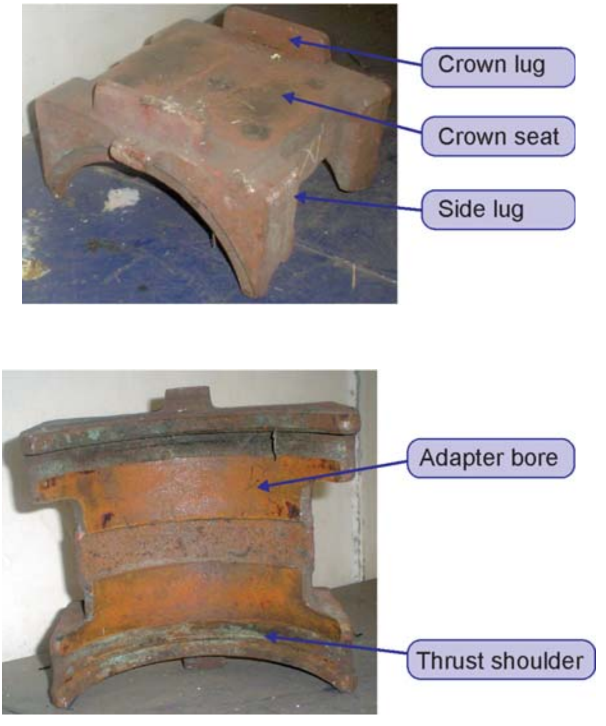


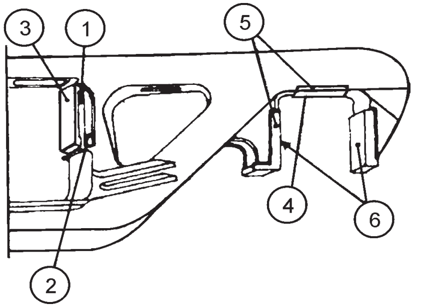
Fig. 4.8 Adapter

4.3.2.5 Side frames with friction wear plates

Wearing surfaces of side frame are shown in **Fig.** 4.9.
The wear limits are given in Table 4.3.

Side frame column has been provided with 10 mm thickness Silico Manganese Steel wear liners to IS: 3885 Pt.-I Gr. IV welded on the columns. It must be ensured that the liners permitted in service up to a thickness of 6 mm only.

Amount of wear on the outside and inside of the Side frame Column Sides is determined by placing the gauge centrally and inserting shims of appropriate thickness.



1. Side frame column sides 2. Anti-rotation lugs 3. Side frame friction liners 4. Pedestal crown roof 5. Pedestal crown side and pedestal sides 6. Pedestal jaw

Fig. 4.9 Locations of wear on side frame

Amount of wear on the Side Frame Anti-rotation Lug is determined by placing the gauge centrally and inserting shims of appropriate thickness.

In a similar manner, wear of Pedestal Crown Roof, Pedestal Crown Side and Sides of Pedestal and Pedestal Jaw is determined by use of appropriate gauges and shims.

The side frame should be checked for its wheelbase (distance between centre lines of the jaw openings) and it should be ensured that the correct button marking is left on the side frame. While pairing the side frame for a bogie, it should be ensured that there should not be any difference between the number of buttons on the two-side frames.

4.3.2.6 Bolster with wear liners

The wearing surfaces of the bolstar are shown in **Fig.** 4.10. (a and b) The wearing surfaces are Bolster pocket, Bolster Land Surface and Rotation Stop Lug and Bolster Column Gibs.

The wear limits are given in Table 4.3.

Bolster pocket has been provided with 8 mm thick Silico Manganese Steel Liners welded with pocket slope. The liners may be permitted in service upto a thickness of 3 mm.

No paint or grease should be applied on the plate.

Some bogie bolsters such as those of CASNUB 22NLB and 22HS bogies have been provided with 5 mm thick wear liners on land surfaces and the same are required to be replaced after 3 mm wear.

4.3.2.7 Spring plank, fit bolts & rivets

Spring plank is a member made of solid steel (flanging quality). It joins two side frames of CASNUB bogie by eight 24 dia rivets and four M24 fit bolts to keep bogie frame square.

Spring plank should be examined for defects like loosening of rivets/cracks/bending, welding failure of spring spigot etc. Whenever spring plank is renewed, the leading dimension of the bogie as per Drg no. SK-69599(W), WD-85054-S/6(22WM), WD-90042-S/1(NLB), WD-92058-S/7(HS) must be measured.

4.3.2.8 Load bearing springs and Snubber springs

The bogies are fitted with two groups of long travel helical spring nests. The spring details are shown in WD-83069-S/1 (Common for all versions except CASNUB- 22HS Bogie). The spring details of CASNUB 22HS are shown in WD-92058-S/5. **Fig. 4.11** shows the group of springs provided at secondary suspension.

Spring is condemned on the basis of Free height.

Nominal and condemning Free Heights are as below

BogieType		Free Height (nominal) (mm)	Recommended free height (condemning) (mm)
All version except CASNUB 22 HS			
	Outer	260	245
	Inner	262	247
	Snubber	294	279
CASNUB 22 HS			
	Outer	260	245
	Inner	243	228
	Snubber	293	278

It is recommended that springs having less than 3 mm free height variation should be assembled in the same group. Mixing of new and old spring must be avoided.

Excessive variation in the height of springs within the same group would result in unequal load on springs, resulting in Off-loading, adversally affecting safety.

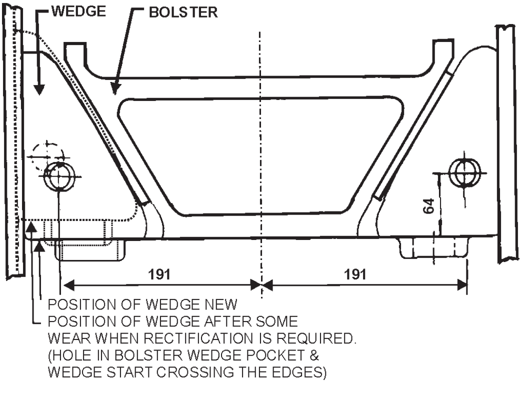
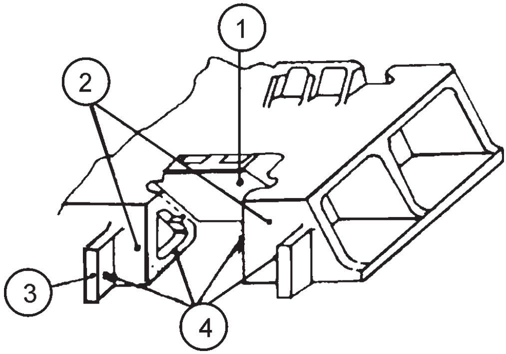
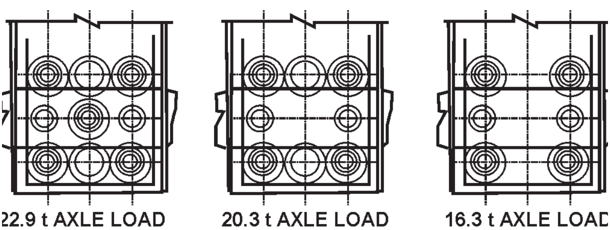


Fig. 4.10 (a) Friction Snubber Wedge Arrangement



1. Bolster pocket slope surface liner 2. Bolster land surfaces 3. Rotation stop lugs/liner 4. Bolster columns gib (inner and outer)

Fig. 4.10 (b) Locations to be reclaimed



	Number of Springs		
Axle Load	Outer	Inner	Snubber
22.9 t	14	10	4
20.3t	12	8	4
20.3 t (22HS)	14	12	4
16.3 t	8	8	4

Fig. 4.11 Details of Spring Group Arrangement

4.3.2.9 Damping System

The suspension is provided with load proportional friction damping arrangement with the help of manganese steel cast wedge supported on the snubber springs. Friction shoe wedges are fitted on snubber springs. The arrangement is shown in Figure 4.10 (a).

Vertical surface of the wedge is in contact with side frame and slope surface is incontact with bolster pocket liners. The dimensions are such that the snubber spring is under compression and, in turn, it presses the wedge upwards. This results in normal forces at both vertical and sloping surfaces. Relative movement due to oscillations generates friction at these surfaces, which causes damping.

Increased wagon load results in increased compression of bearing springs. This increases compression of snubber spring as well, resulting in increased magnitude of frictional forces and higher damping.

Wear limits on vertical surface and slope surface and nominal and recommended condemning dimensions are given in Table 4.3. Extent of wear and need of reclamation can be assessed in the bogie in the assembled condition by observing the relative position of the hole in the bolster wedge pocket and the opening in the wedge (**Fig.** 4.10 (a)).

Investigation of proper working of the Damping System would be important in case of derailments, particularly at high speeds. Defects such as a broken Snubber Spring and excessive wear should be observed.

4.3.2.10 Centre Pivot and Side Bearer arrangement

Centre pivot arrangement for CASNUB 22W bogie is as per RDSO Drg No. W/BE-601 for top centre pivot and W/BE-602 for bottom centre pivot for separate cast bottom pivot. For CASNUB bogies other than CASNUB 22W, centre pivot bottom and centre pivot top are as per RDSO Drg No.WD-85079-S/2.

Centre pivot pin for CASNUB 22W bogie is a headless pin while for other versions, a special type of pin is provided with castle nut/shackle lock for locking.

To determine the seat wear, the gauge should be placed in position **(Figure 4.12**). If the pivot surface (vertical wall top) starts touching the surface on the gauge at any point, repair is to be done by welding. The gauge should be moved on the complete worn surface to be measured. The surface after reclamation shall be the original dimension as per the respective drawings for proper matching of surfaces with top centre pivot. The repaired seat can be checked by applying two 4 mm thick shims between the casting and the gauge surface marked as *, as seen in the Figure.

Wear of the vertical wall is checked by measuring the gap between the wall and the gauge in position. Repairs should be carried out if a 9 mm thick shim in CASNUB 22W bogie (7 mm thick for other bogies) can be inserted for the full depth between the worn surface and the gauge at any point on the vertical wall of the bowl.

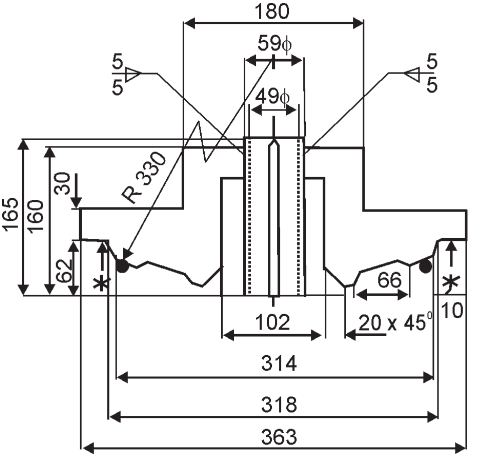


Fig. 4.12 Centre Pivot Gauge

Side bearer

CASUNB 22W Bogies are fitted with roller type side bearers, which are free to move in cast steel housing, rivetted on the bogie bolster. CASNUB 22W(Retrofitted), CASNUB 22W(M), 22NL, 22NLB Bogies are fitted with constant contact type of side bearer rubber pads located in cast steel housing which is riveted to the bogie bolster. CASNUB 22HS Bogies are fitted with helical spring loaded constant contact type side bearer, riveted/bolted on the bogie bolster.

Condemning of Side Bearers is decided based on its physical condition. Criteria for condemning the Rubber Pad is as discussed in para below for Elastomeric Pad.

In case of investigation of derailment, magnitude of wear, pattern of wear and the condition of surfaces at the Center Pivot should be critically analysed to assess if they cause any undue resistance to rotation of bogie with respect to the wagon. A lubrication of Graphte powder is provided for limiting the frictional resistance between the steel surfaces at the Center Pivot. Lack of lubrication and ingress of dirt/ dust etc. may cause a large frictional resistance to rotation, as the normal forces involved are high. Similarly, a large wear on Center Pivot seat would increase the normal force on the Side Bearer surface (which are not provided with any lubrication), increasing resitance to sliding, causing obstruction to bogie rotation. Uneven wear should also be analysed for similar effects. ***Hindrance to bogie rotation is potentially a major cause of derailment in sharp curves, as it would result in a large angularity.***

4.3.2.11 Elastomeric pad

Elastomeric pads are provided in all versions of CASNUB bogie except CASNUB 22W. The main purpose of providing elastomeric pad is to reduce wheel flange wear.

Elastomeric pads to 95005-S/4, Wd-92058-S/8 (for HS) & WD-95005-S-1 and side bearer rubber pads to WD-85076-S/1 shall be condemned and replaced by new ones on the following grounds :-

i. If the top of the bottom plates or intermediate plate in case of side bearer pads show any crack in service.

ii. If any crack of more than 50 mm is developed at any surface of rubber.

iii. If a bond failure giving way more than 40 mm in any direction is developed in service.

iv. If any sign of crushing of rubber is noticed.

v. When in free condition, the pad has taken a permanent set of the order given in Table 4.3.

4.3.2.12 Bogie brake gear

The brake gear mainly consists of Brake Beam (with brake head and brake block assembly), equalizing levers, Push rod, End pull rod, Brake Beam hangers (in CASNUB 22W(M) bogies). The bushes provided are case hardened or through hardened and pins are made from steel.

The maximum permissible wear on the pin diameter and bush inside diameter is limited to 1.5 mm.

In service as the tread diameter of wheel decreases due to wear, pins located in End Pull Rod with underframe is to be relocated.

The standard brake shoe to Drg No. WA/BG 6158 which, is used on BOX wagon, can be locked in position on the brake head by means of a key. The brake shoes should be replaced when worn to 48 mm thickness i.e. when 10 mm metal is left from the base of the shoe.

4.3.2.13 Brake beam (CASNUB 22W, 22NL, 22NLB & 22HS)

Bogies are fitted with unit type fabricated brake beams that slide in the guide cavity provided in the side frame.

Cavities are provided with silico manganese steel liners. The brake heads are integral part of the brake beam.

Brake beam is shown in WD-89033-S/1, however the brake block to WA/BG-6158 is common for all versions.

CASNUB 22W(M)

The bogie is fitted with unit type suspended cast steel brake beam. The brake head is a separate sub assembly which is fixed with brake beam circular end by means of pin passing through brake beam end and brake shoe adjuster along with spring loaded brake head.

Assembly provides rotational flexibility to brake head.

Details are as per Drg No. WD-85084-S/1, WD-88012-S/1 & WD-86034-S/1.

4.3.3 NOMINAL CLEARANCES

The nominal clearances and the tolerances of the bogie assembly are given below.

SN	Description	Type of CASNUB Bogie			
		22W & 22W (Retro)	22W(M)	22NL 22NLB	22HS
1	Lateral clearance between side frame & bolster	18mm	18mm	18mm	25mm
2	Lateral clearance between side frame & adapter	25mm	25mm	16mm	16mm
3	Longitudinal clearance between side frame & adapter	2mm	10mm	9mm	9mm

4	Longitudinal clearance between side frame & bolster	6mm	6mm	6mm	6mm
5	Clearance between anti-rotation lug & Bolster	4mm	4mm	4mm	4mm

Wear limits for various bogie components are given in Table 4.3. The maximum value of lateral and longitudinal clearances (S.N 1 to 5 above) will be a sum of the nominal clearance and wear limit on the corresponding surfaces.

TABLE 4.3

Wear Limits for Bogie Components (Dimensions in mm)

Sr. No.	Description	New or renewed	Worn	Wear limit
1	AXLE BOX			
	Axle Box Crown lugs (Cylindrical Roller Bearings)	159	167	4
	Axle Box Crown seat (Cylindrical Roller bearings)	36.5	33	3.5
	Axle Box Side lugs (Cylindrical Roller bearings)	130	136	3
	Axle Box Sides (Cylindrical Roller bearings)	268	262	3
2	ADAPTER			
	Adapter Crown lugs (Wide Jaw)	156	164	4
	Adapter Crown lugs (Narrow Jaw)	155.5	163.5	4
	Adapter Crown seat	3.5		
	Adapter bore seat to crown seat			
	Wide jaw adapter	48.5	45	3.5

Contd......

	Modified wide jaw adapter Narrow jaw adapter	25.5 26.2	22 22.7	3.5 3.5
	Adapter side Lugs			
	Wide Jaw	130	136	3
	Narrow Jaw	97	103	3
	Adapter sides			
	Wide Jaw	268	262	3
	Narrow Jaw	181	175	3
3	**Side Frames**			
	Side frame wear friction plate	10	6	4
	Side frame column sides	216	206	10
	Side frame and rotation lug	522	528	6
4	**Pedestal Crown roof**			
	Key Seat to pedestal Crown Roof 22W	273	278	5
	Key seat to Pedestal Crown Roof 22W(M)	318	323	5
	Key seat to Pedestal Crown Roof 22NL/NLB/HS	323	32	5
5	**Pedestal Crown sides and Sides of the pedestal**			
	All Bogies – Crown Sides	152	144	4
	Pedestal Sides 22W, 22W(M)	105	101	2
	Pedestal sides 22NL, NLB,HS	81	77	2
6	**Distance between Outer & Inner Pedestal Jaw of CASNUB Bogies**			
	22W & 22W(Retrofitted)	270	278	4

Contd......

	22W(M)	278	286	4
	Pedestal Jaw (Short) for 22NL/NLB/HS	190	198	4
	Pedestal Jaw(Long)for 22NL/NLB/HS	236	244	4
7	BOLSTER			
	Pocket	35 degree on slope		
	Liner	8	3	5
	Bolster land surface Rotation stop lug	444 518	438 512	3 3
8	BOLSTER			
	Outer gib	234/ 241	244/ 251	5
	Inner gib	136	146	5
9	Centre Pivot			
	Wear limit vertical side CASNUB 22W	-	-	5.5
	Others	-	-	4
10	SEAT			
	CASNUB 22W	-	-	4
	Others	-	-	4
11	FRICTION SHOE WEDGE BLOCK			
	Vertical surface from Centre line of spigot	61	54	7
	Slope surface by gauge	-	-	3
12	ELASTOMERIC PADS			
	Type of pad	Nominal Dimension	Dimension after Permanent set	
	Elastomeric pad Side bearer rubber pad	46 114	42 109	

4.3.4 REJECTION RULES

Revision committee of IRCA, New Delhi have finalized rejection rules for casnub bogie which are approved by Railway Board vide their Letter No. 87 M (N) 951/11 dated 7.4.93. The details are given below:-

REJECTIONS - CASNUB BOGIE

1. Bogie Truck

S * Side Frame cracked/broken

S * Bolster cracked/broken

S * Top/Bottom pivot cracked/broken

S * Spring plank broken

@ * Any bush on the plank stripped out

S * Side bearer housing cracked/broken

S * Side bearer housing securing bolts/rivets worked out or improperly secured.

2. Bogie Suspension

S * Any coiled bearing spring cracked or broken

@ * Wedge cracked/broken

3. Bogie brake Gear Defects

S * Brake block deficient (on wagons with operative brake cylinder) or thickness of brake block is less than 10 mm

@ * Any defect in brake rigging preventing application or release of brakes

S * Any safety strap/bracket/hanger nut missing in accordance with rule 2.12.1.4

* Any pin deficient/broken or free to work out in the brake gear

* Brake block key missing or improper fitment

@ * Brake gear adjustment not according to wheel dia

4. Cartridge Bearings

S * Bearing running hot

* Bearing jammed/giving abnormal sound of broken cage or roller

@ * Outer cup broken or cracked, causing leakage of grease

S * Front/Rear seal leakage or seal damaged or leakage of grease on any account

S * End cap screw loose or locking plate missing or tabs of locking plate not properly bent against the face of cap screw head

@ `* Missing side frame key or adapter retaining bolts

5. Adapter

S * Cracked/Distorted Adapter

6. Roller Bearing Axle Boxes

S * Hot Box

S * Cracked or broken axle box

S * Bearing jammed or giving abnormal noise of broken cage/ rollers

S * Axle cap screw loose or locking plate tabs not property bent against the axle cap screw face

7. Air Brake System

S * Stock without air brake connection from coupling to coupling

S * Stock with deficient / damaged or inoperative, distributor valve or its parts, common pipe bracket with control reservoir, cut off angle cocks, isolating cock, check valve, dirt collector brake cylinder, Auxilliary Reservoir and hose coupling assembly for BP.

S * Stock having excessive leakage from the system

@ * Stock without RDSO approved APD for distributor valve

*Stock with damaged pipes and pipe fittings

*Brake gear empty load box defective or handle inoperative

@ * Brake gear slack adjuster or any part of assembly deficient.

8. **CBC Defect**

*Knuckle pin bent, broken or of wrong size

*CBC striker casting cracked more than 25 mm at any location

S * CBC yoke pin support plate missing or its bracket broken

*CBC shank cracked or bent out of alignment

* CBC shank wear plate or housing worn by 10 mm. Shank wear plate missing where originally fitted

*CBC look lift assembly so defective that the coupler cannot be coupled/uncoupled

*Knuckle pin APD deficient

*Overdue ROH/POH by more than one year

*Any wagon loaded more than CC or permitted excess.

S Indicates safety items

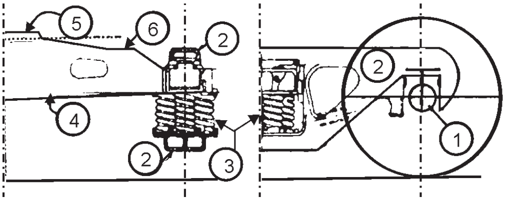
@ shall be applicable for the stock being turned out from Workshops/ROH depts

4.4 AMSTED BOGIE (RDSO REPORT NO. M. 260)

Following are the common types of bogies in this category:

- Mukund National SC-1 bogie (RDSOs Report No.M.262)
- Sumitomo-SM-3 bogie (RDSOs Report No. M. 259)

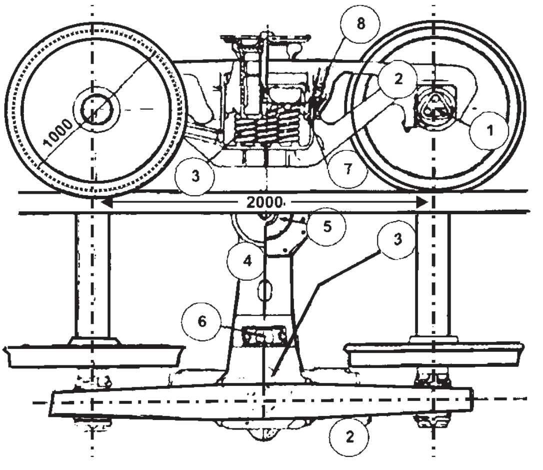
The general arrangement of suspensions of above bogies are shown in **Fig.** 4.13 to 4.15.



1. Roller bearing axle box 2. Side frames 3. Main bearing helical spring nest 4. Bolster (with guide lugs) 5. Centre pivot 6. Side bearer

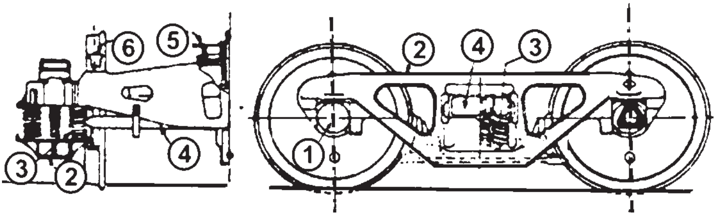
NB : Spring loaded friction snubber is not shown in the **Fig.**

Fig. 4.13 Suspension arrangement of Amsted bogie



1. Roller bearing axle box 2. Side frames 3. Main bearing helical spring nest 4. Bolster (with guide lugs) 5. Centre pivot 6. Side bearers 7. Spring loaded friction snubber shoe 8. Friction snubber spring

Fig. 4.14 Suspension arrangement of Mukund National SC-1 bogie



1. Roller bearing axle box 2. Side frames 3. Main bearing helical. 4. Bolster (with guide lugs.) 5. Centre pivot 6. Side bearers spring nest

NB : Spring loaded friction snubber are not shown

Fig. 4.15 Suspension arrangement of Sumitomo-SM3 bogie

4.4.1 Items and features to be checked during derailment investigation for defects in 3-piece trucks

General

* Defects in wheelsets, axle box assembly, bolster lug side frame members guide surfaces, spring gear,buffing gear and brake gear
* Clearance at the bolster lug-side frame member guide Particular
* Surface of the center pivot for any hindrance to bogie rotation (also check the side bearers in this context)
* Spring loaded friction snubbers.

(**N.B.** These bogies are prone to lozenging or parallelogramming effect while negotiating curves or turnouts.)

Suspension system of a 3-piece truck is similar to that of a diamond frame bogie, except that there is no inter-connecting spring plank between the two side frames. Each of the two spring clusters rests on an enlargement in the side frame itself.

The two side frames and the bolster-three in all are the only structural members, from which feature these bogies derive their name.

Due to this arrangement these trucks suffer from an inherent disadvantage, in that the axles can become angular to the side frames in a parallelogram fashion while negotiating curves or turnouts. This phenomenon is called lozenging or parallelogramming, which gives rise to high flange forces during curve negotiation (see **Fig.** 4.16). This aspect should, therefore, be kept in mind during derailment investigation involving these bogies.

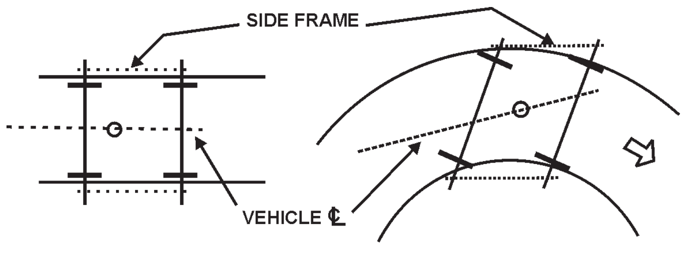


Fig. 4.16 Lozenging or parallelogramming of 3-piece truck

Table 4.4 gives the salient particulars of the above bogies.

Table 4.4
Salient particulars of 3-piece trucks

Description	**Amsted**	**Mukund National**	**Sumitomo-SM3**
Axle load (t)	22.86	22.86	22.86
Bogie rigid wheel base (mm)	2000	2000	2000
Wheel diameter(mm)	1000	1000	1000
Type of axle bearing	AAR Standard cartridge type	AAR Standard cartridge type	AAR Standard cartridge type
Bolster springs (no. per bogie)	14 nests	14 nests	14 nests

Clearance between bolster and bogie frame			
Lateral (mm)	9	8	8
Longitudinal (mm)	4.5	4.5	4.5
Lateral clearance between bearing adaptor and side frame (mm)	4.5	4.5	4.5
Centre pivot	AAR type flat	AAR type flat	AAR type flat
Side bearer	2 roller type	2 roller type	2 roller type
Damping	4 spring loaded friction snubber per bogie	4 spring loaded friction snubber per bogie	4 spring loaded friction snubber per bogie
Stable test speed (KMPH)*	110	110	95
Speed cleared (KMPH)	75	75	75

(***N.B.** : Tested under BOI wagon body. These bogies can also be used under other wagon bodies e.g. BOBS, BOX etc.)

4.5 NEW TYPE OF WAGONS

Following are some of the new type of special wagons for different purpose -

4.5.1 BLCA/BLCB Wagons

BG Low Platform Container Flat wagons (BLCA/BLCB) are used for transportation of ISO containers. These wagons have high pay load (61.0 ton) and a high operating speed (100 kmph). Under frame of the wagon is of all welded construction and is supported over cast steel bogies suitable for 20.32 ton axle load. Low platform height has been achieved with the specific design of bogie frame, bolster and use of smaller wheel diameter. General arrangement of the wagon is shown in **Figure 4.17**. Five cars, one A car at each end and 3 B cars in the central portion, consist a unit. A car is fitted with CBC coupler on the raised end and Slack-less Drawbar on the other end. B car is fitted with Slack-less Drawbars on both the ends. These five cars are to be handled as a unit in rake formation as well as for sending them for ROH/ POH.

Bogie Container Flats are mounted on two cast steel high speed bogies of the type LCCF(C), similar in arrangement to CASNUB bogie (of BOXN /BCN wagon). Flat centre pivot and spring loaded side bearers are provided on the bolster. In a CASNUB bogie, total load is borne by centre pivot. However, this bogie is prone to hunting at higher speeds. Load distribution between centre pivot and side bearers is optimised in LCCF(C) bogie to prevent hunting. Spring loaded side bearers bear 90 % of load under tare condition. 10 % of tare weight and full pay load is borne by centre pivot. The bottom pivot is integral to bolster while the flat top pivot is bolted to the under frame. Manganese steel liners are provided at the bottom and the verticals.

The wagons are provided with a smaller diameter wheel - 840 mm (new), 780 mm (worn). Axles are fitted with Cartridge Tapered Roller Bearing.

A cars are provided with Centre buffer coupler (CBC) on one end and Slack-less draw bar on the other end. B cars are provided with Slack-less draw bars on both ends. Heights of CBC and Slack-less draw bar from rail level are 1105 mm and 845 mm respectively. Thus, A-B and B-B coupling are through Slack-less draw bar and A-A and A-Locomotive coupling are through CBC.

Field data suggest that BLC wagons are extremely prone to derailment while entering a loop through 1 in 8 ½ turnout. A large buff force is likely to be present while the train enters a loop on account of Slack-less Draw Bar system. Under a large buff force, the Slack-less Draw Bar is likely to have increased lateral rigidity, on account of frictional forces in the coupling system. This increased lateral rigidity in combination with a large angular run would increase derailment proneness of the wagon in the sharp curve.

Maintenance Instructions are contained in Maintenance Manual of BLCA & BLCB, April, 2002, by RITES Ltd.

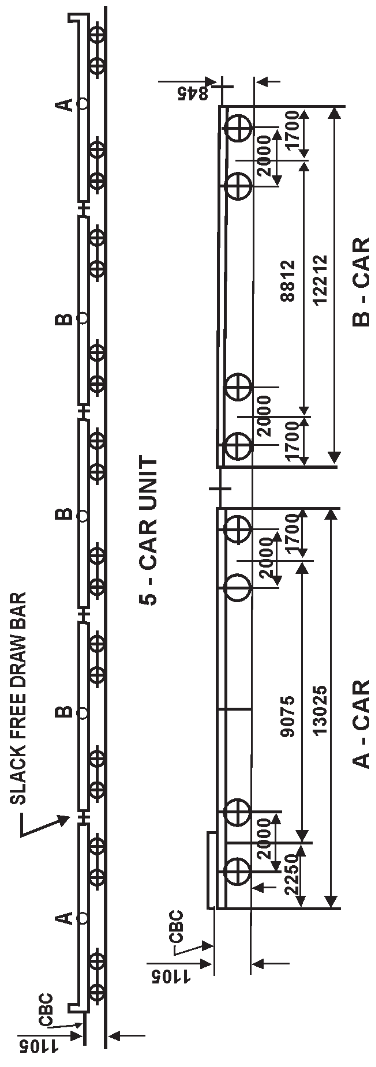


Figure - 4.17- General arrangement of BLCA/ BLCB

4.5.2 BOXNHA

Higher axle load wagon suitable for 22.1 t axle load and 8.25 t/m TLD for coal loading .It has higher side walls as compared to BOXN wagon. Fit for 100 kmph.

4.5.3 BOXNHS

The wagon is a variation of BOXN wagon with operating speed of 100 kmph in empty and 100 kmph in loaded condition. In this wagon, high speed bogies have been provided. Other parameter of the wagon are the same as that of BOXN wagon.

4.5.4 BOXNHL

This bogie open wagon was designed for 22.9t axle load and 250mm longer than BOXNHS wagon. The wagon is manufactured using stainless steel and cold rolled sections. Maximum speed is 100 (empty)/ 75 (loaded) kmph.

4.5.5 BOXNEL

This wagon has been designed for transportation of iron ores, coal etc. BOXNEL wagons are fitted with Casnub-22NLC bogies with a maximum axle load 25 t. Maximum speed is 60 (empty)/ 45 (loaded) kmph.

4.5.6 BCNA

This wagon is an improved BCN wagon having reduce length and increased height by keeping the volumetric capacity as the same. Also, the wagon has fully welded construction.

4.5.7 BCNAHS

This wagon is a variation of BCNA wagon with high speed bogie CASNUB-22HS-BOGIE. The other parameter of this wagon are the same as of BCNA wagon

4.5.8 BCNHL

This wagon was designed at 22.9t axle load in 2006 for transportation of food grain, fertilizer and bag quantities.

4.5.9 BRNA

This bogie rail wagon is an improvement of BRN wagon. The design is riveted cum welded construction. It has higher pay to tare ratio as compared to BRN wagon.

4.5.10 BOBYN

This wagon is designed for transportation of ballast for engineering department. It has two chutes at bottom for regulated discharge of ballast on both sides of rails.

4.5.11 BOXNCR

The wagon is a variation of BOXN wagon with its body in IRS M-44 steel. Its under frame is, however, made of mild steel. This has been done to provide better corrosion resistance to wagon body. All parameters of this wagon are same as that of BOXN wagon.

4.5.12 BFKN

Broad gauge Bogie Container flat wagon type BFKN is modified version of BFKI wagons

4.5.13 BFKNCR

Air Brake CASNUB bogie container flat wagons owned by container corporation LTD. (being converted from BFKI).

4.5.14 BCCN

This is Double Decker Bogie covered wagon for transportation of automobile cars. It has Low platform, 840 mm dia wheel, and is fit for high speed (100 km/h.).

4.5.15. BFNS

Special wagons for transportation of HR coil, Tare weight 23.6 t, payload 57.7 t suitable for accomodating various sizes of coils Adjustable stoppers have been provided for suitable placement of coil in the groove and preventing longitudinal shifting of coils.

4.5.16 BCW

It is privately owned by M/s. Bulk Cement Corporation India Ltd. and are based at Wadi, Sholapur Division of CR, to run between Wadi and Kalamboli (Mumbai Division). Axle Load = 20.32 t. The wagon is fitted with Air Brake.

4.5.17 BTPGLN

Bogie liquefied petroleum gas tank wagon. This wagon is a variant of BTPGL with air brake and casnub trolley.

Annexure -1

WAGON NOMENCLATURE

SN	Wagon Code	Full name	Remarks
1.	**OPEN WAGON**		
1.1	BO*	Bogie Open Wagon	Bogie Open Wagon
1.2	BOC*	Bogie Open Wagon, Coal	Bogie Open Coal Wagon
1.3	BOXC	Box Open Wagon (81.28t)	Variation of general purpose Open Box wagon with CBC couplings.
1.4	BOXT	Box open wagon (81.28 t.)	Variation of general purpose Open Box wagon with Transition coupling.
1.5	BOX-SR	Box Wagon With Sliding Roof	
1.6	BOI	Bogie Gondola Wagon (81.28t)	Specially for the movement of high density items like iron and manganese ore. The height of the body is reduced
1.7	BOZ	Bogie Open Wagon (81.28t)	An improvement over the BOX wagon,

Contd......

SN	Wagon Code	Full name	Remarks
1.8	BOY	Bogie Gondola Wagon (91.4t)	For transportation of minerals/ore in closed circuit with an axle load of 22.9t.
1.9	BOXN	Bogie Open Wagon (81.28t)	20.32 t axle load, operating speed 80 kmph (empty) and 75 kmph (loaded)
1.10	BOXNHS	Bogie Open Wagon (81.28t)	Variation of BOXN wagon with high speed bogie. Operating speed of 100 kmph in both empty and loaded condition.
1.11	BOXNCR	Bogie Open Wagon (81.28t)	Variation of BOXN wagon with better corrosion resistance.
1.12	BOXNHA	Bogie Open Wagon (88.40t)	It has higher sidewalls compared to BOXN wagon, for transportation of coal to axle load of 22.1 t.
1.13	BOST	Bogie Open Wagon (81.28t)	Designed for transportation of coal as well as steel coils.
1.14	BOSTHS	Bogie Open Wagon, HIGH SPEED (81.28t)	This wagon is variation of BOST wagon with high-speed casnub bogie.
1.15	BOXN(LW)	Bogie Open Light Weight Wagon	
1.16	BOXNHL	Bogie Open Higher Load Wagon	Designed for 22.9t axle load and 250mm longer than BOXNHS wagon.
1.17	BOXNEL	Bogie Open Enhanced Loading Wagon	Fitted with Casnub-22NLC bogies with a maximum axle load 25

Contd......

SN	Wagon Code	Full name	Remarks
			t., for transportation of iron ores, coal etc.
1.18	BOXNR	Bogie Open Wagon	Maximum axle load 22.9 t.
1.19	BOYEL	Bogie Open Enhanced Loading Wagon	Casnub-22NLC bogies with a maximum axle load 25 t
2.	**COVERED WAGON**		
2.1	BCN	Standard Covered Bogie Wagon	20.32 t axle load
2.2	BCNA	Standard Covered Bogie Wagon	Improved BCN wagon having reduce length and increase height
2.3	BCNAHS	Standard Covered High Speed Bogie Wagon	Variation of BCNA wagon with high speed bogie CASNUB-22HS
2.4	BCR*	Bogie Covered Wagon	With cast steel bogie, provided with narrow door opening.
2.5	BCX Mk-I	Bogie Covered Wagon Special	Includes special feature of BOX wagon and provided with wider sliding doors.
2.6	BCX Mk-II	Bogie Covered Wagon Special	Includes all the special features of BOX wagon. It has provision of swing cum flap doors.
2.7	BCXC	Standard Covered Bogie Wagon	UIC bogie and center buffer coupler.
2.8	BCCN	Bogie Covered Double Decker Wagon	A set of five wagon consisting of two end consisting of two end wagons known as ‘A’ wagon and three

Contd......

SN	Wagon Code	Full name	Remarks
			middle wagon known as ‘B’ wagon. Both the wagons are double decker having two floor for transportation of automobile cars.
2.9	BCCNR	Bogie Covered Wagon	For transportation of automobile with single tier system transportation.
2.10	BCNHL	Bogie Covered Wagon	22.9 t axle load
2.11	BCXN	Standard Covered Bogie Wagon	Modified form of BCX wagon provided with air brake, Casnub NLB bogie with center buffer coupler.
3.	**FLAT WAGON TYPE**		
3.1	BR	Bogie Rail Wagon	16.3t axle load for transportation of rails, with cast steel bogie.
3.2	BRH	Bogie Rail Wagon, Heavy	20.3t axle load and plate fabricated bogie of BOX type for transportation of rails and heavy steel sections.
3.3	BRS	Bogie Rail Wagon, Special	Improvement over BR, end loading of special defense vehicles.
3.4	BRHC	Standard Bogie Rail Wagon Heavy	Similar to BRH wagon. Provided with stronger CBC and reduced coupling length.
3.5	BRN	Standard Bogie Rail Wagon Heavy	Improvement of BRH, with air brake and welded construction.

Contd......

SN	Wagon Code	Full name	Remarks
3.6	BRHT	Standard Bogie Rail Wagon Heavy	Similar to BRH wagon, with provision of transition screw coupling.
3.7	BRNA	Bogie Rail Wagon	Improvement of BRN wagon, having higher pay to tare ratio
3.8	BRNAHS	Bogie Rail Wagon (With High Speed Bogie)	Similar to BRNA wagon, with High Speed CASNUB bogie.
3.9	BFNS	Bogie Flat Steel Coil Wagon	For transportation of hot rolled/cold rolled coils, plates, sheets and billets etc.
3.10	BRHNEHS	Bogie Rail Wagon	For departmental use of Engineering department for Track Relaying Trains (TRT) especially loading RCC sleepers.
4.	**HOPPER WAGON TYPE**		
4.1	BOB	Bogie Hopper Wagon	Conventional hopper wagon with side discharge system for ballast transportation, provided with diamond frame bogie.
4.2	BOBC	Bogie Hopper Wagon	Having center discharge system and cast steel bogie.
4.3	BOBX	Bogie Hopper Wagon	For transportation of ore etc. with center and side discharge system
4.4	BOBX (MOD)	Bogie Hopper Wagon	Identical with BOBX wagon. Modified repose angle of hopper to 400.
4.5	BOBS	Bogie Hopper	For transportation of

Contd......

SN	Wagon Code	Full name	Remarks
	(TALBOT)	Wagon Special	iron ore, with diamond frame ISW design bogie.
4.6	BOBS (ISW)	Bogie Hopper Wagon Special	For transportation of iron ore, with diamond frame ISW design bogie.
4.7	BOBSMK-II	Bogie Hopper Wagon	A variant to BOBS wagon with Casnub bogie.
4.8	BOBSN	Bogie Hopper Wagon	Variation of BOBS wagon with air brakes and Casnub bogie
4.9	BOBR	Bogie Open Rapid Discharge Hopper Wagon	For transportation of coal with bottom discharge facility
4.10	BOBRN	Bogie Open Rapid Discharge Hopper Wagon	Variation of BOBR with air brake and has two doors.
4.11	BOBRNHS	Bogie Open Rapid Discharge Hopper Wagon	Variation of BOBRN with high speed Casnub 22 HS bogie.
4.12	BOBY	Bogie Hopper Wagon	For transportation of ballast with regulated discharge
4.13	BOBYN	Bogie Open Rapid Discharge Hopper Wagon	Variant of BOBY with air brake
4.14	BOBYNHS	Bogie Open Rapid Discharge Hopper Wagon	Variant of BOBYN with high speed Casnub 22HS bogie.
4.15	BCBFG	Bogie Covered Hopper Wagon For Food Grain	For transportation of food grain in Bulk. Provided with CASNUB-22HS MOD-I. bogie,

Contd......

SN	Wagon Code	Full name	Remarks
5.TANK WAGON TYPE			
5.1	BTAL	Bogie Liquid Ammonia Tank Wagon	
5.2	BTALN	Bogie Ammonia Tank Wagon	Provided with air brake
5.3	BTO	Bogie Oil Tank Wagon	
5.4	BTPN	Bogie Oil Tank Wagon	For transportation of petroleum products, provided with air brake.
5.5	BTPGL	Bogie L.P.G. Tank Wagon	fitted with UIC bogie, Vacuum brake, Transition coupling & Roller bearing.
5.6	BTALM	Bogie Ammonia Tank Wagon	Fitted with UIC bogie, Transition coupling & Vacuum brake.
5.7	BTPGLN	Bogie L.P.G. Tank Wagon	This wagon is a variant of BTPGL with air brake and casnub bogie.
5.8	BTCS	Bogie Caustic Soda Tank Wagon	
5.9	BTAP	Bogie Alumina Tank Wagon	For transportation of Alumina powder. Fitted with 22-NLB bogie, Single pipe Air Brake system & non transition high tensile CBC.
6.	**BRAKE VAN TYPE**		
6.1	BVZI	Bogie Brake Van	This 8-wheeled brake van with ICF bogie.
7.	**CONTAINER WAGON TYPE**		
7.1	BLLA	Bogie Low Platform Longer Container Flat Wagon	For transportation of 22' 24' & 45' containers along with 20' & 40' long

Contd......

SN	Wagon Code	Full name	Remarks
			ISO containers at an operating speed of 100kmph with LCCF 20 (C) Bogie. BLLA (A-Car) wagons are provided at extreme ends in formation of one unit of 5- cars (with 3 BLLB wagons in middle).
7.2	BLLB	Bogie Low Platform Longer Container Flat Wagon	
7.3	BLCA	Bogie Low Platform Container Flat Wagon	For transportation of 20’ & 40’ long ISO containers at an operating speed of 100kmph with LCCF 20 (C) Bogie. BLCA (A-Car) wagons are placed at extreme ends in formation of one unit of 5- cars (with 3 BLCB wagons in middle).
7.4	BLCB	Bogie Low Platform Container Flat Wagon	
7.5	BFKN	Bogie Container Flat Wagon	Modified version of BFKI wagons, Air brake & enhanced pay load
7.6	BLCAM	Bogie Low Platform Container Flat Wagon	Modified BLC wagons for carrying enhanced pay load , with axle load of 22t.
7.7	BLCBM	Bogie Low Platform Container Flat Wagon	Modified BLC wagons

■ ■ ■

CHAPTER 5

COACHING STOCK

There are, basically, four types of conventional bogie coaching stock on the Indian Railways :

- IRS
- ICF Laminated (Schlieren)
- BEML (Bharat Earth Movers Ltd.)
  (this bogie was earlier known as HAL-MAN)
- ICF all-coil

Recently, FIAT bogie (on LHB coach) has been inducted on the Indian Railways. This can be called as the modern coaching stock. Features of this bogie are very different and, hence, they are discussed separately.

All the above conventional stock have two-stage suspension bogies. In general, the suspension arrangement of such bogies is as under:

5.1 VERTICAL LOAD TRANSMISSION

A bogie frame rests on primary springs which are supported on the axle boxes. From the bogie frame are hung swing links, which support (either directly or through a spring plank) the secondary springs. The secondary springs support the bolster on which the vehicle body rests, with its weight primarily carried either on centre pivot, or two side bearers.

Fig. 5.1 illustrates the schematic arrangement of such a suspension.

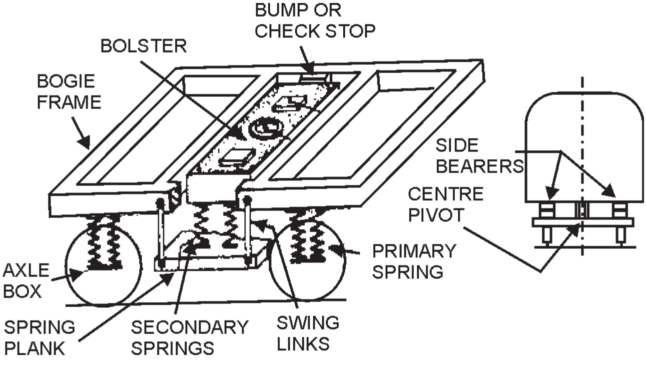


Fig. 5.1 Schematic arrangement of suspension system of a typical coaching stock bogie

In IRS & ICF Laminated bogies, body weight is primarily supported on centre pivot on the bolster at each of the two bogies. Since one-point support at either end of the vehicle body would be unstable (in rolling mode), two side bearers on the bolster provide the necessary support during vehicle motion, otherwise, there is clearance between the body and the side bearers.

In BEML & ICF all-coil bogies, body weight is supported on the two side bearers on the bolster, the centre pivot not carrying any vertical load. However, in all the four types of bogies, the centre pivot acts as the centre of bogie rotation and transmits the tractive and braking forces from the body to the bolster.

Hence, for defects in the context of hindrance to bogie rotation, surfaces of the following should be checked (see Bogie rotation Chapter 3).

IRS ICF Laminated	Mainly, the centre pivot. Since one of the two side bearers also carries load during vehicle motion, the side bearers also should be checked for any defecs or features that would tend to cause jamming or hinder free bogie rotation.
BEML ICF all-coil	Mainly, the two side bearers

5.2 Transmission of tractive and braking forces

The vehicle body transmits these forces to the bolster through the centre pivot. Thence the transmission is directly to the bogie frame through swivelling anchor links, bypassing the secondary suspension. From bogie frame to the axle, these forces are transmitted as under.

IRS	:	axle guard bearing against the axle box groove
ICF Laminated	:	through dash-pot
BEML	:	through axle holding arms (also called axle links)
I CF all-coil	:	through dash-pots cum axle guide

Lateral restraint for axle held in the bogie is provided as under :

IRS	:	axle guard in axle box groove
ICF Laminated	:	through dash-pot
BEML	:	by bronze rollers with ferrozol bushes moving up and down in precom pressed state on vertical rubber bonded plates. There are a set of one plate and 2 rollers (one roller on either side of the plate) on either side of each axle box i.e. 32 rollers in all per coach (16 per bogie.)
ICF all-coil	:	by dash-pot cum axle guide

In BEML bogie, the axle holding arm is provided with silent block (rubber bonded) bushes at its ends where swivelling action takes place with vehicle oscillations. As the rubber wears out, the longitudinal play of the axle in the bogie frame increases, leading to severe vibrations of the wheelsets. The condition of these bushes and bearings and play of the axle holding arms should, therefore, be checked.

The lateral sway of the bolster is checked beyond certain limits by bump or check stops on the sides of the bogie frame . A heavy indentation at such locations is indicative of the bogie having been subjected to high flange forces.

5.3 ICF ALL-COIL BOGIE

Various components of ICF All-coil Bogie and their features are discussed in the following paras.

5.3.1 Bogie Assembly

The bogie frame and components are of all-welded light construction with a wheel base of 2.896 metre. The maximum axle load bearing capacity is 13 t and 16.25 t. The wheel sets are provided with self-aligning spherical roller bearings mounted in cast steel axle box housings. Helical coil springs are used in both primary and secondary suspension. The weight of the coach is transferred through side bearers on the bogie bolsters. The ends of the bogie bolsters rest on the bolster helical springs placed over the lower spring beam suspended from the bogie frame. A photograph of ICF All-coil Bogie is shown in **Fig.** 5.2. General arrangement of ICF All-coil Bogie is shown in Figure 5.3. A photograph of the Secondary Suspension arrangement is shown in **Fig.** 5.4.

5.3.2 Swing Links

Swing Link (also called Hanger) isolates lateral movement of the coach from that of the bogie frame to a large extent, thus, reducing lateral oscillations of the coach and lateral flange forces. It should swivel freely in the lateral direction. Any tendency of jamming of swing links (at points of articulation in bogie frame

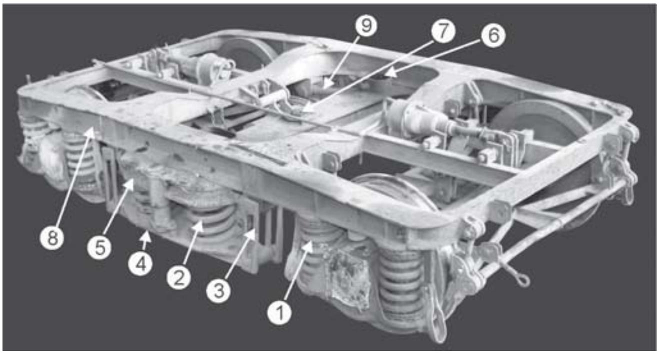
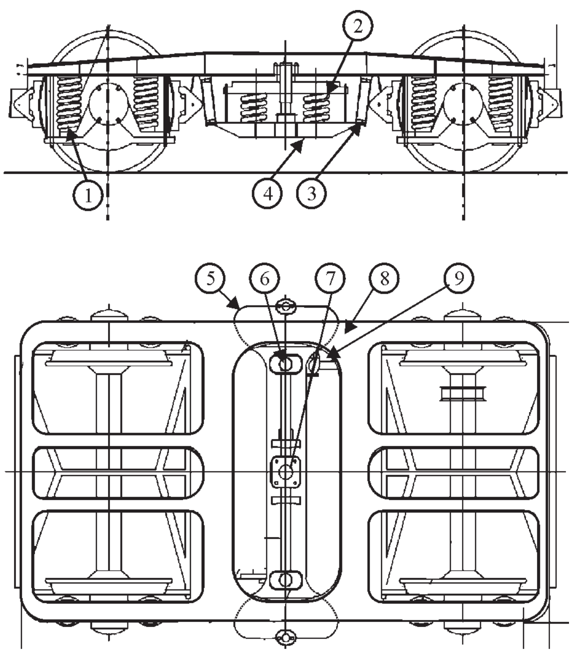


Fig. 5.2 ICF All-coil Bogie



1.Axle box spring 2. Bolster spring 3.Swing link 4. Spring plank 5. Bolster 6. Side bearer 7. Centre pivot 8. Bogie frame 9. Anchor link

Fig. 5.3 ICF All-coil Bogie – General arrangement

and spring plank) during lateral swing would result in occurrence of higher flange forces (Y) and higher lateral accelerations in the coach.

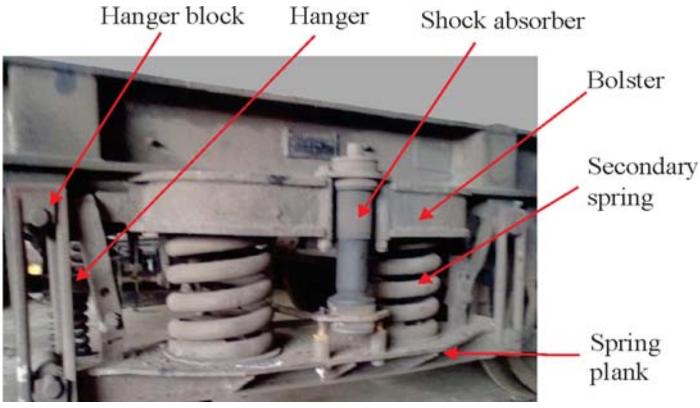


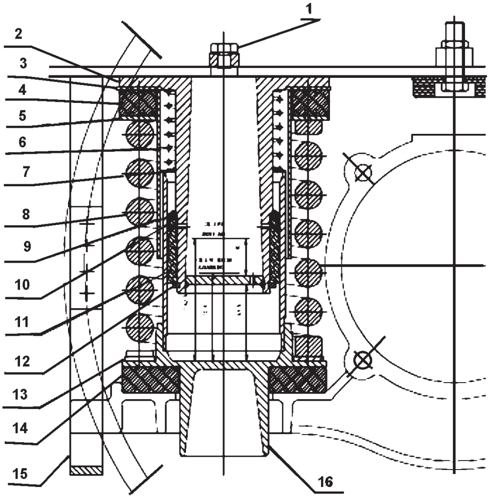
Fig. 5.4 ICF All-coil Bogie – Secondary suspension

Hanger is an important component of bogie as it transfers both vertical and lateral loads at the secondary suspension stage (Figure 5.4). Due to swivel action, wear occurs on the Hanger block and inside surface of Hanger. Wear limits of Hanger block and limit of clear inside length of Hanger are stipulated. Upon exceedence of the limits, these components are replaced. Else, chances of failure of Hanger would be high.

Swing link is a vulnerable component. Hence, its failure should be observed at the derailment site.

5.3.3 Axle box guide with dash pot

Axle box guides are of cylindrical type welded to the bottom flanges of the bogie side frame (Figure 5.5). These guides together with lower spring seats located over the axle box wings, house the axle box springs and also serve as shock absorbers. These guides are fitted with guide caps having nine holes of diameter 5 mm equidistant through which oil in the lower spring seat passes under pressure during dynamic oscillation of coach and provide necessary damping to primary suspension to enhance better riding quality of coach. This type of rigid axle box guide arrangement eliminates any longitudinal or transverse relative movement between the axles and the bogie frame, through provision of bronze bushes at the lower end of the axle guide, and transmits lateral and longitudinal forces. The assembly is made oil tight by provision of sealing rubber rings.



1.Screw with sealing washer 2. Guide 3. Protective tube 4. Rubber washer 5. Top spring seat 6. Dust shield spring 7. Dust shield 8. Spring 9. Guide ring 10. Rubber packing ring 11. Guide bush 12. Circlip 13. Compensating ring 14. Rubber washer 15. Safety strap 16. Lower spring seat

Figure 5.5 Axle Guide with Dash pot

On the bogie side frames, directly above the dash-pots, tapped holes are provided for replenishing oil in the dash pots. Special screws with copper asbestos washers are screwed on the tapped hole to make it air tight. Oil level (above top of guide cap) is measured by insertion of a flexible wire through the hole. A lower oil level would result in decrease in damping, which would adversally affect comfort and safety.

5.3.4 Centre pivot and side bearer arrangement

The centre pivot pin joins the body with the bogie and transmits the tractive and braking forces on thebogies (Figure 5.6). It does not transmit any vertical load. It is equipped with rubber silent block bushes which tend to centralise the bogies with respect to the body and, to some extent, control and damp the angular oscillations of the bogies. During assembly, graphite grease is applied on the Centre Pivot Pin to reduce friction and it is secured with a cotter and pin arrangement.

The side bearer arrangement consists of a machined steel wearing plate immersed in an oil bath and a floating bronze-wearing piece with a spherical top surface kept in it, on both sides of the bogie bolster (Figure 5.7). The coach body rests on the top spherical surface of these bronze-wearing pieces through the corresponding attachments on the bottom of the body-bolster. The whole arrangement is provided with a cover to prevent entry of dust in the oil sump. During Schedule Inspection, oil is replenished if its level has goes down below the level of last thread of oil filling cup.

Thickness of the Wear plate are 10 mm (new) and 8.5 mm (condemning). Thickness of Bronze piece are 45 mm (new) and 42 mm (condemning). Condemning thickness for High Speed ICF Bogie for Rajdhani are 9.0 mm and 43.5 mm, respectively.

Obstruction to bogie rotation increases vulnerability to derailment, particularly in sharp curves, to a very large extent.. Therefore, defects which cause such obstruction, such as excessive/ uneven wear or formation of an obstruction/ notch/ ridge on wear plate/ bronze piece, lack of lubrication in oil bath; lack of verticality, tendency of jamming of centre pivot pin etc. must be thoroughly checked during investigation of derailment.

5.3.5 Anchor link

The floating bogie bolster which supports the coach body is eld in position longitudinally by the anchor links which are pinned to the bolster sides and the bogie Transoms. One anchor link is provided on each side of the bolster diagonally across. The links

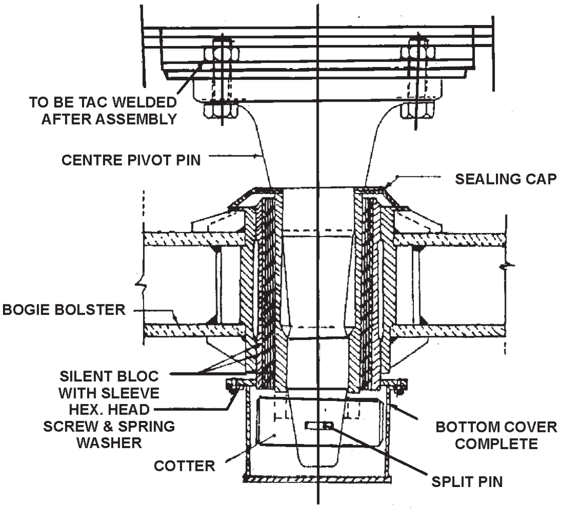


Fig. 5.6 Centre Pivot Pin

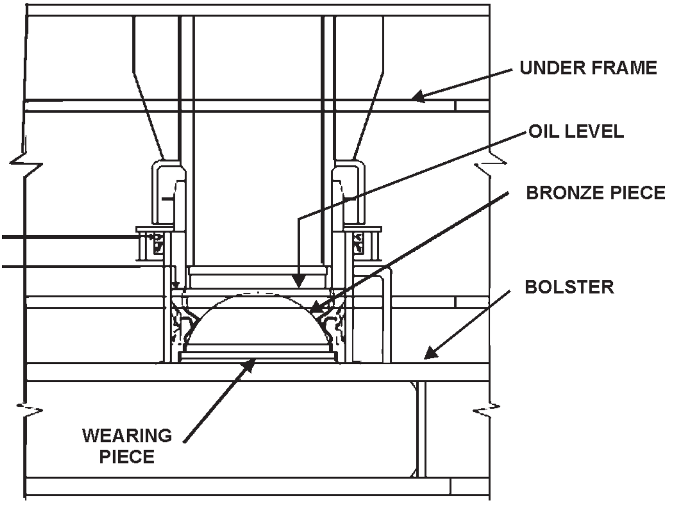


Fig. 5.7 Side Bearer

can swivel universally to permit the bolster to rise and fall and sway side wards. They are designed to take the tractive and braking forces. The anchor links are fitted with silent block bushes, in order to reduce the jerks during acceleration/ braking.

A broken Anchor Link would result in eccentric transfer of logitudinal load, causing increased angular run and lateral forces.

5.3.6 Equalising stay

This device has been provided on bogies between the lower spring plank and the bolster to prevent lateral thrust on the bolster springs which have not been designed to take the lateral forces. These links have pin connections at both ends and, therefore, can swivel freely. A photograph of Equalising stay and Anchor link arrangement is shown in Figure 5.7.

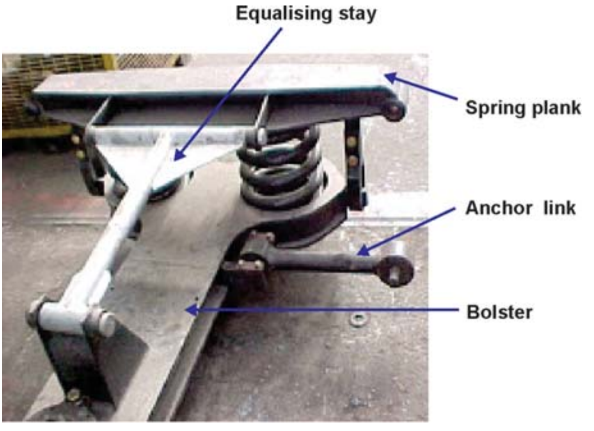


Figure 5.8 Equalising Stay Arrangement

5.3.7 Helical spring and shock absorbers

Drawing number and Free height, Loaded height and Grouping details of spring for BG Main Line coaches is shown in Table 5.1, 5.2 and 5.3. Springs should be grouped in three groups as shown in the Table, depending upon their height under test load. Spring groups are to be marked as per the colour code : A - Yellow, B -

Oxford Blue, C – Green. For pairing, springs should be selected from the same group. Same group of spring is to be used on a particular bogie.

Defects of spring and their role in derailment has already been discussed in Chapter 3 – Rolling Stock Defects. Springs should be observed and measured accordingly during derailment investigation.

Three vertical clearances, between Axle Box crown and bottom of Bogie Frame, top of Bolster and bottom of Bogie Frame , and top of Bogie Frame and bottom of coach body are stipulated for various ICF coaches. Abnormal variation in these clearances may be caused by either defect in springs or excess/ uneven load. In such situations, the cause should be thoroughly investigated.

Two Hydraulic shock absorbers (one on each side), having a capacity of ± 600 kg at a speed of 10 cm/sec. are fitted to work in parallel with the bolster springs to provide damping for vertical oscillations. Leakage of oil and physical damage are the common items to be observed.

Table 5.1
Drawing code of springs for ICF BG coaches
(Reference RDSO Amendment slip no.5 of September 2001 to STR WD-01-HLs-94) (Rev.1 May 95)

Type of spring	Type of bogies	ICF Drg.No.	Drg.Code No.
Axle Box	All Non AC ICF Type	F-0-1-006	A01
	All AC ICF type	WTAC-0-1-202	A03
	Power car	WLRRM2-0-1-202	A04
	Double decker	DD-0-1-001	A06
	High capacity Power Cars	WLRRM8-0-1-802	A09
	High capacity parcel van	RDSO/SK-98017	A10
Bolster	All Non AC ICF type	F-0-5-002	B01
	All AC ICF type	WTAC-0-5-202	B03
	Power car	WLRRM2-0-5-202	B04
	Double decker	DD-0-5-003	B06
Bolster	High capacity Power Car	WLRRM8-0-5-802	B11 B13
	High capacity parcel van	RDSO/SK-98018	B15 B16

Table 5.2
Grouping of Axle box spring

Code	Free Height (mm)	Test load (Kg)	Acceptable Height under test load (mm)	Groups as per loaded spring height (mm)		
				A Yellow	B Oxford Blue	C Green
A01	360	2000	279-295	279-284	285-289	290-295
A03	375	2800	264-282	264-269	270-275	276-282
A04	372	3000	265-282	265-270	271-276	277-282
A06	337	2400	269-284	269-273	274-279	280-284
A09	360	3000	277-293	277-282	283-288	289-293
A10	315	1800	276-289	276-279	280-284	285-289

Table 5.3
Grouping of Bolster spring

Code	Free Height (mm)	Test load (Kg)	Acceptable Height under test load (mm)	Groups as per loaded spring height (mm)		
				A Yellow	B Oxford Blue	C Green
B01	385	3300	301-317	301-305	306-311	312-317
B03	400	4800	291-308	291-296	297-303	304-308
B04	400	6100	286-304	286-291	292-297	298-304
B06	416	4200	280-299	280-286	287-292	293-299
B11 **B13**	386	6700	306-322	306-311	312-317	318-322
B15 **B16**	393 386	6000	256-272	256-261	262-267	268-272

5.4 Summary

Various suspension particulars, damping arrangements, relevant RDSOs Reports, Speeds tested and cleared for, etc. in respect of the foregoing four coaching stock bogies are summarised in Table 5.4. The particular suspension arrangements and defects and features to be checked during derailment investigation covered in the above paras are shown in **Fig.** 5.9 to 5.11.

The suspension arrangements of M.G. IRS and ICF All coil bogies are more or less similar to those of B.G. ones. The former have been cleared for maximum speeds of 80 km/h and 100 km/h respectively, speeds tested and found stable for, being 100 km/h and 110 km/h.

5.5 Trolley frame defects - rejections (Prefixed by S)

- Trolley frames visibly out of square or damaged.
- Any member of trolley frame cracked or welding failed.This shall include knee plates, gusset plates and diagonal bars with their rivets broken or deficient
- Bearing spring hanger bracket cracked or broken
- Bolster or spring plank cracked or broken, or bolster improperly secured
- Bolster safety bracket cracked, broken or its pin or retainer deficient
- Rocker bar/swing link broken and/or top or bottom arch bar displaced, cracked or broken
- Pivot plate/casting or pin broken, cotter and/or nut deficient
- Any part of side bearer assembly improperly secured or deficient

- Bolster rubbing block/plate (on BEML) deficient
- Centre bearing carrier (on BEML) broken or cotter deficient

- Swing bolt or its safety bracket (on BEML) broken or deficient

- Bolster top truss (on BEML) broken/damaged and any of its pins deficient broken or free to work out

- Bolster bottom bars (on BEML) cracked, broken or damaged or its bracket or bracket bolts deficient

- Axle box holding arm (on BEML type trolley) broken or deficient or any of its securing studs/nuts loose or deficient

- Anchor link broken or its pin deficient or free to work out.

- Any safety strap bracket of wrong size/broken deficient or improperly secured

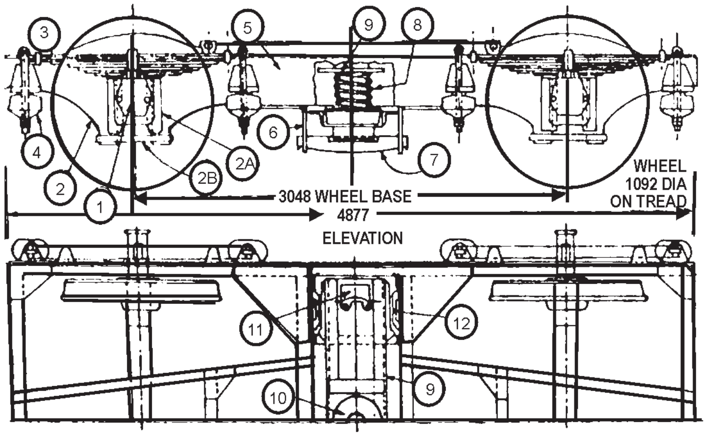
- Equalizing stay (on ICF all coil type) broken or deficient.

TABLE 5.4 : COACHING STOCK

Sr. No.	Item	IRS (Rivetted under frame)	Schlieren (ICF Laminated) (Welded)	BEML(HAL-MAN)	ICF all coil
	RDSO's report no.	M-130 M-277	M-130 M-188 M-277	M-130	M-274 C&M-1
(1)	(2)	(3)	(4)	(5)	(6)
1.	Type of bearings at Journal	Conventional plain bearing axle box	Roller bearings	Roller bearings	Roller bearings
2.	**Suspension Vertical**				
	i) Primary springs (axle level)	Laminated springs plus auxiliary springs or rubber pad.	Coil-springs with dash-pot.(Rubber pad on top).	Coil springs Bronze rollers with ferrozol bushes pressing against rubber bonded vertical plates – a set of 2 rollers and a plate on either side of axle box provide the lateral restraint as well as vertical damping – 32 rollers per coach)	Coil springs with dash-pot (rubber pads- optional)
ii)	Secondary springs (Bolster level)	Coil springs (no damping). 4 nests of coil springs (each	Laminated springs (grease lubricated)	Coil springs (4 nests of 2 springs each). (with vertical shock absorbers 2 per bogie).	4 Nos. Coil springs (on independent spring planks). (vertical shock abs-orbers 2 per bogie).

(1)	(2)	(3)	(4)	(5)	(6)
3.	Suspension Lateral swing links (Length & angle of inclination)	343mm (13 ½") 110 to vertical	286mm (11 ¼") 70 to vertical	450/400mm 6.50 to vertical	410/425 mm 70 to vertical
4.	Lateral shock absorbers	No	No (provided on some)	Yes(2 Nos. Per bogie) (between under-frame and bogie)	Yes (in some of them e.g. for Rajdhani express)
5.	Loading (vertical)	On centre pivot lubricated by oil soaked cotton waste. (2 side bearers also provided for restraining rolling oscilliation).	On centre pivot. (intermediate pieces with fabric liners serve as side bearers for restraining rolling oscillations)	On 2 side bearers (immersed in oil bath with oil seal)	On 2 side bearers 1600 mm apart (immersed in oil bath with oil seal)
		(Centre pivot also transmits the longitudinal forces between the vehicle body and the bolsters).		(Centre pivot acts only as centre of rotation and for transmitting longitudinal forces between the vehicle body and bolster).	
6.	Location of axle in frame. (for transmission of longitudinal forces between the axle and the bogie frame).	Conventional axle guard-hom cheek guides.	Telescopic cylindrical guides immersed in oil	Two axle holding arms (with silent block bushes).	Telescopic cylindrical guides immersed in oil.
7.	Transmission of longitudinal forces between bolster and bogie frame.	Check guides of the bogie frame	2 anchor links	2 drag (anchor) links, working in parallel with 2 leaf springs.	2 anchor links.

(1)	(2)	(3)	(4)	(5)	(6)
8.	Permissible swing of bolster in lateral direction.	31mm	41mm stops are provided at end of travel)	25mm (Rubber bump	57mm
9.	General	1) Lateral riding better than ICF Laminated, but more rolling oscillations. 2) Bolster starts hitting the bogie frame due to excessive swing on 30curve at 1 03km/h(equivalent to cant deficiency of 129 mm) Ride index >3.5 hence O.K. only for speeds equivalent to 110mm cant deficiency on track maintained to Rajdhani	While vertical riding is satisfactory, lateral riding becomes uncomfortable at higher speeds. Lateral shock absorbers do improve lateral ride, but as ICF all coil coaches are used for higher speeds, these alterations are unnecessary.		Transverse damping by friction between links and hanger blocks.
10.	Speeds cleared for	110 km/h.	100km/h.	110 km/h	130 km/h
11.	Maximum test speed upto which found satisfactory.	125 km/h.	110 km/h.	125 km/h	145 km/h

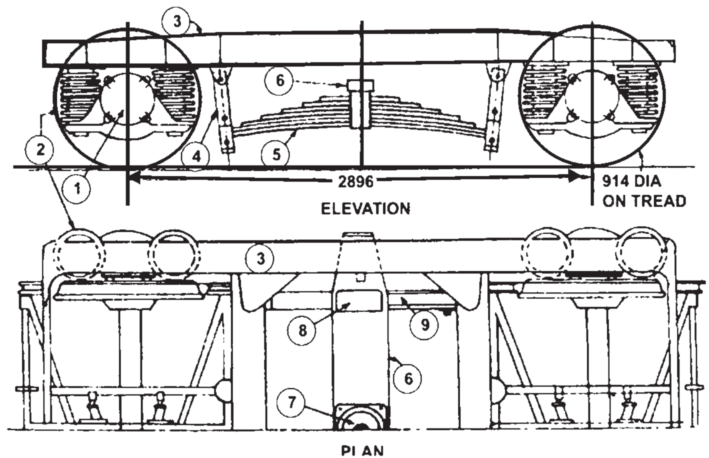


1. Plain Bearing Axle Box
2. Axle Guard
2A. Horn check
2B. Birdle Bar
3. Primary Springs (Laminated)
4. Auxiliary Rubber Block (Or Spring)
5. Bogie Frame (Rivetted)
6. Swing Links
7. Spring Plank
8. Secondary Spring (Helical)
9. Bolster
10. Centre Pivot
11. Side Bearers
12. Check Guides (Connected to Bogie Frame)

Fig.5.9 Suspension arrangement of IRS (B.G.) bogie.

Particular items and features to be checked

* Surface of centre pivot - for any hindrance to bogie rotation (also check surfaces of side bearers in this context).
* Bolster sway - any heavy indentation marks indicative of sway being excessive.
* Swing links - whether swivelling freely and not jamming.
* Helical springs - specific deflection, to check possibility of springs getting fully compressed in dynamic condition.
* Condition of auxiliary rubber block or spring.

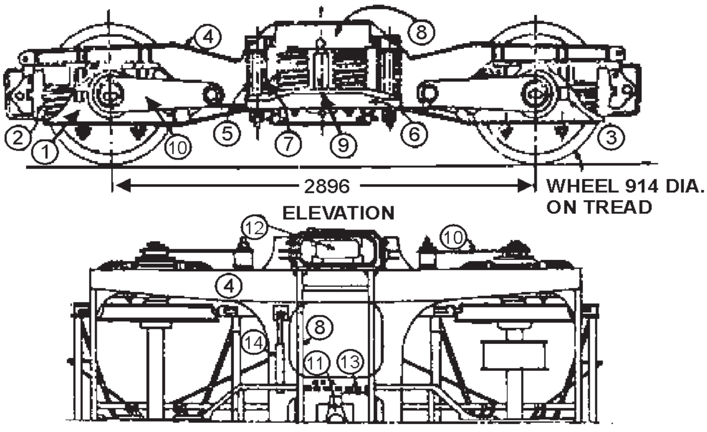


1. WING TYPE ROLLER BEARING AXLE BOX
2. PRIMARY SPRINGS (HELICAL) WITH DASH-POT INSIDE
3. BOGIE FRAME
4. SWING LINKS
5. SECONDARY SPRINGS (LAMINATED)
6. BOLSTER
7. CENTRE PIVOT
8. SIDE BEARERS
9. ANCHOR LINKS

Fig.5.10 Suspension arrangement of ICF Laminated bogie.

Particular items and features to be checked

* Surface of centre pivot- for any hindrance to bogie rotation (also check surfaces of side bearers in this context.
* Bolster sway - any heavy indentation marks indicative of sway being excessiva.
* Swing links-whether swivelling freely and not jamming.
* Helical springs-specific deflection, to check possibility of springs getting fully compressed in dynamic condition.
* Level of oil in the dash-pot and condition of dash-pot assembly.
* Lubrication and condition of roller bearing axle boxes.
* Anchor links.



1. WING TYPE ROLLER BEARING AXLE BOX
2. PRIMARY SPRINGS (HELICAL)
3. AXLE GUIDE ROLLER (16 NOS. PER BOGIE I.E. 32 NOS. PER COACH).
4. BOGIE FRAME.
5. SWING LINKS.
6. SPRING PLANK.
7. SECONDARY SPRINGS (HELICAL).
8. BOLSTER.
9. VERTICAL SHOCK ABSORBER (HYDRAULIC).
10. AXLE LINK (OR AXLE HOLDING ARM).
11. CENTRE PIVOT.
12. SIDE BEARERS
13. ANCHOR LINKS.
14. LATERAL SHOCK ABSORBER (HYDRAULIC).

Fig.5.11 BEML Bogie Suspension Arrangement.

Particular items and features to be checked

* Surface of the two side bearers-for any hindrance to bogie rotation.
* Bolster sway - any heavy indentation marks indicative of sway being excessive.
* Swing links-whether swivelling freely and not jamming.
* Helical springs-specific deflection, to check possibility of springs getting fully compressed in dynamic condition.
* Axle guide rollers-whether deficient, having a flat, etc.
* Axle holding arms-longitudinal play and condition of rubber bonded silent block bushes.
* Wheel bases on either side of the same bogie (to detect error in fitment of axle holding arms).
* Lubrication and condition of roller bearing axle boxes.
* Anchor links.

5.6 EMU Stock

The EMU stock running on suburban traffic contains two different type of coaches.

a. Motor coach

b. Trailer coach

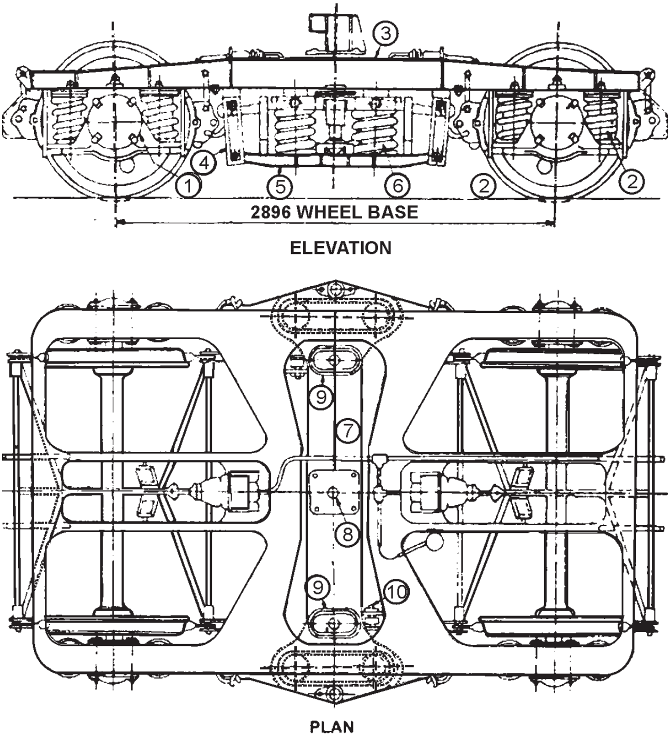
While the trailer coach bogie is identical to ICF All - coil coach bogie, certain modifications had to be carried out in the motor coach bogie for accommodating traction motors. The axle load of the motor coach is, therefore, 20 tonnes while that of the trailer coach is 17.5 tonnes.

5.6.1 Brief description of bogies

Bogies of motor coach and trailer coach (**Fig.** 5.12) are of all welded, light weight construction. Axles for these bogies with their self-aligning spherical roller bearing mounted inside the cast steel axle boxes are rigidly guided by telescopic dashpots and axle guide assemblies. Helical springs working in parallel with dashpots are used for primary suspensions. Coach body is supported on two side bearers located 1700 mm (trailer bogie) and 1200 mm (motor bogie) apart on a floating bogie bolster which in turn rests on two pairs of helical springs supported on a spring plank hung through swing links from bogie frame. The helical springs at each end of bolster of trailer bogies are damped by hydraulic shock absorbers. Side bearer consists of metal slides immersed in oil bath well protected from dust ingress.

No weight is transferred through the bogie pivot which is located in the centre of the bolster. The pivot acts merely as a centre of rotation and serves to transmit acceleration and retardation forces.

The floating bolster in trailer coach bogie is secured in the longitudinal direction to bogie frame by means of two anchor links with silent block bushes, located diagonally opposite to each other and transmit draw and braking forces between bogie



1. Wing Type Roller Bearing Axle Box 2. Primary Springs (helical) with dashpots inside 3. Bogie Frame 4. Swing Links 5. Spring Plank 6. Secondary Springs (helical) 7. Bolster 8. Centre Pivot 9. Side Bearers 10. Anchor Link

Fig. 5.12 Trailer Coach Bogie General Arrangement

frame and coach body through the centre pivot. The motor bogie bolster is located between bogie transoms and transmits draw and braking forces through nylon rubbing plate fixed at the bolster ends.

Rigidly guided axles mounted on self aligning spherical roller bearings, with practically no play in the longitudinal and lateral directions and helical springs working in parallel with dashpots/ shock absorbers of specified characteristics contribute to improved riding index. Therefore, it is essential to maintain these features within the prescribed limits.

5.6.2 Roller Bearings and Axle Box

Both motor and trailer coach bogies have been fitted with double row self-aligning roller bearings. All roller bearings are to be cleaned, inspected and filled with fresh grease at every POH or 2 years or 2 Lakh km, whichever is earlier.

Inspection involves removing the axle box body, cleaning different components, especially bearings, re- lubricating the bearings and re-assembly.

These self-aligning spherical roller bearings are housed in accurately machined cast steel axle boxes. The axle boxes are provided with front and back covers secured by four bolts. To protect bearing from dust and moisture, the axle box housing forms a water-tight assembly.

5.6.3 Springs

The springs provided at the primary and secondary suspension levels are all helical springs. The free height and other specifications of these springs are shown in **Fig.** 5.13 and 5.14 for the trailer coach and motor coach, respectively.

All springs should be scrag tested upto home height unless otherwise specified in the drawing and free height measured. Springs whose free height after scrag test is not within the tolerance shall be rejected.

5.6.4. Dashpots and Axle Guide Assembly

Axle box guides are accurately machined hollow forgings welded to the bogie frame to ensure that the wheel sets are rigidly guided in parallel when assembled. These guides are fitted

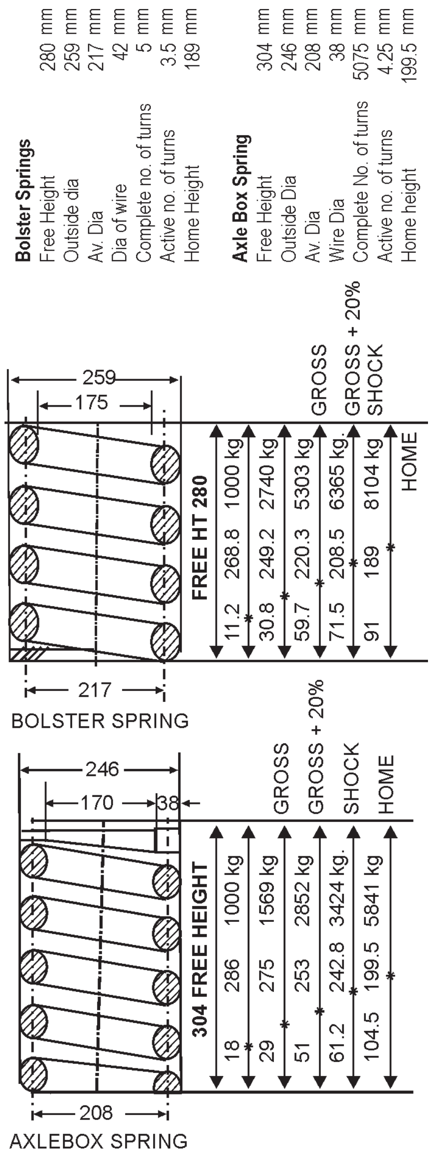


Fig. 5.13 Bolster and Axle Box Spring (Trailor coach)

Bolster Springs	Outer	Inner	Combined
Free Height	293 mm	282 mm	
Outside dia	248 mm	149 mm	
Av. Dia	206 mm	122 mm	
Complete no. of turns	5.5	7.5	
Wire Dia	42 mm	27 mm	
Active no. of turns	4	6	
Home Height	210 mm	189 mm	
Deflection/t	1.083	2.005	0.703
Stress/t	708.1	1580 kg/sq.cm	

Axle Box Spring	
Free Height	290 mm
Outside Dia	251 mm
Av. Dia	213 mm
Wire Dia	213 mm
Complete No. of turns	5.5
Active no. of turns	4
Home height	190 mm
Deflection/t	1.79
Stress/t	988.3kg/sq.cm

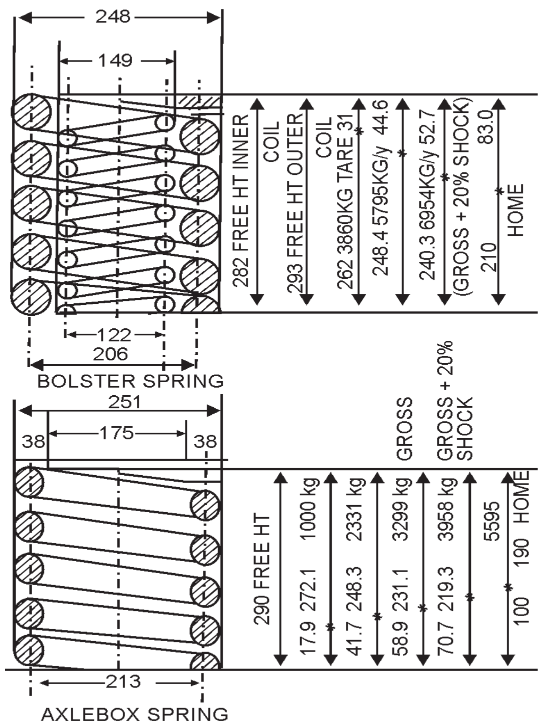


Fig. 5.14 Bolster and Axle Box spring (Motor coach)

The prescribed level of oil in dashpots for the trailer bogie is 77 mm and that for the motor coach bogie is 97 mm, from guide cap inner surface, to be measured when vehicle is empty.

5.6.5 CENTRE PIVOT AND SIDE BEARERS

with bushes at the lower end to give the close guidance of the wheel set both in lateral and longitudinal directions, exactly in the similar fashion as in other ICF coaches.

The centre pivot for the trailer and motor coach is designed to transfer the tractive and braking forces and the vertical load is totally transmitted through the side bearers.

Side bearer consists of the hard-wearing ground steel plate immersed in an oil bath with a floating bronze wearing piece. The oil well is provided with a cover to prevent ingress of dust. The hard ground plate and the spherical bronze wearing piece are likely to wear in service. The permissible wear on the hard ground plates is 1.5 mm and that on the bronze wearing piece is 3 mm. The replacement is resorted to either in case of wear being more than the permissible or in case of observing ridge formations or noticing damages on either of these two.

5.6.6 ANCHOR LINKS

These are provided only in the trailer coaches similar to that in ICF coaches. No coach should be permitted to run with broken anchor links as this is the medium to transmit the draw and braking forces from the body to the bogie and vice-versa.

In case of motor coach bogie, the tractive and braking forces are transmitted from the bogie frame to the bolster by means of Nylon rubber plates fixed to the ends of the bolster. The gap between the nylon plate and the bogie frame shall be between 1 mm to 3 mm.

5.6.7 HANGER AND HANGER BLOCKS

A motor coach can be easily identified by the short hangers at the bolster level. For trailer and the motor coach, hangers and hanger blocks are shown in **Fig.** 5.15. The extent of permissible wear on these components is indicated below into table 5.5 :

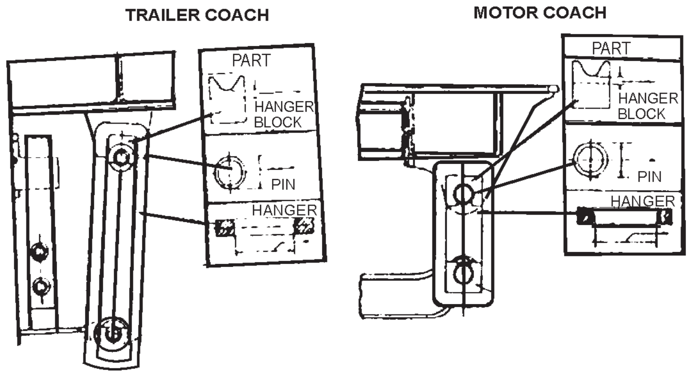


Fig.5.15 Hanger and Hanger block

Table 5.5

Component	Size new (mm)	Condemning (mm)	Wear (mm)	Shop issue size (mm)
Trailer Bogie				
Hanger block	8.0	6.5	1.5	7.0
Pin	34.0	32.5	1.5	33.0
Hanger	354.0	358.0	4.0	356.0
Motor Bogie				
Hanger block	9.5	8.0	1.6	8.5
Pin	45.0	43.5	1.5	44.0
Hanger	246.0	250.0	4.0	248.0

5.6.8 WHEEL AND WHEEL TREAD DIAMETER

The EMU stocks are provided with composite design of wheels consisting of rolled steel wheel centres with renewable tyres. The tyres are fastened to the wheel disc with glut rings and four locking keys to ensure positive securing.

The permissible variation in tread diameter of trailer and motor coach bogies at the time of tyre turning or while replacement shall be as under;

1. Wheels on the same axles	-	0.5 mm
2. Wheels on the same bogie	-	5.0 mm
3. Wheels on the two bogies under the same coach	-	13.0 mm

5.7 HIGH SPEED PASSENGER BOGIES ON IR

ICF all-coil bogie with its 13 t / 16.25 t axle load version has been the main stay for IRs Broad Gauge coaches for the last three decades. With the use of air brakes and improvements in the track structure, speed potential of this bogie has been raised in steps from 110 to 120, 130 and 140 km/h but the bogie has not been found suitable for further higher speeds.

With the thrust on development of technology for speeding up the passenger services initially to 160 km/h and later to 200 km/h, the following bogies were developed/imported:-

5.7.1 IR-20 BOGIE

A completely new bolsterless bogie for operation at 160 km/h and upgradable for higher speeds was designed anddeveloped indigenously. Its bogie frame is suitable for 16.25 t axle load. It is made of corten steel, consisting of two box section dished side frames connected by two cross tubes. The bogie has a wheel base of 2440 mm. Flexi-coil secondary suspension springs are in series with top and bottom rubber pads resting in central housings of side frames. Primary coil springs rest over axle box and go into the end housings of side frames. Transverse beam sitting over the secondary suspension springs carries the centre pivot and forms the interface between bogie and coach body.

890 mm dia. wheels, tapered roller bearings, axle mounted disc brakes, anti-roll bar, bogie rotation stops, transverse resilient stops, noise isolation through an elastic connection between

bogie and coach body and yaw dampers in addition to the vertical dampers in primary stage and vertical and lateral dampers in secondary stage of suspension are the other important design features. These bogies were fitted under IR-Y shell. Later on, their production discontinued due to import of FIAT bogie.

5.7.2 FIAT BOGIE

FIAT bogies with LHB coach were imported from Linke Hoffman Bush of Germany alongwith technology transfer. The bogie is of a Eurofima type construction, consisting of Y shaped longitudinal box section beam connected by two tubular members. The bogie is of two stage suspension. The primary suspension is combined effect of nested helical springs, control arm and rubber elements. The axle guidance between axle to bogie is achieved by control arm assembly. The secondary stage suspension is a combined effect of nested flexi-coil helical spring, rubber elements and rubber bellows. The unique feature of the suspension is the fitment of anti-roll bar, which controls the rolling of the vehicle body without reducing the vertical flexibility and even stiffens the secondary suspension.

The tractive and braking forces from bogie to coach body are transmitted through rocker arm assembly, which swivels around centre pivot. A rectangular shaped mounting frame connected with tubular members are housed with longitudinal and lateral rubber bumps through unique body-bogie connection above secondary suspension through which body weight is transmitted to bogie. The bogie is provided with disc brake system, tapered roller bearing and permanent earthing connections to avoid passage of current through roller bearings. It is also provided with wheel slip protection arrangement.

Figure 5.16 shows a photograph of Fiat bogie. Figure 5.17 shows the General arrangement of the bogie.

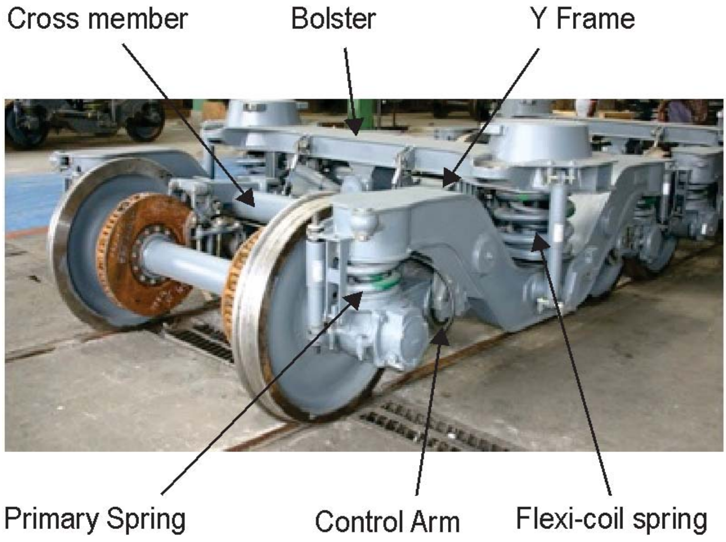


Fig. 5.16 Fiat Bogie – General Arrangement

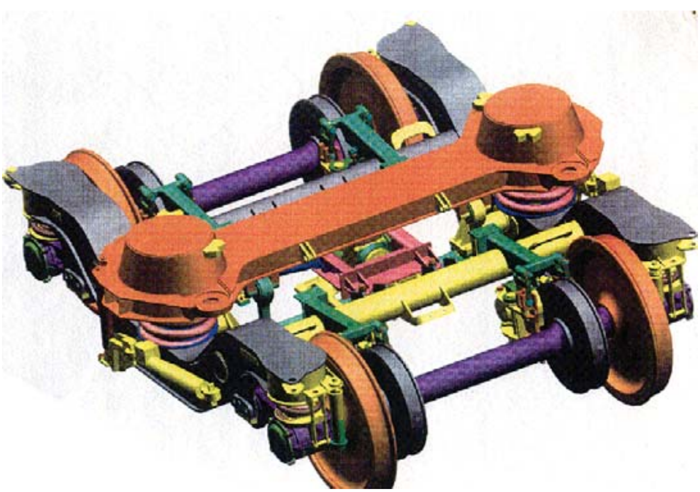


Fig. 5.17 Fiat Bogie – General Arrangement

Some of the important dimensions of the bogie are

Wheel base	2.560 mm
Diameter of new wheels	915 mm
Diameter of max. worn wheel	845 mm
Distance between the wheels	1600 mm
Brake disc diameter	640 mm
Bogie width	3030 mm
Bogie lenght	3534 mm
Bogie weight	6300 Kg

Details of various components of the bogie are discussed below.

The bogie frame is a solid welded frame made by steel sheets and forged or cast parts. The frame is made up of two longitudinal components connected by two cross-beams, which also support the brake units. Various supports which connect the different bogie components are welded to the frame.

Primary suspension comprises of two units of two steel coil springs (internal and external) laid out on the control arm upper part by a centering disk and adjustment shims, (if required). It has a control arm fitted with twin-layer elastic joints, connecting the axle bearing to the bogie frame and transmitting, flexibly, lateral, longitudinal and part of the vertical forces. One vertical hydraulic damper is provided for damping of vertical oscillations. Taper roller cartridge type bearing is used and it makes up a preassembled unit.The axle bearings on the bogie are fitted with sensors for detecting speed, whose signal is analysed by the antislipping system. A photograph of the Primary Suspension arrangement is shown in **Figure 5.18** and its various components are shown in **Figure 5.19**.

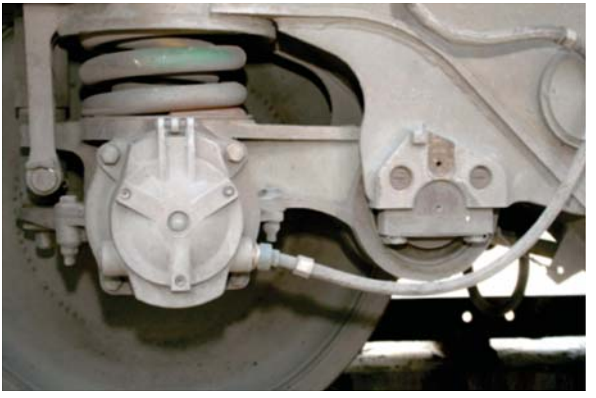


Fig. 5.18 Primary Suspension Arrangement

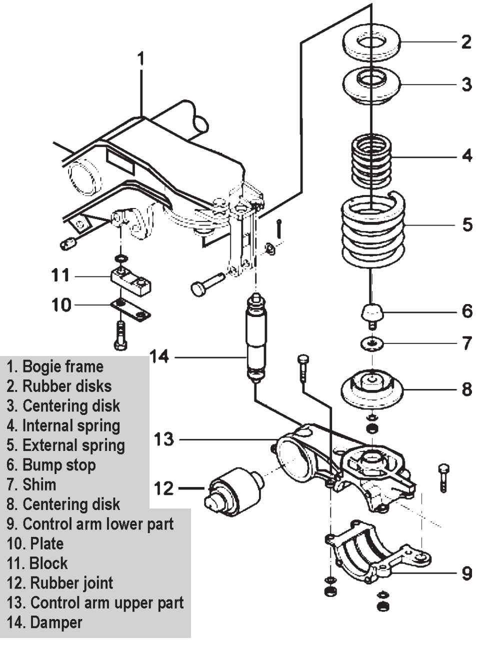


Fig. 5.19 Components of Primary Suspension

Rubber element in Ball Joint Control Arm is to be examined for external condition, crack and detachment during maintenance. If crack length is more than 10 mm, it is to be replaced (**Figure5.20**).

The Secondary Suspension (photograph at **Figure 5.21**) enables lateral and vertical displacements and bogie rotation with respect to body when running through curves. It is achieved by two spring packs which sustain the bolster beam over the bogie frame. Each spring pack is made up by an internal and an external spring, mounted and positioned through the centering discs. The bogie frame is linked to the bolster beam through two vertical dampers, a lateral damper, and the traction rods. The bogie frame is also linked to the carbody through two yaw dampers. Components of Secondary Spring and Dampers are shown in **Figure 5.22** and **Figure 5.24.**

Rubber spring is to be physically examined.Limit of length and depth of crack for vertical and horizontal cracks on the rubber are (10 mm, 4 mm) and (30 mm, 4 mm), respectively. These are to be measured with gauge and depth callipers. Rubber springs with cracks exceeding these limits are to be replaced during maintenance. Also, its load – deflection characteristics is to be checked. Please refer **Figure 5.23**. During Scheduled inspection, physical damage, crack of outer and inner coil spring and damage, crack and aging of upper and lower rubber ring is to be checked.

An anti-roll bar, fitted on the bogie frame **(Figure 5.25),** realizes a constant, reduced inclination coefficient during running. The arrangement results in twisting of the Anti-roll bar due to rolling oscillation of the coach. This arrangement is a special feature of high speed bogies, as it is able to control rolling oscillations without reducing the flexibility of secondary springs.

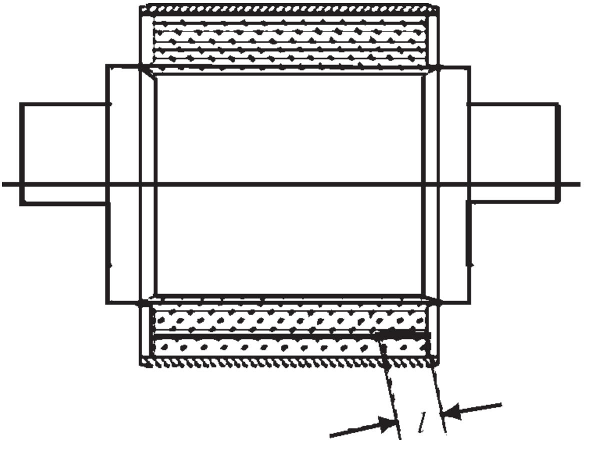


Figure 5.20 Rubber Element Ball Joint Control Arm

Figure 5.21 Secondary Suspension

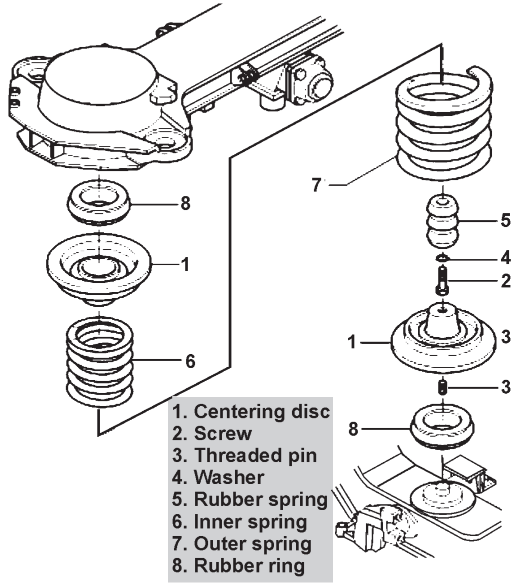


Figure 5.22 Secondary Spring Details

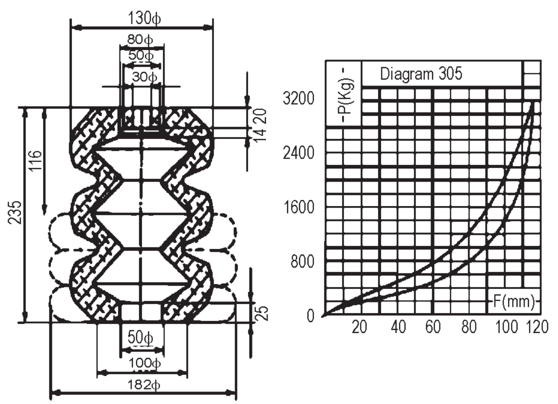
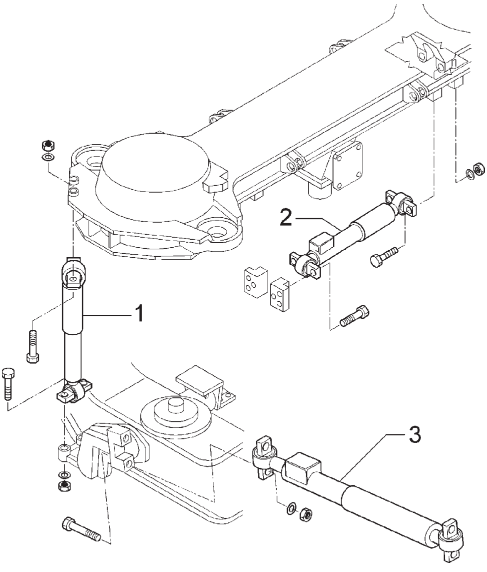


Figure 5.23 Secondary Suspension – Rubber Spring



1. Vertical damper 2. Lateral damper 3. Yaw damper

Figure 5.24 Secondary Suspension – Dampers Arrangement

The Traction Centre **(Figure 5.27)** transmits traction and braking forces between bogie frame and body by a traction lever, on the bolster beam pin, and two rods.The traction lever is connected to the bolster beam by means of a rubber bush, and plates, while rods are connected to the bogie frame and to the traction lever by elastic joints.Traction Centre Rubber Bush and Ball Joint are to be examined for physical condition, crack and detachment during maintenance.

The bogie is provided with a device for limiting the longitudinal and lateral dispacements of the bolster beam, made by four bump stops **(Figure 5.28)**, two longitudinal and two lateral. Rubber elements in the Bump Stops are to be inspected for physical

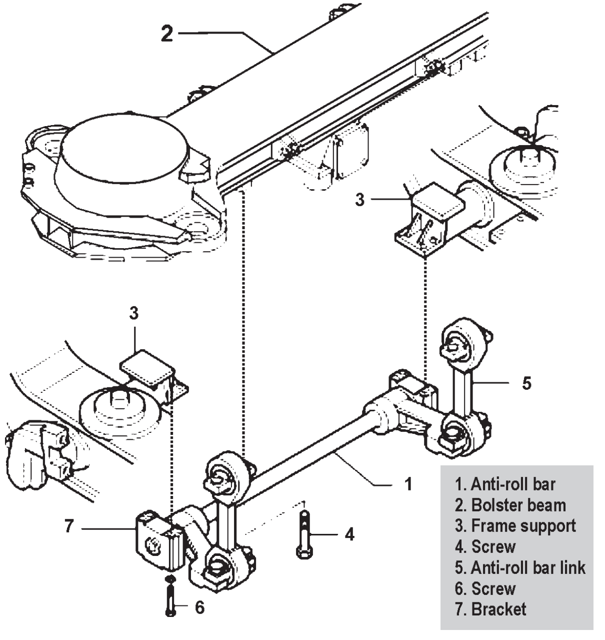


Figure 5.25 Anti – roll Bar Arrangement

damage. During maintenance, Lateral Bump Stop is to be replaced if L becomes less than 30 mm (refer **Figure 5.29**). Its Load – deflection curve must be within specified limits.

A photograph of Anti-roll bar and Traction Centre arrangement in the bogie is shown in **Figure 5.26**

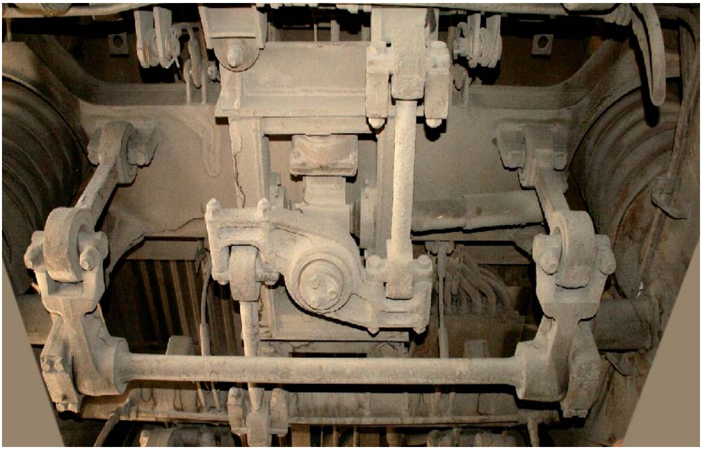


Fig. 5.26 Anti-roll Bar and Traction Centre

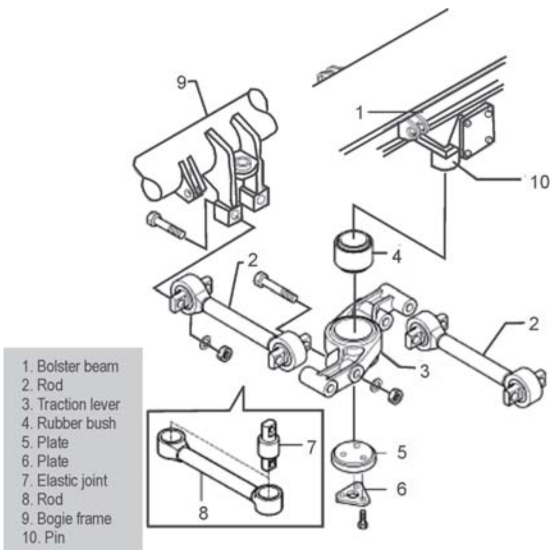


Figure 5.27 Traction Centre

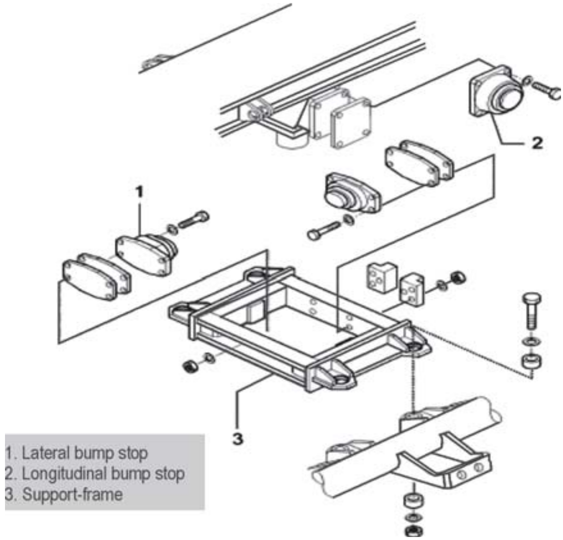


Fig. 5.28 Bump Stop Arrangement

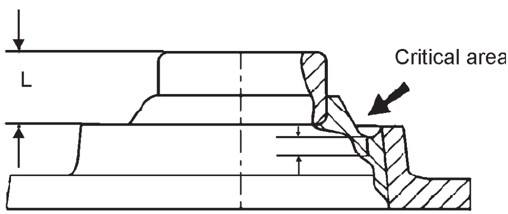


Fig. 5.29 Damage to Lateral Bump Stop

5.8 COMPARAISON OF BOGIES

A comparison among these passenger bogies is given in the following table 5.6

Table 5.6
Comparison among FIAT, ICF all coil & IR-20 bogies

Features	FIAT	ICF all coil	IR-20
Speed Potential	160 km/h	140 km/h	160 km/h
Ride Index (max.)	2.75 at 180 km/h	3.5 at 140 km/h	3.0 at 160 km/h
Weight (t)	6.5	5.72 (13t) 6.5 (16.25t)	6.8
Wheel base (mm)	2560	2896	2440
Wheel dia. (mm)	915	915	890
Axle box guidance	Articulated	Rigid	Articulated
Dampers-Primary	Hydraulic	Dashpot	Hydraulic
Bogie frame	Without headstock	With headstock	Without headstock
Brake	Disc	Conventional	Disc
Bearing	Tapered	Spherical	Tapered

■ ■ ■

CHAPTER 6

LOCOMOTIVES

6.1 GENERAL

The terms diesel or electric, so far as locomotive is concerned, mainly refer to the energy source. In a diesel locomotive the power is generated by a diesel engine, whereas in the case of an electric locomotive, electric power is tapped from over-head equipment.

Transmission of power to axles in the case of diesel locomotive is commonly electric or hydraulic. The locomotive is accordingly called diesel-electric or diesel-hydraulic.

Main line diesel locomotives on Indian Railways are mainly diesel-electric. Diesel-hydraulic locomotives are generally used for shunting only.

In diesel-electric locomotive, the diesel engine produces electric current through a generator. The electric current is transmitted via cables to electric traction motors, which drive the axles.

In the case of electric locomotive, transmission of power to axles is through electric traction motors, which are run by electric current drawn through over head equipments.

On the Indian Railways, axle-hung nose suspended type of traction motors, powering individually each axle (i.e. one traction motor for each axle) have been largely adopted for high speed diesel and electric locomotives . In this type of traction motor, the motor casing covers the axle as well (like a bearing), and thus is hung from the axle at one end while the other end (nose) is supported on the bogie frame. Gear of the motor meshes with gear on the axle (**Fig.** 6.1).

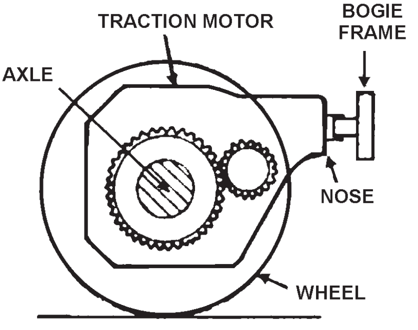


Fig. 6.1 Axle hung nose suspended traction motor

6.1.1 DESIGNATION OF LOCOMOTIVES

There are two systems of designation of locomotives.

These are based on

- Gauge, power source and service
- Wheel and traction arrangement

6.1.1.1 Gauge, Power source and Service

Type	Symbols used are
Gauge	
BG	W
MG	Y
NG	Z
Power source	
Diesel	D
Electric,dc	C
Electric,ac	A
Electric, dc/ac	CA
Service	
Passenger traffic	P
Mixed traffic	M
Goods traffic	G
Shunting	S

For instance, WDM stands for B.G., Diesel, Mixed traffic locomotive. The designation also carries a suffix which indicates a particular design in the relevant group, e.g.

WDM2, WDM4, WCG2 and so on.

6.1.1.2 Wheel and traction arrangement

Numerals stand for number of carrying axles in each bogie i.e. axles, the function of which is only to carry a part of the locomotive superstructure weight and not to develop tractive effort. Alphabets (capital) stand for number of axles (in each bogie) which develop tractive effort, e.g.

A =1; B = 2; C = 3

Individually driven axle are suffixed ‘o’.

Some examples of designations from consideration of wheel and traction arrangement are shown in **Fig.** 6.2.

The high speed diesel and electric locomotives on the Indian Railways are mainly of Bo-Bo or Co-Co arrangement.

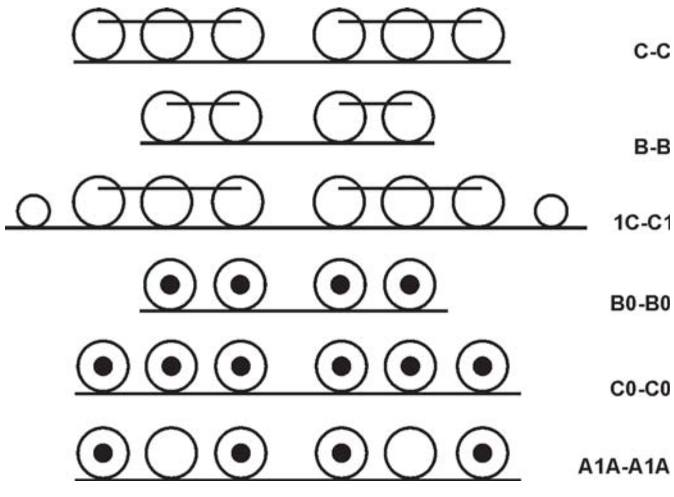


Fig. 6.2 Axle and Traction Arrangement

6.1.1.3 Locomotives and bogie type

There are various type of locomotives on the Indian Railways. They may be functionally different or their features, such as bogie design, number and type of traction motors etc. may be different. Study of the features and critical maintenance parameters of locomotive bogies is of prime interest from the point of view of derailmant.

List of bogies and the locomotives on which they are provided, for the common locomotives on the Indian Railways is given in Table 6.1.

Table 6.1
Bogie type for different locomotives

Sl. No.	Bogie Type	Locomotive	Remarks
1.	CO-CO Tri-mount Bogie	WDM2	Max speed 120 kmph (WDM2 (modified)), 2600/2400 HP. Many WDM2 have been converted to WDM3A
		WDM3A	Max speed 120 kmph
		WAM4	Max speed 110, 3600 HP, kmph,WAM 4P (being heavily used) and 4B were regeared versions for passenger / goods
		WAG5	Max speed 80kmph,3850HP. Variants of WAG-5: WAG-5A (Alstom motor), WAG-5H (Hitachi motor), WAG-5P (fast passenger traffic), Variants used for mixed operation
2.	High Adhesion Bogie	WAG7	Max speed 100 kmph, 5000 HP
		WDG3A	Max speed 100kmph, 3100 HP
		WDM3D	Max speed 120 kmph, 3300 HP
3.	Bo-Bo mitsubishi	WAM2/ WAM3	Imported (in 1960s), condemned after completion of their codal life.
4.	Flexi coil Mark I Cast bogie	WDM4	CO-CO bogie (GM design), Max speed 130 kmph,, Was run on Howrah Rajdhani. Since phased out
		WAP1	CO-CO bogie, WAP1-HE (Hitachi) - 3900 HP, Max speed 140 kmph

		WAP4	WAP1 (5000 HP) was renamed WAP4
5.	Flexi coil Mark IV bogie	WAP3	CO-CO fabricated bogie. All WAP3 have been converted to WAP4
		WAP6	CO-CO fabricated bogie. All WAP6 have been converted to WAP4
6.	Bo-Bo fabricated bogie (ABB)	WAP5	5500 HP, Max speed 160 kmph, upgradable to 220 kmph. Imported from ABB and now being manufactured at CLW with TOT.
7.	HTSC bogie for GM EMD Loco	WDG4	4000 HP, Max speed 105kmph. Imported from General Motors and now manufactured at DLW
		WDP4	4000 HP, Max speed 160 kmph, (designed for 180 kmph)
8.	Co-Co Flexi coil fabricated bogie (ABB)	WAG9	6000 HP, Max speed 100 kmph. 3-phase loco imported from Bombardier and now manufactured at CLW with TOT
		WAP7	6000 HP, Max speed 140 kmph. Same design as WAG9, with changed gear ratio.

6.1.2 Wheel wear and General

Important items of the Instruction Bulletin on 'Wheel Wear and Application Management' No. MP.IB.BD.02.16.01 (Rev.01) dt. 31.12.2009 are as follows

(i) Brake cylinder pressure is to be maintained as stipulated to avoid excessive thermal loading of wheels due to repeated application.

(ii) Dynamic brake is provided on all locomotives on IR (except for WAP4). They should be used, specially while operating in a ghat section, Use of loco brake should be limited to the minimum.

(iii) Matching of spring based on free height parameter is not adequate. They should be matched as per height under working load. Springs should be colour coded as per instructions of the specified Instruction Bulletins.

(iv) Pins and bushes of the Equaliser beam in Trimount/ High Adhesion bopgie should be of recommended design and these should be renewed as specified in the Maintenance Instruction. Otherwise, inter-axle load transfer gets affected.

(v) Locomotives having wheel guidance through pedestal are provided with liners at pedestal as well as on axle box. Lateral and longitudinal clearances to be maintained and the corresponding reference documents are specified in the Instruction. Proper record of these clearances is to be kept for all specified schedules.

(vi) Wear adapted profile as per RDSO Drg. No. SK.DL-2561 (Alt.8) shall be applicable to all BG Diesel& Electric locomotives irrespective of speed or IRS Thick flange profile as per CSL-3040 would be applicable beyond 110 kmph (**Fig.** 6.3 and 6.4).

(vii) Correct cause of biased wear on a wheel should be identified considering Brake block shifting, Excessive lateral or less Longitudinal clearance between axle box and pedestal, Mis-alignment of bogie frame and excessive wheel diameter variation.

(viii) Variation in diagonal dimensions of bogie should not be more than 3 mm in order to prevent mis-alignment.

(ix) General limit of variation in wheel diameter is specified as below *(Note- Any other specific instruction, if issued, may be followed)*

Location	At wheel turning/ change (mm)	Service limit (mm)
On same axle	0.5	2.5
On same bogie	2.0	8.0
On same loco	15.0	25.0

The Instruction includes limits of wheel diameter for various locomotives.

(x) Limits of root, flange and tread wear are given for various locomotives.

(xi) Permissible variation in Wheel gauge is as below

New assembly : 1596 (+- 0.5) mm

Service limit : 1596 (+3, -0.5) mm

(xii) Recommended time for application and release of Full service and Emergency brake application, brake pressure and suitable brake application practice are specified for reducing wheel skidding.

Limits of Wheel diameter, Variation in wheel diameter, Wear limits of Root, Flange and Tread and Lateral and Longitudinal clearances at the axle box for different locomotives, as specified in the Maintenance Instruction, are reproduced as **Annexure- I** to this chapter.

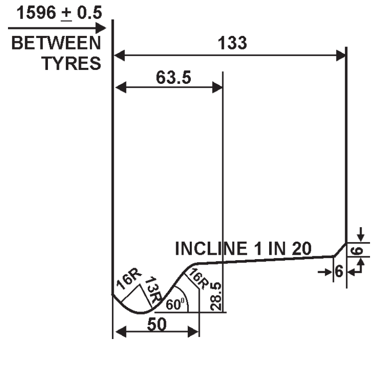


Fig. 6.3 Thick IRS Tyre Profiles for BG Loco

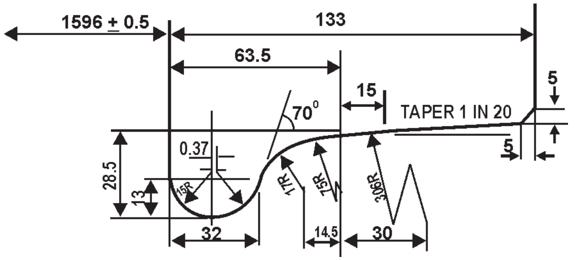


Fig. 6.4. Wheel Tread Profile (Wear adopted) for BG loco

6.2 LOCOMOTIVES

Bogie of common locomotives on the Indian Railways are discussed in the following paras.

6.2.1 CO – CO Tri-mount bogie

6.2.1.1 WDM2 (RDSO Report No.M.195)

This diesel-electric locomotive with Co-Co bogies has been designed for mixed traffic service with a rated horse power of 2600/2400. The transmission system consists of six axle-hung nose-suspended traction motors, one for each axle, and the motor torque is transmitted to the wheels through reduction gears. The design is adapted from ALCO (American Locomotive Company, U.S.A.)

The general arrangement of the bogie is shown in **Fig.** 6.5 (a) and (b). The bogie which is of ALCO special tri-mount type, has a centre pivot and two side bearers which form a three point support for the superstructure weight. The pivot carries 60% of the vertical load and also transmits tractive and braking forces. The two side bearers working in oil bath carry the remaining 40% of the vertical load equally distributed. The bogies are so oriented that the side bearers are located towards the centre of the locomotive.

The tri-mount bogie design incorporates a single stage suspension. The spring system consists of four groups of two nests of springs, each nest consisting of one outer and one inner

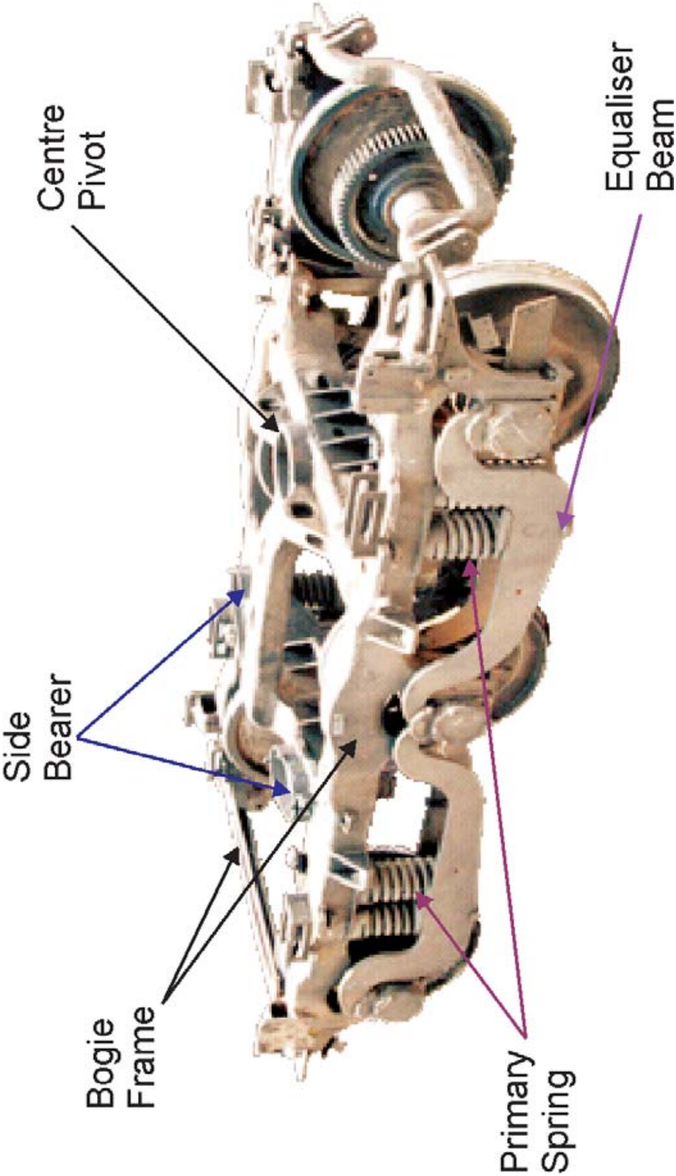


Fig. 6.5(a) WDM2 Locomotive

1. Roller bearing axle boxes.
2. Equalising beams.
3. Main bearing springs (helical).
4. Bogie frame-cum-bolster.
5. Horn-guides or bogie pedestals.
6. Centre pivot.
7. Side bearers.
8. Spring loaded friction snubbers (working inside main springs).

Items and features to be checked for defects during deraliment investigation :

General * Defects in wheelsets, axle box assembly, horn guide axle box assembly, spring gear. buffing gear and brake gear. * Clearances and play at axle box level.

Particular * Surfaces of the centre pivot and the two side bearers-for any hindrance to bogie rotation. * Spring loaded friction snubbers.* Equalising beams whether swivelling freely at their points of support axle boxes.

Additional for WDM_2 (modified) locomotive bogie :

* Resilient thrust units in the axle boxes of outer axles of each bogie.

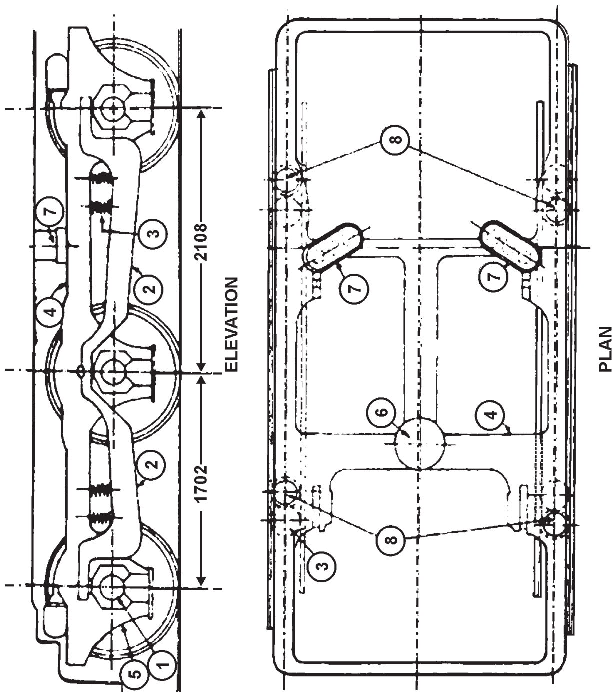


Fig. 6.5(b) Suspension Arrangement WDM2 Locomotive

Fig. 6.6 Spring Loaded Friction Snubber

helical coil. The spring nests are mounted on equalizer beams which in turn rest on axle boxes.The cast steel bogie frame containing axle-hung nose suspended traction motors and gearing rests on the spring system.

Equalising beam arrangement is such that the vertical load is equally transferred to all the three axle. For achieving this, the beam should get supported at the correct location on the manganese steel liner welded on top of the axle box.

Damping is provided by four units of spring loaded friction snubbers working in conjunction with one of the inner coils in each spring group. The snubber assembly consists of 3 alloy steel friction shoes working up and down in a case hardened sleeve (**Fig.** 6.6). The shoes are pressed against the sleeve by 3 precompressed springs equally spaced. Rubbing up and down, the sleeve provides a constant force of about 500 kg per snubber unit. Principal design particulars of the locomotives are given in **Table** 6.2 :

Table 6.2

Principal design particulars of WDM_2 locomotive	
Horse power	2600/2400
Wheel arrangement	Co-Co
Wheel base	
a) Total locomotive	12633 mm
b) Bogie	3810 mm
Wheel diameter	1092 mm
Tyre Profile	Wear adopted
Maximum overall dimensions	
a) Height	4185 mm
b) Width	3010 mm
c) Length overall	17119 mm
d) Length over headstocks	15849 mm
Driving arrangement	One traction motor per axle
Axle load	18.8 t

The axle boxes are guided vertically and laterally between pedestals cast integral with the bogie frame. To provide negotiability over curves and turnouts transverse play has been provided between the axle boxes and the bogie frame.

A transverse play ranging from 3.175 mm minimum to 6.35 mm maximum is available between axle boxes and hornguides in case of outer axles. Transverse play between the centre axle boxes and the hornguides ranges from 12.7 mm minimum to 15.8 mm maximum. The total longitudinal clerance is 1.59 mm in all cases, the maximum permissible being 4.76 mm. With Timken roller bearings fitted on the journals, there is no lateral play between the axle and axle boxes.

The wheel profile is wear adopted.

This locomotive has been tested and found stable upto 115 km/h, and has been cleared for a maximum operational speed of 110 km/h.

6.2.1.1.1 DEFECTS

Apart from the defects discussed for rolling stocks in general, following need particular mention :

i) The surfaces of the centre pivot as well as of the two side bearers should be checked to see if there is any tendency for causing hindrance to bogie rotation. Crack in the oil bath leading to leakage of oil and lack of lubrication in the centre Pivot are common defects.

ii) The condition of the spring loaded friction snubbers should be checked to see whether damping is adequate or not.

iii) The equalizing beams should swivel freely at their points of support on the axle boxes, otherwise off-loading of one wheel will not be shared by the adjacent wheel or wheels. It can be checked by rotating the roller pins by hand, which should get rotated freely in position.

6.2.1.2 WDM2 (modified)

Beyond 115 km/h, WDM2 locomotive exhibits unsatisfactory riding characteristics in the lateral mode. To increase the speed potential of this locomotive, certain modifications were carried out and the modified version is called WDM2 (modified). The main modifications are :

(i) provision of resilient thrust units at the axle boxes of outer axles of each bogie. This comprises an annular rubber cone which when fitted has a precompression of 2 mm equivalent to 1 t force. This resilient restraint improves lateral stability on the straight as well as negotiation around curves and turnouts.

(ii) The middle axle has been given a free float (lateral freedom) of 10 mm either way, enabling the locomotive to negotiate curves of 174 m radius in conjunction with the 5 mm permissible compression of the rubber cones in the outer axles.

(iii) Lateral clearances between the bogie pedestals and axle boxes have been reduced to 1.5 mm each way.

With above modifications, the WDM2 (modified) has been cleared for a maximum operational speed of 120 km/h (having been tested and found stable upto 140 km/h).

It will be obvious, that when the rubber cone in the resilient thrust unit wears out, the beneficial effect would be lost and the locomotive would behave as if it were WDM2 only for which the maximum speed is only 110 km/h. In WDM2 (modified), therefore, the resilient thrust unit should particularly be checked in case the locomotive was scheduled for running at maximum speed of 120 km/h.

6.2.1.3 WAM4 (RDSO Report No.336)

WAM4 type ac electric locomotive, designed for operation on 25 kV ac single phase 50 Hz overhead lines, has a rated horse power of 3600. The bogie and the suspension details are identical to those of the WDM2 locomotive.

Nominal and maximum limits of transverse play between axle boxes and horn guides for different axles are given below:

Axles	Nominal	Maximum
Outer axles	6 mm	12 mm
Middle axles	25 mm	31 mm

No axle play has been provided between the journal and the bearings. Total longitudinal clearance per pedestal is 1.5 mm nominal, maximum permissible being 5 mm. Salient particulars of this locomotive are given in Table 6.3.

Table 6.3

Design particular of WAM4 locomotive	
Horse power	3600
Wheel arrangement	Co-Co
Wheel base	
a) Total	14898 mm
b) Bogie	3810 mm
Wheel diameter (new)	1092 mm
Tyre profile	Wear adopted
Maximum overall dimensions	
a) Height	4162 mm
b) Width over body panels	3055 mm
c) Length over buffers	19974 mm
Axle load	18.8 t

This locomotive has been cleared for a maximum operational speed of 110 km/h. However, some of the Railways have reported hunting tendencies in this locomotive at speeds around 80-90 km/h, particularly when coasting down a gradient. Improvement in running has been reported by removal of the traction motors links which were provided between the traction motors and the bogie frame.

The defects to be checked for, during derailment investigation would be the same as those in case of WDM2 bogies.

6.2.1.4 YDM4 and YDM4A (RDSO Report No. M-317 and C-138)

The wheel arrangement and suspension system of these locomotives is more or less similar to that of WDM2.

6.2.1.4.1 YDM4

The YDM4 diesel-electric Co-Co type locomotive (design of M/s. ALCO Products Inc. U.S.A.) is an M.G. mixed traffic service loco, with a rated horse power of 1400. Power transmission is through six axle-hung nose-suspended traction motors, one for each axle.

The locomotive has a single stage verticalsuspension with helical springs. Each bogie has a centre pivot and two side bearers which form a three point support (tri-mount) for the superstructure weight. The bogie pivot carries 64% of the vertical load and also transmits tractive and braking forces. The remaining load is shared equally by the two side bearers. Damping is provided by a set of friction snubbers in each spring group. Each spring cluster, consisting of two spring group, is provided with one snubbers spring group.

6.2.1.4.2 YDM4A

This locomotive (design of M/s. Montreal Locomotive Works, Canada), is identical to the YDM4, in all principal features, except in regard to the transfer of load of the superstructure to the bogie.

The body weight is transferred to each bogie through three load bearing pads. Two of the pads are located on either side of the bogie pivot while the third pad is located along the longitudinal centre line of the bogie frame towards the inside axle of the bogie. The two load pads on the sides are each designed to carry 33 per cent of the total weight over the bogie and the inner pad carries the remaining 34 percent. The bogie pivot is not designed to carry any vertical load but acts only as a centre of rotation for the bogie and also transmits the tractive and braking forces.

These locomotives have been cleared for a maximum operational speed of 100 km/h (having been tested and found stable upto 110 km/h.)

(N.B. : On M.G. for speeds above 75 km/h, track has to be maintained to tolerances laid down in RDSOs Report No.C-138).

Same defects should be looked for, as in case of WDM2 locomotive, except that the surfaces to be checked for any hindrance to bogie rotation would be those of

- the centre pivot and two side bearers in the case of YDM4 locomotive;
- the three load pads in the case of YDM4A locomotive.

6.2.2 High Adhesion Bogie

6.2.2.1 WAG7

WAG7 is a 5000 HP, 100 kmph locomotive provided with High Adhesion Fabricated CO-CO bogie. These locomotives have been a work horse for freight traffic on the electrified sections. The bogie achieves a higher tractive effort through better adhesion charecteristics with reduced weight transfer.

This is a 3-axle type bolsterless bogie with two stage suspension, floating pivot and uni-directional arrangement of axle hung, nose suspended traction motors. Bogie frame is of straight and fabricated box type construction with three transoms to carry nose suspension. The general arrangement of bogie is shown in Figure 6.7.

The locomotive body is supported on bogie frame through four rubber side bearers. Shims have been provided below outer side bearers to distribute load on side bearers in the ratio 60:40. Centre Pivot does not take any vertical load and is used only for transfer of tractive/ braking forces.

Bogie frame is in turn supported on helical coil springs mounted on equaliser beams. The equalising mechanism consists of equalisers hung directly on end axle boxes and supported on middle axle box through a link and compensating beam arrangement. The equalising arrangement enables achievement of equal axle loads and reduces weight transfer at start. A typical photograph of the Primary suspension arrangement of High Adhesion Bogie is shown in Figure 6.8.

The bogie has two stage suspension with helical coil at primary and rubber sandwitch (side bearers) at secondary stage. Lateral stiffness of the rubber springs is utilised to provide lateral guidance at the secondary stage. The total static deflection (vertical) is 119 mm, 102 mm primary and 17 mm secondary.

Four vertical hydraulic dampers of 750 kg at 10 cm/sec capacity are provided, one with each nest of primary springs. Two lateral hydraulic dampers are provided, capacity 1150 kg at 10 cm/sec, at secondary stage, to supplement the damping provided by side bearers in lateral and yaw mode.

Lateral spacing of side bearers provides stability against tipping forces on curves and damping provided by them prevents nosing at high speed.

Four lateral rubber stops are provided on each bogie on either side of middle axle to limit lateral movement between bogie and locomotive underframe. Vertical stops are provided on bogie frame to limit vertical movement between axle boxes and bogie frame.

Some important design and maintenance data are as below

i) Axle load: 20.50 ton

ii) Length over buffers – 20394 mm

iii) Bogie wheel base – 3800 mm

iv) Height of side buffers from rail level -1080 - 1105 mm

v) Primary spring – Total deflection 102 mm, Stiffness – Outer (40.80 kg/mm), Inner (15.46 kg/mm)

vi) Secondary spring – Outer- deflection 17 mm, Inner – deflection 12 mm

vii) Wheel tread diameter – 1092 mm (nominal), 1016 mm (service limit)

viii) Difference in wheel tread diameter (service limit) – On same axle 2.5 mm, on same bogie 8.0 mm, on same loco 25.0 mm

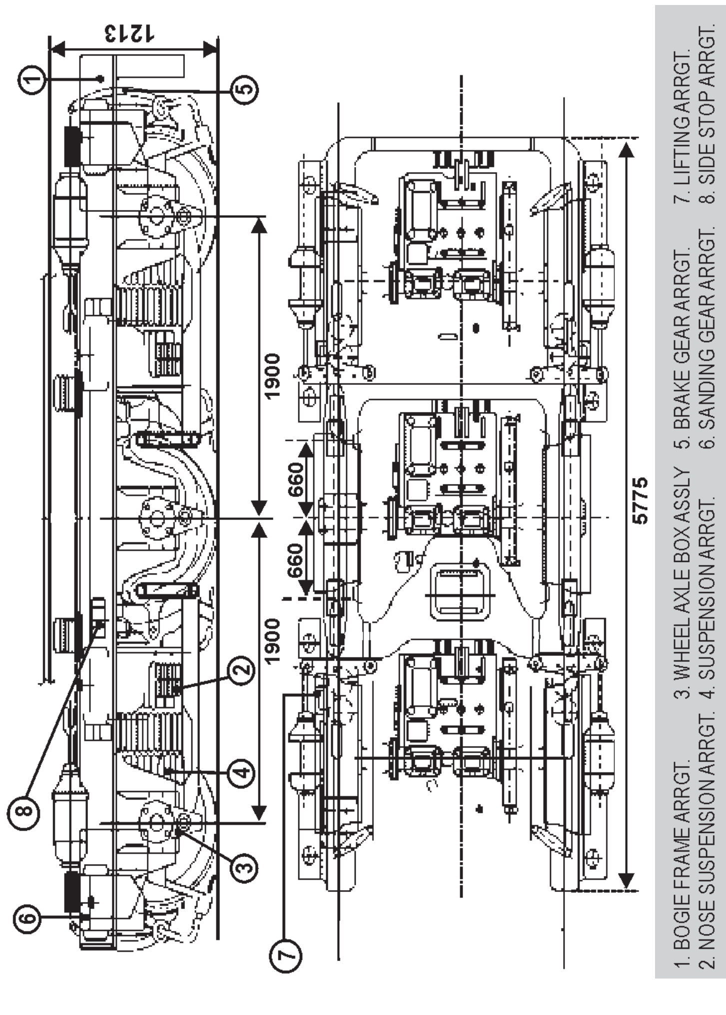


Fig. 6.7 WAG7 bogie: General Arrangement

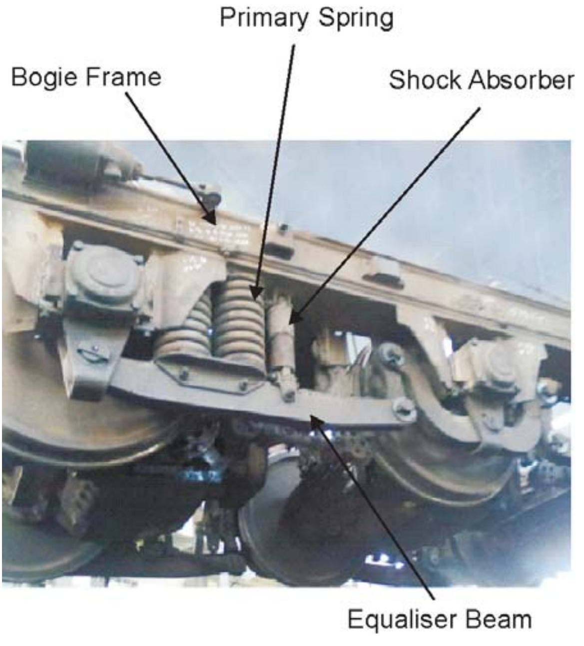


Fig. 6.8 High Adhesion Bogie: Primary suspension

ix) Pedestal clearance (service limit) – Lateral clearance: End axles 9.5 mm, middle axles 11.5 mm; Longitudinal clearance: 6.0 mm

x) Centre pivot housing (service limit) – 366 (+-) 1 mm

xi) Centre pivot pin (service limit) – 360 +1/ -0 mm

WDG3A locomotive (old nomenclature WDG2 class) is provided with High Adhesion Bogie, having the same arrangement.

6.2.2.2 WDM3D

WDM3D is a 3300 HP Mixed traffic Diesel locomotive, with maximum permissible speed of 120 kmph. It is provided with

High Adhesion Fabricated CO-CO bogie, similar to that of WAG7.

The bogie has two stage suspension with helical coil at primary and rubber sandwitch (side bearers) at secondary stage. Inner spring of Primary stage has a stiffness of 15.46 kg/ mm, Free height of 512 mm (permissible variation + 9 mm) and working height of 402 mm (permissible variation + 4 mm, - 6 mm and working load 1701 kg). The springs are to be grouped and colour coded as A (white) and B (red) based on the working height range of 396 – 401.5 mm and 402 – 406 mm, respectively. Outer spring of Primary stage has a stiffness of 40.8 kg/ mm, Free height of 552 mm (permissible variation + 9 mm) and working height of 447 mm (permissible variation + 4 mm, - 6 mm and working load 4288 kg). The springs are to be grouped and colour coded as A (white) and B (red) based on the working height range of 441 - 446.5 mm and 447 – 451 mm, respectively. Spring of same working height (group) are to be used on the same bogie.

Height of the Side bearers is 165 mm. Its defects are Crack of 50 mm length or more on rubber surface, Bond failure of 40 mm length or more, Crushing or Crumbling of rubber and Permanent set of 4 mm or more. It has a service life of 3 years.

Height of Side Buffer is to be maintained between 1105 and 1080 mm. Shims are provided at Primary Spring seat to ensure this height. Whenever shims are used, height at all spring seats is to be checked to ensure equal load distribution on springs.

Service limit for Wheel gauge is 1596 (+3, -0.5) mm

Service limit for Centre pivot housing is 369 +1/ -0 mm and Centre pivot pin is 358 +1/ -0 mm.

With increasing demand of mixed traffic on Indian Railways, a new high adhesion high speed bogie (HAHS) has been developed by RDSO for high horse power locomotives to achieve higher tractive effort at start. This bogie has been provided on a modified version of WDM3D and WDM3E and WDM3F locomotives. The Equaliser beam has been eliminated in this bogie design. Transfer of vertical load in this modified design of bogie takes place directly from bogie frame to axle box (wing type) through a pair of helical springs located thereon. Primary suspension arrangement of the bogie is shown in the photograph at Figure 6.9.

Fig. 6.9 Modified WDM3D Bogie

These locomotives are fitted with cylindrical roller bearings. Movable axle journal boxes are mounted between pedestals or horns. Lateral play for negotiating curves and turn-outs is obtained by the movement of axle boxes in horns. End axle boxes are provided with rubber thrust pads to cushion lateral thrust, while 10 mm lateral play is provided on middle axle boxes.

In order to reduce the lateral forces between the rail-wheel in locomotive bogies at high speed, axial resilient thrust pad is incorporated in Axle Box of end axles of the bogie. The axle boxes of central axles of the bogie do not have Axial Resilient Thrust Units. Arrangement of end axle box is shown in **Fig. 6.10.**

The lateral thrust passes through the conical rubber thrust pad (C) held between inner and outer thrust collars (A) and (B).

The centre axles are not provided with resilient thrust units. Adequate clearance has been provided in the bearing of these axle boxes to negotiate the sharpest curve. In actual service, the axle will be more or less ‘floating’ within 10mm lateral clearances.

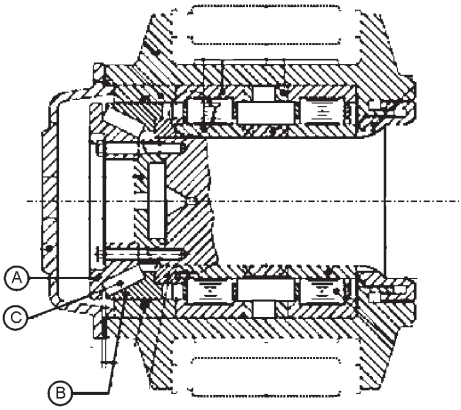


Fig. 6.10 End Axle box- With Resilient Thrust Unit

Primary suspension springs are to be checked for Free height and Working height at specified load. Permissible limits and colour coding scheme is as below

Free height – maximum 543.8 mm, condemning 531.8 mm

Working height – Working load 3942 kg, nominal ht 441 mm, maximum ht 449 mm, condemning ht 433 mm

Colour code – Brown 444 – 449 mm, Blue – 438 – 444 mm, Green – 433 – 438 mm

Springs of the same group is to be used on a bogie. Springs with green colour code are to be used with shims such that their static height is in the range of Blue colour springs.

Side bearers (rubber spring)is to be checked for cracks or damages and permanent set. Side bearer shall be replaced, if It has completed a service life of 4 years, any of the metal plates show cracks, cracks of 50 mm length or more on the rubber surface, bond failures of 40 mm length or more, crushing or crumbling of rubber and a permanent set of 10 mm or more.

6.2.3 Flexi-coil Bogie Mark-I

6.2.3.1 WAP1/ WAP4

Flexi-coil Mark-I bogie was designed by RDSO for opearation of Rajdhani Express on electrified section, which were being hauled by imported WDM4 locomotive on diesel section. Design principles of this bogie are same as that of WDM4 loco bogie (General Motor design). WAP1 was inducted for hauling Rajdhani Express on NDLS – HWH section in 1981, at a speed of 130 kmph. Later on, the locomotive was cleared to operate at 140 kmph.

A higher tractive effort version of the locomotive has since been developed (5000 HP) and the loco has been renamed WAP4.

The bogie frame and bogie bolster of "FLEXICOIL" bogie Mark-I are of steel casting box type construction. The locomotive body weight is transferred to the bolster through a centre pivot. The bowl is fitted with phenolic oil lubricated vertical and horizontal liners which provides rotational freedom between body and bogie in operation.

The cast steel "H" type bolster is supported on cast steel bogie frame at four corners, by pair of helical springs placed in spring pockets of main longitudinal member of the bogie frame. The bolster is located with respect to bogie frame by upright pedestals which are an integral part of the bogie frame. This arrangement serves to transmit force from bolster to the bogie frame and vice versa. Spring loaded snubbing piston (2 nos. per bogie) made of phenolic material are provided to have high friction between bolster and bogie frame for damping in both vertical and lateral modes of oscillation. Lateral stops are also provided on the bolster as well as on the bogie frame to limit the side movement by flexing action of the springs, which is of the order of 32mm. General arrangement of the bogie has been shown in **Fig.** 6.11 (a) and (b).

Flexicoil bogie Mark-I has two stage of [vertical] suspension in which helical springs have been used at both primary and secondary stages. Loco flexibility on the primary and secondary stages are 0.75mm/t and 1.13mm/t, respectively. Total static deflection is 150 mm, comprising 66.0mm in primary and 84.0 mm secondary. The transverse flexibility between the body and the bogie has been achieved by flexicoil action of the helical spring at the secondary stage. The support of the bolster springs have been placed on wider arm to give better stability in rolling. Shims of various thickness, already provided on the bogie frame and bolster below the spring seats of primary and secondary suspension, are to be retained and ensured during assembly since they are matched to get equal working heights of springs after calibrations.

The bolster spring friction device consists of a phenolic piston, steel washer and a spring contained within a cylindrical housing in the bolster. A suitable fixture and a jack may be used to apply or remove the friction piston. After the piston is installed, a safety stop is welded in position **(Figure 6.12).** After continued use, the wear on the friction piston will reduce its length and lessen the effect of the spring on the piston. The piston can be continued in use until the piston length 70mm is reached. This dimension will allow 16mm wear on the piston. However, when the piston reaches a dimension of 80mm or 6mm wear, a compensating washer 6mm thick is added to the standard 10mm washer.

In place of conventional pivot liner, “phenolic” synthetic liners are used in the Centre Pivot of this bogie. The load on these parts and the relative movement between them cause the parts to wear. Outside diameter of the underframe pivot assembly and inside diameter of the bolster bowl should be checked to determine the total clearance between these parts. Condemning limit of this clearance is 6.0 mm. Lubricant is applied to the outside diameter of the vertical liners and then the liners are fit in the bowl 90 deg from the longitudinal center line of the locomotive. The re-used and condemning limits of these liners are shown in SK.MI-90/80/032. Condemning limit of thickness of horizontal liner is 6.5 mm. Phenolic liners are oil lubricated. Oil level to be maintained is 10mm above the horizontal liner top surface (**Fig.** 6.13).

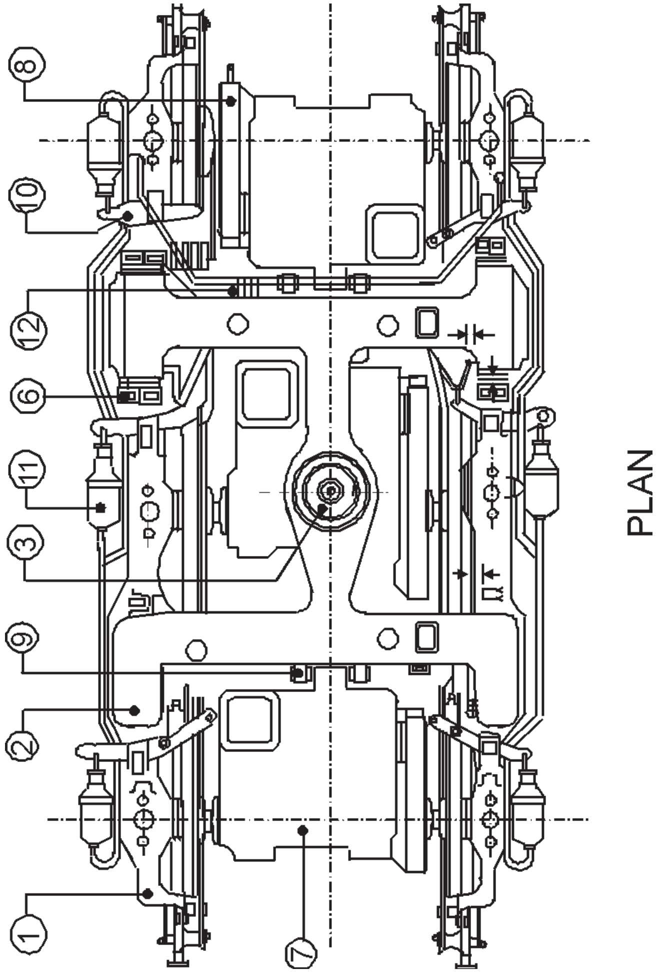


Fig. 6.11 (a) Flexi-coil Bogie Mark-I : General Arrangement

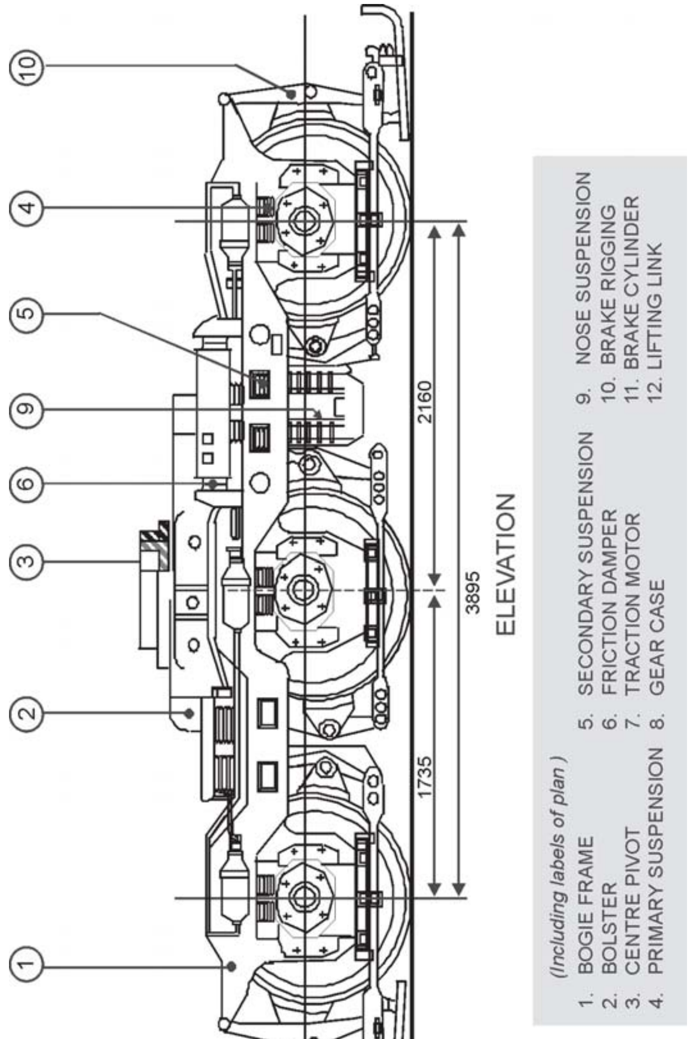
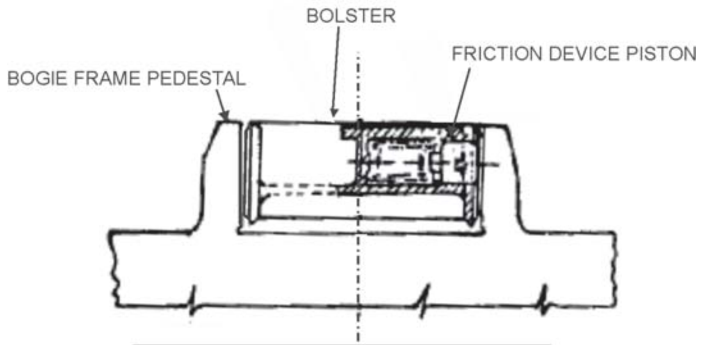


Fig. 6.11 (b) Flexi-coil Bogie Mark-I : General Arrangement



Clearance limit	Total (mm)
New	3.2
Condemning	9.6

Fig. 6.12 Bolster Spring Friction Device

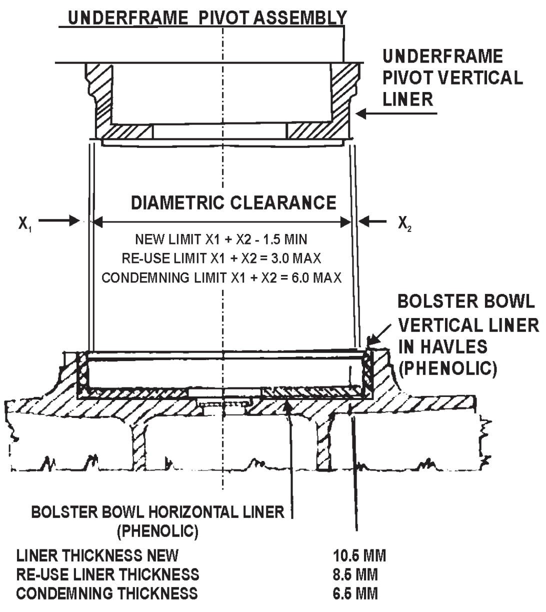


Fig. 6.13 Centre Pivot Assembly

All suspension springs are pretested for proper load/ deflection characteristics as per specified value. Springs are grouped as per variation of loaded height (with respect to nominal) of new spring and colour coded as brown (+5 to +1.6) mm, blue (+1.6 to -1.6) mm and green (-1.6 to -5) mm. The service limit for loaded height is (-5 to -8) mm, and white colour code is to be applied for use of used springs in this range. White coloured springs are to be used with proper shims such that overall static height falls within the limits of brown , blue or green colour coded springs.

6.2.4 High Tensile Steel Cast (HTSC) GM Bogie

6.2.4.1 WDG4/ WDP4

WDG4/ WDP4 locomotives of EMD (Electro-Motive Division, General Motors) design are provided with High Tensile Cast Steel (HTSC) bogie. These 4000 HP locomotives were originally imported from M/s General Motors, USA and are now manufactured by DLW, Varanasi. The maximum permissible speed of WDP4 loco is 160 kmph and that of WDG4 is 105 kmph.

Unlike conventional rigid trucks/bogies, in which axles are held in parallel with each other, the HTSC truck/bogie is designed as a powered "bolsterless" unit. Although the bogie or truck frame itself is rigid, the design allows the end axles to move or "yaw" within the frame. This movement will allow the wheels to position themselves tangent to the rails on curves for reduced wheel and rail wear. Traction loads are transmitted from the truck or bogie to the locomotive underframe through the carbody pivot pin assembly.

The bogies are equipped with three axles. All three axles are powered by AC traction motors in case of WDG4 (Co-Co arrangement), whereas, in case of WDP4, only two axles are powered (AA1-1AA arrangement). The driving force is transmitted to the bogie frame through traction rods attached to the journal bearing adapter and the frame.

The unsprung weight of the locomotive carbody is transferred directly to the bogie frame through four rubber "compression" spring assemblies. These four spring assemblies are located at corner positions formed on the bogie where the side beams and cross beams intersect, thus providing the yaw stiffness for tracking stability. The soft primary suspension, made up of twelve single coil journal springs (two at each journal), is designed to provide ride quality and equalization of wheelset loads for operation over track irregularities. The bogie is equipped with vertical primary shock absorbers (6 no.) and lateral secondary shock absorbers (2 no.) for high-speed operation. General Arrangement of the bogie is shown in Figure 6.14. (a) and (b)

The journal bearing adapters transmit the vertical load from the springs to the axles. Cartridge type grease lubricated journal bearings are provided. These cartridge type bearings are self-contained, preassembled, pre-adjusted, pre-lubricated, and completely sealed. The adapter serves to position the journal springs between the truck/bogie frame and the axle to transmit the vertical loads. It also provides the means to position and control the axle laterally within the frame, as well as longitudinal control through the attached traction rod.

Rubber deflection pads on the adapters and nylon wear plates on the frame control the lateral thrust loads of the axles within the bogie frame. These pads and wear plates are renewable and provide the means by which the lateral clearances can be maintained within limits. Limits of total lateral clearance are 0.38 "(9.6mm) for end axles and 0.62" (15.7mm) for middle axle for WDG 4. These values for WDP4 are 0.24" (6.1mm) and 0.62" (15.7mm) respectively. Service limit of these clearances are 0.62" (15.7mm) and 1.0" (25.4mm) for WDG4 and 0.5" (12.7mm) and 1.0" (25.4mm) for WDP4. The arrangement is shown in **Fig.** 6.15.

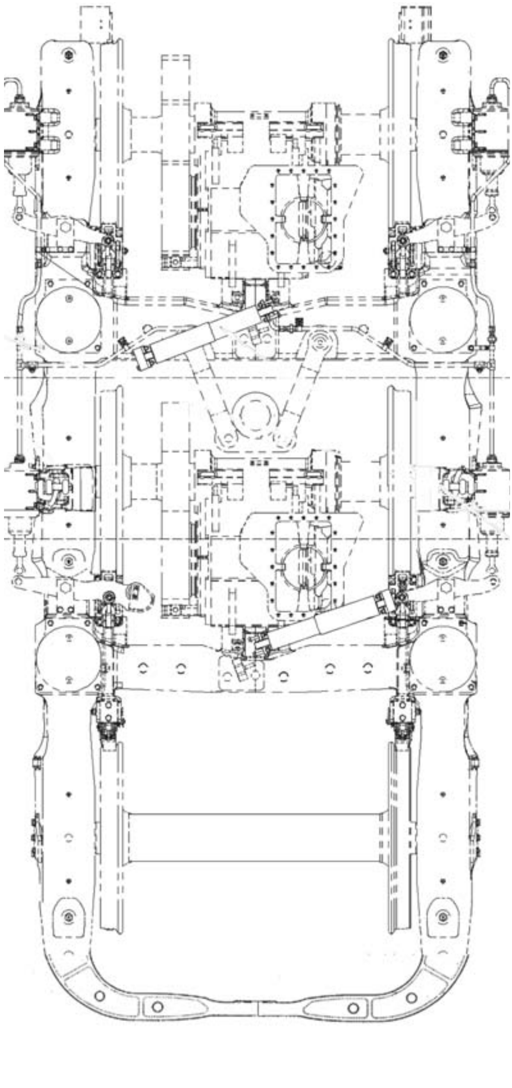


Fig. 6.14 (a) HTSC bogie – General arrangement

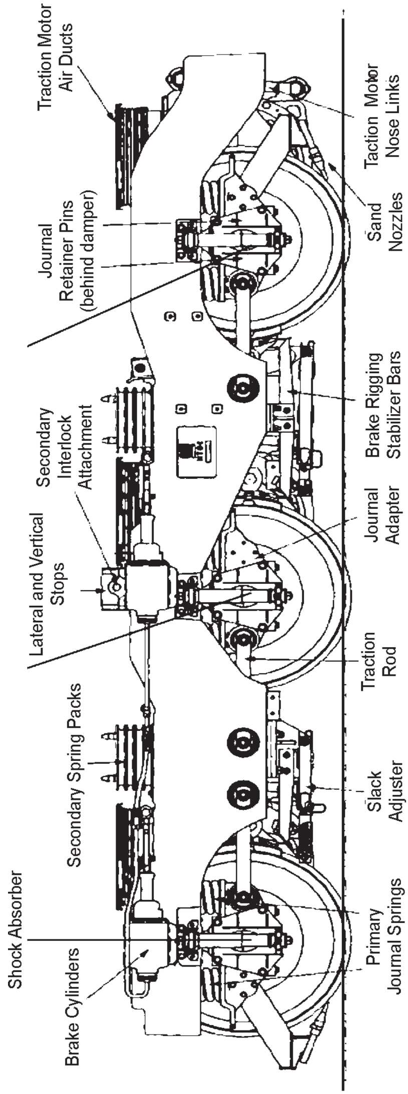


Fig. 6.14 (b) HTSC bogie – General arrangement

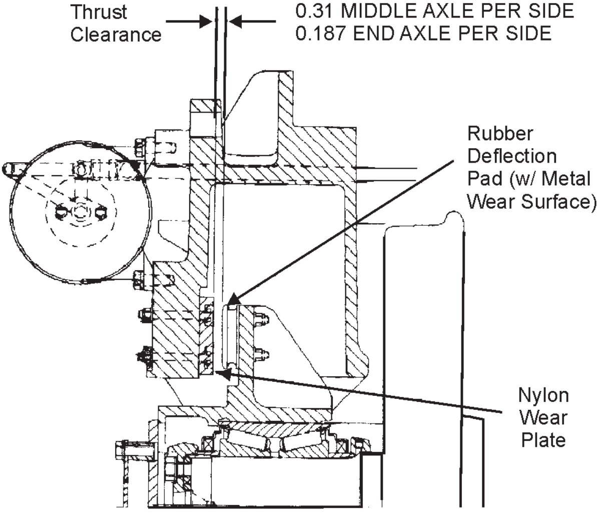


Fig. 6.15 Arrangement for lateral thrust load

The bogie frame is equipped with lateral stops at the center axle position to limit the lateral movement between the bogie and the locomotive underframe. Vertical stop clearance is established between the bogie frame and the underframe at (0.62" +/- 0.12") using shims under the four rubber compression springs and at locations inward of the lateral stops at the center axle position. All shims are tack welded in place(**Fig.** 6.16).

The carbody pivot pin assembly (**Fig.** 6.17) is lined with two Nylon bushing halves. The pivot pin assembly requires regular inspection.The pivot pin is to be sprayed with a bonded type spray lubricant any time the bogie is overhauled or the locomotive carbody is lifted from the bogie. Condition of the wear ring and bushing halves is to be checked when ever accessible and replaced if worn excessively or damaged.

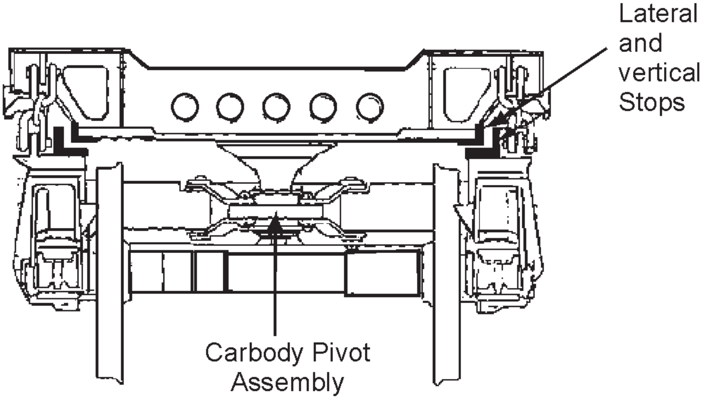
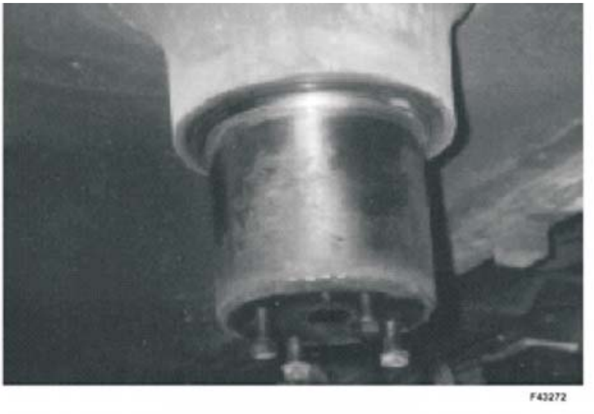


Fig. 6.16.Lateral and Vertical Stops



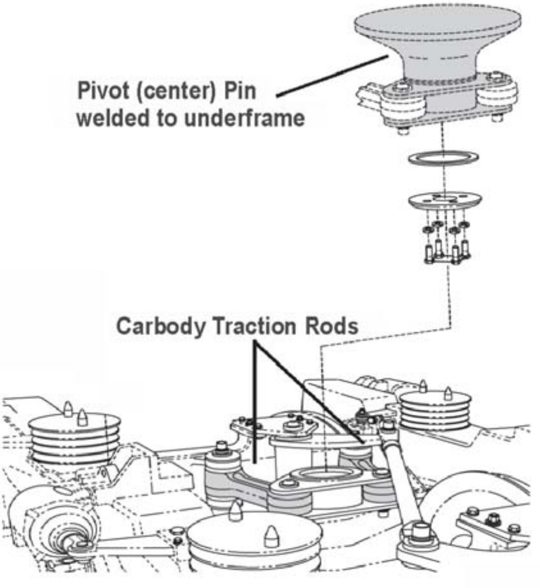


Fig. 6.17. Car Body Pivot Pin and Socket arrangement

Single coils are used in primary suspension that provide for large amounts of deflection. This assists in wheel load equalization, and aids in allowing yaw movement of the traction motor/axle wheel assemblies within the bogie. In order to secure the coil springs on the journal spring adapters, spring pilot tubes are used along with pilot wear plates between the springs and the adapter. Spring pilot pins and shims are also located in the bogie frame spring pockets to perform the same function (Figure 6.18). The pilot plates and shims are chosen to maintain 434.8-mm (17.12") installed spring length. These, along with the shims used between the underframe and the rubber compression secondary springs, serve to maintain proper locomotive height for clearance from the rail to the underframe. This maintains proper coupler height and distributes equal axle loads.

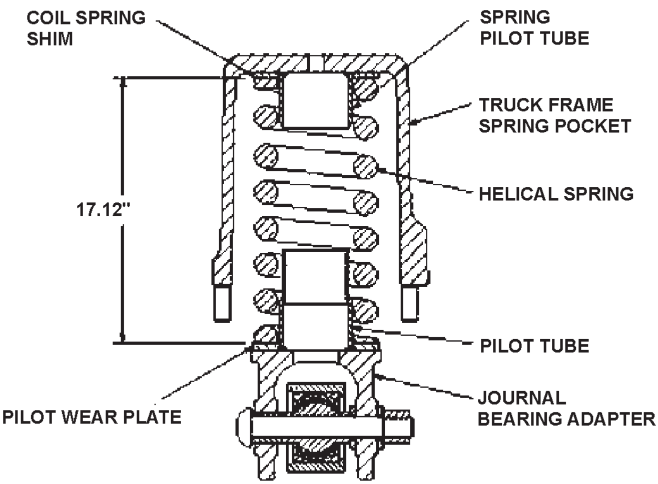


Fig. 6.18. Coil Spring Seat

Other than the specific items mentioned above, common items of rolling stock defects, as discussed earlier in chapter 3, should be checked in case of derailment of these locomotives.

6.2.5 Co-Co Flexi-coil Fabricated Bogie (ABB)

6.2.5.1 WAG9/ WAP7

WAG9 locomotive is a 6000 HP, 3-phase loco having maximum permissible speed of 100 kmph, imported from Bombardier and now being manufactured at CLW with transfer of technology. WAP7 locomotive is having the same design as WAG9, with changed gear ratio for passenger service, having maximum permissible speed of 140 kmph.

The bogie frame consists of two longitudinal sections, which form the side members, and three lateral transoms, which form the cross members. The cross members are welded to the side members forming a rectangular frame. Isolation and absorption of shock loads and vibration is performed by the primary and secondary suspension. General Arrangement of the bogie is shown in **Fig.** 6.19.

The primary suspension (**Fig.** 6.20), located between the axles and the bogie frame, is provided by twin coil springs on the axle journal box fore and aft of the axle line. Vertical hydraulic dampers on the end axles are used to dampen the rebound rate of the springs. Each end axle (wheelset) has four coil springs. The middle axle has an inner and an outer coil spring situated on the axle box fore and aft of the axle centre line. Each middle axle (wheelset) has eight coil springs. This "Flexicoil" arrangement permits lateral movement of the axle. Longitudinal control of the axle, and the transmission of tractive and braking effort to the bogie frame, is provided by guide rods connected between the axle journal boxes and bogie frame. Spheribloc rubber bushes in the guide rods allow the axle lateral movement without undue restriction.

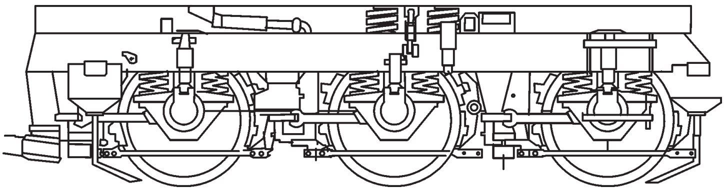
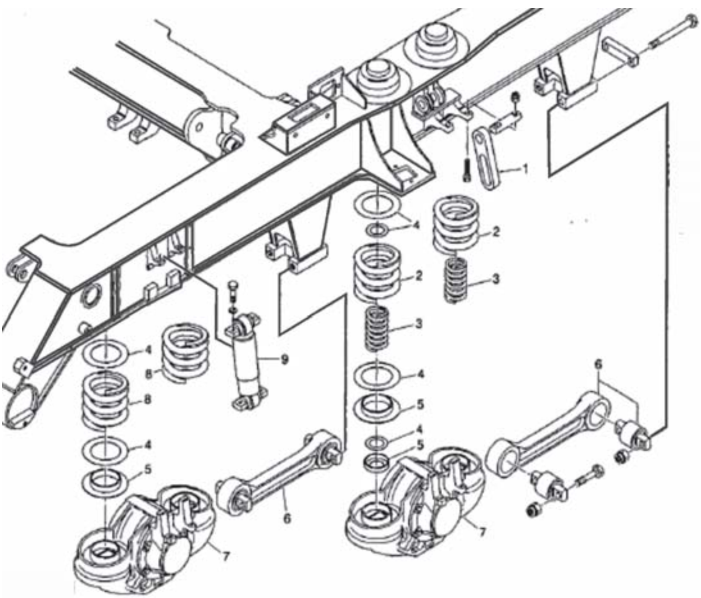
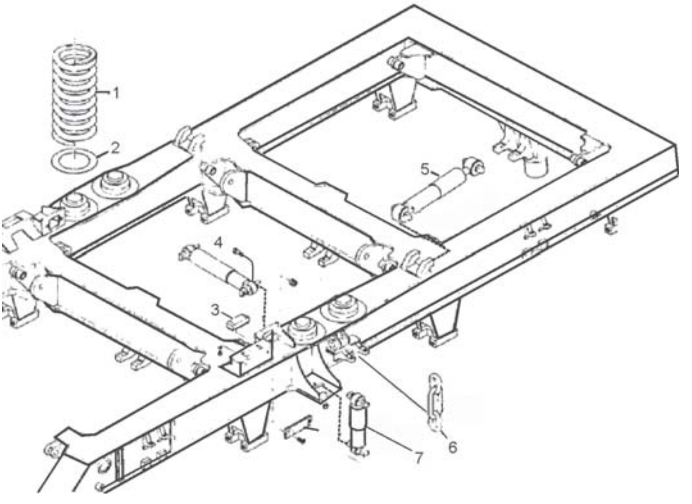


Fig. 6.19 General Arrangement



1	Middle Axle Safety Link	6	Guide rod
2	Middle Axle Outer Coil Spring	7	Axle Box
3	Middle Axle Inner Coil Spring	8	End Axle coil spring
4	Compensating Plate	9	Vertical Damper
5	Insulating Base		

Fig. 6.20 Primary Suspension



1 Coil springs	6 Safety chains
2 Compensating plates	7 Vertical Hydraulical Dampers
3 Vertical Bump Stops	
4 Lateral Hydraulic Dampers	
5 Yaw Hydraulic Dampers	

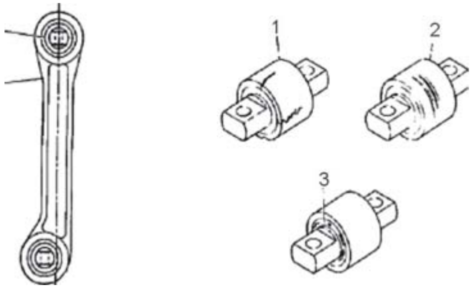
Fig. 6.21 Secondary Suspension

Secondary suspension (**Fig.** 6.21), is also provided by coil springs and vertical hydraulic dampers located between the bogie frame and the locomotive underframe on each side of the bogie. The weight of the locomotive car body is carried by the secondary suspension springs. The "Flexicoil" arrangement of the secondary suspension allows the locomotive car body to move both laterally and vertically within certain limits relative to the bogies. Although the springs allow movement in any direction, lateral buffers and dampers limit the amount and rate of lateral movement, while rebound limit chains and vertical dampers limit the amount and rate of the vertical rebound of the locomotive car body. Longitudinal (yaw) dampers are provided to dampen the rate of pitch of the car body. Push-pull link rod, connected between the bogie transem and locomotive underframe, control the fore and aft movement between the bogies and the locomotive and transfer the logitudinal forces.

Slowing or stopping the locomotive is achieved by use of electrically regenerative braking or pneumatically operated drum brakes. The brake cylinders and rigging are positioned to the end. The Co-Co bogie is fitted with wheel flange lubrication equipment to minimize wheel wear and sanding equipment is installed to provide additional traction between the wheels and rail when required.

The Guide Rod, which is bolted to the axle box housing and to the bogie frame, provides a longitudinal guide for the axle box housing. The guide rod is fitted with spherical rubber joints (Spheriblocs) at each end, which provide positive longitudinal guidance while allowing negligible resistance to lateral movement.

The Spheriblocs in the axle guide rods are critical to maintaining wheel alignment. No damage or wear is permissible on either the metal or rubber parts. During maintenance, damaged or suspect components are to be replaced with new parts. Where one spheribloc on a bogie is deemed unfit for service it is good engineering practice to thoroughly inspect all Spheriblocs on the bogie. Spheriblock Axial Wear Test of is prescribed in the Maintenance Manual, wherein an axial load (of 1500 kg) is applied on the cross pin and then released. Amount of residual displacement is restricted to 0.6mm, else, the spheriblock is to be replaced. Common damages of spheriblock are shown in **Fig.** 6.22.



Guide Rod

1. Spheriblock collar – crack
2. Spheriblock collar – gouge
3. Rubber element- tear, crack, disintegration

Figure 6.22 Guide Rod and Common damages of Spheriblock

Stipulated heights corresponding to different load for Primary Spring are given in **Table** 6.4. During maintenance, the free height of spring is measured and it is replaced if below the stipulation. Further, the spring are tested in a testing fixture. Loads are applied, as in the table and heights are measured at each given load. Any spring not within specification is replaced.

Table 6.4

Spring	Spring condition / Item	Unloaded	Static load	Load at stop	Load at solid
End axle spring	**Deflection (mm)**	0.0	46.8	76.8	91.2
	Spring ht. (mm)	238.8	192.0	162.0	147.6
	Load (kN)	0.0	40.6	66.6	79.1
Mid axle outer spring	**Deflection (mm)**	0.0	66.6	96.6	113.7
	Spring ht. (mm)	258.6	192.0	162.0	144.9
	Load (kN)	0.0	31.5	45.7	53.8
Mid axle inner spring	**Deflection (mm)**	0.0	64.4	94.4	110.5
	Spring ht. (mm)	252.4	188.0	158.0	141.9
	Load (kN)	0.0	9.3	13.7	16.0

Similar action is taken during maintenance with Secondary Spring. Data for these springs is given in **Table** 6.5.

Table 6.5

Spring condition	Deflection (mm)	SpringHeight (mm)	Load (kN)
Unloaded	0	733.2	0
Static load	158.2	575.0	96.8
Load at stop	195.2	540.0	118.2
Load at solid length	250.9	482.3	153.6

All springs vary slightly from batch due to the manufacturing process. The springs used on WAG-9 locomotives are marked with one or two aluminum bands to identify their tolerance range. The number of bands also corresponds to the number of compensating plates below the springs. All the primary and all the secondary suspension springs on the bogie must be of the same tolerance range. Springs of different tolerance ranges should not be mixed.

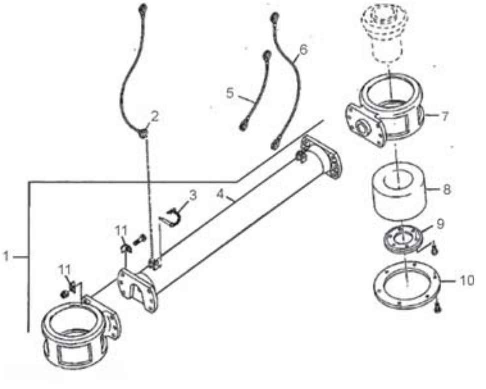
Adjustment of individual wheel loads is performed by adding compensating plates between the axle boxes and primary suspension springs. Wheel loads for a bogie are measured on weighbridge. The adjustment compensates for twist within the bogie frame or variations in spring lengths or stiffness. After the load adjustment, the distance between the axle and bogie frame must be within the range of 27-35 mm.

Adjustment of individual axle loads is performed by adding compensating plates between the bogie frame and the secondary suspension springs. The adjustment compensates for variations in the bogie frame, locomotive underframe and spring lengths or stiffness. After the load adjustment, the clearance between vertical bump stops and locomotive underframe must be within 32-60 mm.

The nominal axle load is 20.5 tonnes. Axle loads must be within 2% of the nominal axle load and the wheel loads must be within 4% of the measured axle load.

During maintenance, correct number and location of compensating plates is required to be ensured. Vertical and lateral bump stop arrangement is to be inspected for wear and damages, and exceedence of the clearances..

The **Traction Link** transmits traction and braking forces from the bogie to the locomotive superstructure (underframe). Although the traction link maintains the relative longitudinal position of the bogie to the locomotive underframe, it permits lateral movement between the two structures. The link rod is situated between two pivot points, one on the locomotive underframe the other on the end transom of the bogie, permitting lateral movement but restraining longitudinal movement. A pivot head, situated at each end of the link rod, has a ring of pliable material between the pivot post and head. The rings are secured to the pivot head by an outer retaining ring and a retaining plate bolted to the post **(Figure 6.23).**



1	Traction link	7	Pivot head
2	Safety cable, bogie end	8	Ring
3	Chain	9	Inner retaining plate
4	Link rod	10	Outer retaining ring
5	Safety cable, short	11	Locking tab
6	Safety cable, long		

Fig. 6.23 Traction Link Arrangement

Physical examination, such asr crack, breakage, deterioration of material etc. is important for the assembly. Maximum permissible inside diameter of the pivot head is 251.5mm. The ring is to be replaced if there is a deterioration of material or depth of damage exceeds 10 mm.

Some of the other important features of the locomotive bogie are

Nominal Axle load	20,500 kg
Wheel bases	1,850mm
Overall length	6209mm
Overall width	2962mm
Overall height	1240mm

Service limite of clearances

i)	Vertical clearance between axle box and bogie frame.	27 - 35 mm
ii)	Lateral clearance between axle box and bogie frame.	15 - 22 mm
iii)	Vertical clearance between bogie frame and under frame	32 - 40 mm
iv)	Lateral clearance between bogie frame and under frame	45 - 55 mm

6.2.6 Bo-Bo Flexi-coil Fabricated Bogie (ABB)

6.2.6.1 WAP5

WAP5 is 5500 HP, high speed (Max speed 160 kmph, upgradable to 220 kmph) locomotive. It was imported from ABB (now Bombardier) and is now being manufactured at CLW with transfer of technology.

Basic features of this bogie are similar to that of WAG9, discussed above, except that it is a BO – BO design.

Bogie frame is a fabricated steel structure. Wheel base is 2.800 m. Primary Suspension comprises of twin coil spring and a hydaulic damper on each axle box. Longitudinal control of the axle and transfer of longitudinal forces is done by Guide Rods connected between axle box and bogie frame. Flexi-coil action of primary springs permits lateral movement of axle with respect to bogie frame. The system is similar to WAG9 locomotive.

Vertical and lateral movement between wheel set and bogie frame are limited by vertical and lateral bump stop clearances (Figure 6.24). Limit of clearances between bogie and axle box are 27 to 35 mm for vertical (A) and 15 to 20 mm for lateral (B).

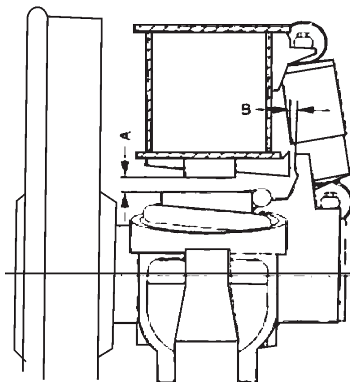


Figure 6.24 Bump Stop Clearances at primary stage

Primary spring height at specified loading is shown in **Table** 6.5

Table 6.5

Spring condition	**Deflection (mm)**	**Spring Height (mm) (min – max)**	**Load (kN)**
Unloaded	0	228.2 – 231.0	0
Static load	53.2	175.0 – 177.8	41.1
Load at stop	83.2	145.0 – 147.8	64.3
Load at solid length	95.0	133.2 – 136.0	73.4

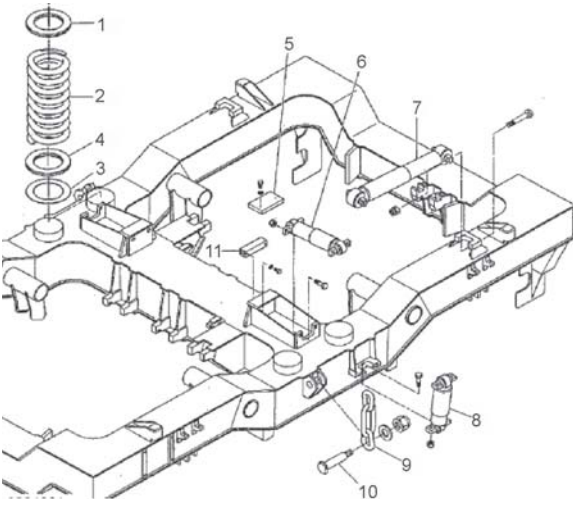
Secondary Suspension comprises of coil springs and vertical, lateral and yaw dampers. Secondary suspension arrangement is shown in Figure 6.25. Vertical and lateral movement between locomotive body and bogie frame are limited by buffer and chain arrangement. Limit of clearances between bogie and under frame are 30 to 40 mm for vertical (D) and 20 to 40 mm for lateral (C) (**Fig.** 6.26).

Secondary spring height at specified loading is shown in **Table** 6.6

Table 6.6

Spring condition	**Deflection (mm)**	**Spring Height (mm)**	**Load (kN)**
Unloaded (nominal)	0	664.5	0
Static load Group 1* (min-max)	184.5	480 - 490	61.5
Static load Group 2* (min-max)	184.5	470 - 480	61.5
Load at stop (nominal)	219.5	445	73.2
Load at solid length (nominal)	286.1	378.4	95.4

*Group 1 – one band, Group 2 – two band



1.Insulating bases top 2.Coil spring 3.Compensating plate 4.Insulating base bottom 5.Vertical bump stop 6.Lateral hydraulic damper7. Yaw hydraulic damper 8.Vertical hydraulic damper 9.Safety chain 10. Pin 11. Lateral bump stop

Figure 6.25 Secondary suspension

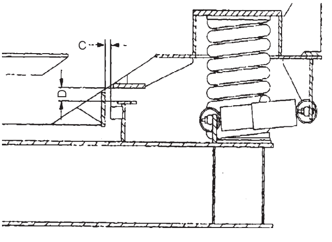


Figure 6.26 Bump Stop Clearances at secondary stage

Arrangements of Guide Rod and Traction Link are similar to that in WAG9 locomotive discussed above.

Table 6.7 and 6.8 show some important parameters ofdifferent diesel and electric locomotives, respectively.

Table 6.7

Diesel locomotives

Class	Wheel Arrangement	Power kW	Speed km/h	Weight tones	First built	Mechanical	Engine	Transmission
Broad-gauge								
WDM2 (Mod)	Co-Co	1,790	120	112.8	1962	DLW	Alco/V251B	EBHEL
WDM3A	Co-Co	2,310	120	112.8	1994	DLW	Alco/V-251B/Upgraded	EBHEL
WDG3A	Co-Co	2,310	100	123	1996	DLW	Alco	EBHEL
WDP1	Bo-Bo	1,715	120	80	1995	DLW	Alco	EBHEL
WDM4	Co-Co	1,965	120	113	1962	GM	GM567D3	EGM
WDM6	Bo-Bo	895	75	70	1981	DLW	DLW/251.B	EBHEL
WDM7	Co-Co	1,475	105	96	1987	DLW	DLW-Alco/ 251-B 12Eye	EBHEL
WDS4	C	520	65/27	60	1969	Mak/CLW	Mak/CLW/ 6M282A(k)	H/HM KPC Voith L4v2U2
WDS4A	C	490						H
WDS4B		520						H
WDS4D		450	65	60	1968	CLW	Mak/6M282A (k)	H Voith L4r2u
WDP3A	Co-Co	2,310	160	117	1998	DLW		DLW
WDG4	Co-Co	2,984	105	126	2000	GM/DLW	GM/DLW	
WDP4	Co-Co	2,984	160	126	2001	GM/DLW	GM/DLW	

WDS5	Co-Co	795	109	126	1967		Alco/251-B	EGE
WDS6	Co-Co	1,045	62.5	126	1977	DLW	Alco/DLW/251D	EBHEL
Metre-gauge								
YDM1-IR	B-B	472	80	46.6	1955	Mak/CLW	Mak/CLW/ 6M282A	H Voith
YDM2	B-B	520	75	48	1986	CLW	CLW/Mak/ 6M282A(k)	H Voith
YDM3	B-B	1,055	80	58.5	1961	GM	GM 12 567c	E GM
YDM4	Co-Co	1,045	96	72	1961	Alco/DLW	Alco 251-D	E BHEL
YDM4A	Co-Co	1,045	96	72	1964	MLW Canada	MLW/25 1-D	E GE Canada
YDM5	C-C	1,035	80	69	1964	GM	GM 12 567c	E GM
Narrow-gauge								
NDM1	B-B	110	33	29	1955	Am-Jung	MWM TRHS/ 518 S	H Voith L33U
NDM5	B-B	365	50	22	1987	CLW	Cummins/KTA 1150L	H Voith L2r2zu2
ZDM2R	B-B	520	50	32	1964	Mak	Maybach/MD 435	HM Mak
ZDM3	B-B	520	50	35	1971	CLW	CLW/Mak/ 6M282A(k)	HM Kirloska
ZDM4A	B-B1	520	50	38.5	1984	CLW	CLW/Mak/ 6M282A(k)	H Voith L4r 22
ZDM5	B-B	365	50	22	1990	CLW		

Table 6.8

Electric locomotives

Class	Wheel arrangement	Output kW continous/ one hour	Speed km/h	Weight tones	No in service	First built	Builders Mechanical	Builders Electrical
Broad-gauge 1.5 kV DC								
WCM6	Co-Co	3,430	105	120	2	1996	CLW	CLW
WCG2	Co-Co	3,130/1,220	90	132	57	1971	CLW	CLW
Broad-gauge 25 kV AC								
WAM4	Co-Co	2,715/2,870	110	112.8	471	1971	CLW	CLW
WAP1/3	Co-Co	2,800/2,910	130/140	107	64	1980	CLW	CLW
WAP4/6	Co-Co	3,730/3,990	140	107	294	1995	CLW	CLW
WAP5	Bo-Bo	2,985/4,000	160	78	14	1996	ABB	ABB (3 phase loco)
WAP7	Co-Co	4,565	140	123	17	2001		Bombardier/CLW (3 phase loco)
WAG5	Co-Co	2,870/3,250	80	118.8	1,171	1978	CLW	CLW/BHEL
WAG7	Co-Co	3,730/3,990	100	123	673	1992	CLW	CLW
WAG6A	Bo-Bo-Bo	4,475/4,560	100	123	6	1988	ASEA	ASEA

WAG6B	Bo-Bo-Bo	4,475/4,560	100	123	6	1988	Hitachi	Hitachi
WAG6C	Co-Co	4,475/4,560	100	123	6	1988	Hitachi	Hitachi
WAG9/WAG9H	Co-Co	4,565	100	123	61	1996	Adtranz	Adtranz/CLW 3 phase loco
Broad-gauge dual voltage 25 kV AC/1.5 kV DC								
WCAM1	Co-Co	2,715/2,870 2,185	120/80	113	52	1975	CLW	CLW
WCAM2	Co-Co	3,505 2,160	120/80	113	20	1995	BHEL	BHEL
WCAM3	Co-Co	3,730 3,432	105	121	53	1996	BHEL	BHEL
WCAG1	Co-Co	3,730 3,432	100	128	12	1998	BHEL	BHEL
Metre-gauge 25 kV AC								
YAM1	Co-Co	1,215/1,300	80	52	18	1965	Mitsubishi	Mitsubishi

First figures apply to AC operation, subsequents figures to DC

ANNEXURE 1

CLEARANCES FOR DIESEL LOCOMOTIVES
(All Dimensions are in mm)

Loco	New wheel diameter	Root wear limit	Flange wear	Total tread wear	Lateral & Longitudinal clearances			Difference in wheel diameter		
					Measurement	New condition	Service limit	Location	Diff. in Dia.	Permissible limit
					As per MP.MI 71/78			As per MP.IB.BD.01.01.05 dt.01/02/05		
WDM2 (Co-Co trimount bogies without conical thrust pad)	1092 (+5/-0)	6.0	3.0	6.5	Lateral clearances End Axle (C1+C2) per axle	6.0	12.0	On same axle	0.5	-
					Lateral clearances middle axle (B1+B2) per axle	25.0	31.0	On same bogie	2.0	8.0
					Longitudinal clearances for middle & End axles (A1+A2) per axle box	0.363-1.9	5.0	On same loco	15.0	25.0
					As per MP.IB.VL – 03.04.06 (Rev 01) dt. 28.11.08			As per MP.IB.BD.01.01.05 dt.01/02/05		
WDM3A & WDM3C (Co-Co Trimount Bogies with conical thurst pad)	1092 (+5/-0)	6.0	3.0	6.5	Lateral clearances End Axle (C1+C2+C3+C4) per axle	22.4-24.8	30.0	On same axle	0.5	-
					Lateral clearances middle axle (B1+B2+B3+B4) per axle	4.0-6.6	12.0	On same bogie	2.0	8.0
					Longitudinal clearances for middle & End axles (A1+A2) per axle box	0.4-1.9	3.5	On same loco	15.0	25.0

Loco	New wheel diameter	Root wear limit	Flange wear	Total tread wear	Lateral & Longitudinal clearances			Difference in wheel diameter		
					Measurement	New condition	Service limit	Location	Diff. in Dia.	Permissible limit
					As per MP.IB.VL.01.02.06 (Rev.0.00)dated 27.01.2006			As per MI.VL.02/96		
WDG3A/ WDM3D	1092 (+5/-0)	6.0	3.0	6.5	Lateral clearances End Axle (C1+C2+C3+C4) per axle	22.0-25.2	30.7	On same axle	0.5	2.5
					Lateral clearances middle axle (B1+B2 +B3+B4) per axle	2.4-6.0	11.5	On same bogie	2.0	8.0
					Longitudinal clearances for middle & End axles (A1+A2) per axle box	2.0-4.0	6.0 (SV.WDG2 dt.19.02.99)	On same loco	15.0	25.0
					As per MP.IB.VL.05.06.06 (Rev.0.00) dated 03.08.2006			As per DLW MI-CHS=009		
WDP1	1092 (+5/-0)	6.0	3.0	6.5	Lateral clearances All Axles (C1+C2+C3+C4) per axle	22.0-25.0	31.0	On same axle	0.5	1.5
					Longitudinal clearances for middle & End axles (A1+A2) per axle box	0.6-2.2	5.0	On same bogie	2.0	5.0
								On same loco	15	20
					As per Mi.VL-04/98					
WDP3A	1092 (+5/-0)	6.0	3.0	6.5	Lateral clearances middle axle (C1+C2) per axle box	1.2-3.0	6.0	On same axle	0.5	2.5
					Longitudinal clearances (Middle axle) (A1+A2) per axle box	2.0-4.0	6.0	On same bogie	2.0	8.0
								On same loco	15	25

Loco	New wheel diameter	Root wear limit	Flange wear	Total tread wear	Lateral & Longitudinal clearances					Difference in wheel diameter		
					Measurement	New condition	Service limit			Location	Diff. in Dia.	Permissible limit
					IB No.MP.IB.VL-03.28.08(Rev.00) dated. 30.09.08					EMD maintenance instruction no.1517 (Rev. A)		
						Nominal Inch/ m m	**Min. Inch/ m m**	**Max. Inch/ m m**	**Service limit Inch/mm**			
WDG4	1092 (+5/-0)	6.0	3.0	6.5	**For End axle Lateral** clearances between thrust pad & bearing adapter (A1+A2) per	0.38/9.6	0.3/7.6	0.5/12.7	0.62/ 15.7	On same axle	0.5	1.6
					For middle axle Lateral clear-ances between thrust pad & bearing adapter (B1+B2) per axle	0.62/ 15.7	0.54/ 13.7	0.74/ 18.8	1.0/ 25.4	On same axle	3.2	6.4
										On same axle	14.2	31.8

Loco	New wheel diameter	Root wear limit	Flange wear	Total tread wear	Lateral & Longitudinal clearances					Difference in wheel diameter		
					Measurement	New condition	Service limit			Location	Diff. in Dia.	Permissible limit
					IB No.MP.IB.VL-03.28.08(Rev.00) dated. 30.09.08					EMD maintenance instruction no.1517 (Rev. A)		
WDP4	1092 (+5/-0)	6.0	3.0	6.5	**For End axle Lateral** clearances between thrust pad & bearing adapter (A1+A2) per	**Nominal Inch/ m m**	**Min. Inch/ m m**	**Max. Inch/ m m**	**Service limit Inch/mm**	On same axle	0.5	1.6
						0.24/ 6.1	0.16/ 4.1	0.36/ 9.1	0.5/ 12.7			
					For middle axle Lateral clear-ances between thrust pad & bearing adapter (B1+B2) per axle	0.62/ 15.7	0.54/ 13.7	0.74/ 18.8	1.0/ 25.4	On same axle	3.2	6.4
										On same axle	14.2	31.8

CLEARANCES FOR ELECTRIC LOCOMOTIVES
(All dimensions are in mm)

Loco	New wheel diameter	Wear limits			Axle Box Clearances				Difference in Wheel diameter		
		Root	Flange	Tread	Measurement	New condition Min.	New condition Max.	Service limit	Location	Diff. in Dia.	Permissible limit
					As per MP.IB.VL-03.04.06(Rev.0.0) dated 28.11.08				As per MP.MI/71/78 Jul.92		
WAM4	1092 (+5/-0)	8	3	6.5	Lateral clear-ances End Axle per axle (C1+C2+C3+C4)	22.4	24.8	30.0	Same axle	0.5	2.5
					Lateral clear-ances Middle Axle per axle (B1+B2+B3+B4)	4.0	6.6	12.0	Same Bogie	2.0	8.0
					Longitudinal clearances Middle & End Axle per axle box.	0.4	1.9	3.5	Same Loco	15.0	25.0
					As per MP.IB. VL.04.05.06(Rev.00)dt.03.08.06				No.SD.WA1 dated. 10.10.2003		
WAP1/ WAP4	1092 (+5/-0)	4	2.5	6.5	Lateral clear-ances End Axle per axle (C1+C2+C3+C4)	15.4	18.2	24.0	Same axle	0.5	1.5
					Lateral clear-ances Middle Axle per axle (B1+B2+B3+B4)	3.6	7.2	12.0	Same Bogie	2.0	5.0
					Longitudinal clearances Middle & End Axle per axle box.	0.4	2.0	4.0	Same Loco	15.0	20.0

Loco	New wheel diameter	Wear limits			Axle Box Clearances				Difference in Wheel diameter		
		Root	Flange	Tread	Measurement	New condition Min.	New condition Max.	Service limit	Location	Diff. in Dia.	Permissible limit
WAP7	1092 (+0.5/-0)	4	2.5	6.5	Technical Circular No.ELRS/TC/0082(Rev00) dated 29.06.05				No.SD.WA1 dated. 24.06.05		
WAG9	1092 (+0.5/-0)	6	3.0	6.5	Vertical Clearance between Axle Box & Bogie Frame	30	35	27 to 35	Same axle	0.5	2.5
					Lateral Clearance between Axle Box & Bogie Frame	15	19	15 to 22	Same Bogie	2.0	4.0
					Vertical Clearance between Bogie Frame & Under Frame	35	40	32 to 40	Same Loco	15.0	20.0
					Lateral Clearance between Bogie Frame & Under frame	45	50	45 to 55			

Loco	New wheel diameter	Wear limits			Axle Box Clearances				Difference in Wheel diameter		
		Root	Flange	Tread	Measurement	New condition Min.	New condition Max.	Service limit	Location	Diff. in Dia.	Permissible limit
					Report No.MT-60 September 96 & drawing no.1A011-00140 Sheet 2				No. SD. WA1 dt. 01.09.02		
WAP5	1092 (+0.5/-0)	4	2.5	6.5	Vertical Clearance between Axle Box & Bogie Frame	27	35	--	Same axle	0.5	2.5
					Lateral Clearance between Axle Box & Bogie Frame	15	20	--	Same Bogie	2.0	4.0
					Vertical Clearance between Bogie Frame & Under Frame	30	40	--	Same Loco	15.0	20.0
					Lateral Clearance between Bogie Frame & Under frame	20	40	--			

Loco	New wheel diameter	Wear limits			Axle Box Clearances				Difference in Wheel diameter		
					Measurement	New condition		Service limit	Location	As turned /new	Service Limit
		Root	Flange	Tread		Min.	Max.				
WCAM1, WCAM2	1092 (+5/-0)	6	3	6.5	As per MP.IB.VL-03.04.06(Rev.00) dt.13.06.06				As per MP/MI 36/73 April, 79		
					Lateral clearances End Axle per axle (C1+C2+C3+C4)	22.4	24.8	30.0	Same Axle	0.5	1.5
					Lateral clearances Middle Axle per axle (B1+B2+B3+B4)	4.0	6.6	12.0	Same Bogie	2.0	5.0
					Longitudinal clearances Middle & End Axle per axle box(A1+A2)	0.4	1.9	3.5	Same Loco	15.0	20.0
WCAM3/ WCAG1	1092 (+5/-0)	6	3	6.5	Letter no.EL/3.2.108 dated 16.10.07				As per VL.MI-01/96		
					Lateral clearances End Axle per axle (C1+C2+C3+C4)	22.0	25.2	30.0	Same Axle	0.5	2.5
					Lateral clearances Middle Axle per axle (B1+B2+B3+B4)	2.4	6.0	11.5	Same Bogie	2.0	8.0
					Longitudinal clearances Middle & End Axle per axle box(A1+A2)	0.6	2.0	4.0	Same Loco	15.0	25.0

Loco	New wheel diameter	Wear limits			Axle Box Clearances				Difference in Wheel diameter		
					Measurement	New condition		Service limit	Location	As turned /new	Service Limit
		Root	Flange	Tread		Min.	Max.				
WAG7	1092 (+5/-0)	6.0	3.0	6.5	As per Mp.IB.VL-02.03.06(Rev.0.00) dated 27.01.2006				As per MI.VL-01/96		
					Lateral clearances End Axle per axle (C1+C2+C3+C4)	22.0	25.2	30.7	Same Axle	0.5	2.5
					Lateral clearances Middle Axle per axle (B1+B2+B3+B4)	2.4	6.0	11.5	Same Bogie	2.0	8.0
					Longitudinal clearances Middle & End Axle per axle box(A1+A2)	2.0	4.0	6.0	Same Loco	15.0	25.0

CHAPTER 7

TRACK DEFECTS

Track defects can be roughly categorized into the following:

- Failure or defects in various components of the track (including formation)
- Nature and magnitude of irregularities in track geometry
- Defects concerning special track features e.g. curves, points and crossings, bridge approaches, etc.

7.1 FAILURE OF TRACK COMPONENTS

7.1.1 FAILURE OF FORMATION

This feature is obvious and would not need any elaboration. Sudden subsidence of embankment, slope or base failure, slips, etc. are well known examples of formation failures which could cause a derailment.

Other subtle formation failures e.g. ballast puncturing into the formation soil and forming ballast pockets, settlement of formation top accompanied by cess heave, mud-pumping conditions, problems in formation of expansive soils e.g. black cotton soil, etc. are gradual processes which do not affect safety directly but only through effect on track geometry, retentivity of which is reduced appreciably, needing more frequent attention.

7.1.2 FAILURE OF BALLAST

Ballast performs a vital role in the track structure and could be aptly called the muscle of the track. It absorbs noise, shocks, vibrations and energy, distributes load over the formation, ensures provision of a well drained support, provides a convenient means of bringing up the track geometry, and most important of all, contributes overwhelmingly to the lateral and longitudinal strength of the track (roughly 55 to 70% of the lateral strength of the track is contributed by ballast). Care is, therefore, needed in respect of factors which affect the lateral and longitudinal ballast resistance.

Lateral resistance is essential for ensuring safety against track distortion and buckling.

Longitudinal resistance, if reduced, increases creep in fishplated track (which could lead to jamming of joints and lateral buckling of track) and increases the breathing length in long welded track and, hence, the rail end movements at the switch expansion joint.

Factors which affect the ballast resistance are :

- Ballast material
- Size of ballast
- Shape of ballast particle
- Ballast profile
- State of consolidation
- Deep screening

7.1.2.1 Ballast material

A hard, sound and durable ballast material ensures good resistance for a far longer period of time as compared to inferior ballast, which under traffic and weathering action, wears out and becomes rounded sooner.

7.1.2.2 Size of ballast

There is an optimum size which ensures the maximum resistance. Any size smaller or larger would mean a slight reduction in resistance. 65 mm is the size found to be optimum on the Indian Railways (a certain grading of size is desirable for better resistance).

7.1.2.3 Shape of ballast particle

A crushed stone ballast, angular or cubical in shape, has greater resistance than that of rounded ballast.

7.1.2.4 Ballast profile

To ensure adequate lateral resistance, a minimum width of ballast shoulder is necessary. For long welded track, which is subjected to thermal compressive forces, additional strength is provided by heaping of ballast, as widening the shoulder alone does not give proportionate increase in lateral resistance.

This is amply borne out by studies on the Continental Railways, as reported in ORE Report on Question D 117/RP 8. Some of the relevant results are extracted below: (refer Fig.7.1).

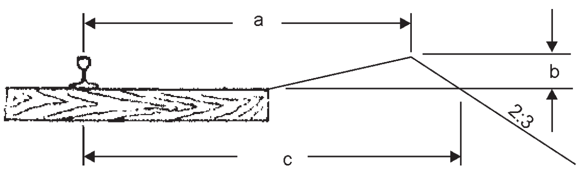


Fig. 7.1 Ballast shoulder and heap

a (cm)	b (cm)	c (cm)	Wooden sleeper
90	0	90	100% (reference value)
120	0	120	102.6%
90	10	105	130.7%
120	10	135	149.2%

7.1.2.5 State of consolidation

Ballast is cohesionless, the best means of compacting which is vibration. Vibrations due to passage of traffic gradually consolidate the ballast to a peak value. As the ballast consolidates, it settles and with it does the track. Since the settlements are not expected to be uniform, the differential settlements of the track result in deterioration of the track geometry. It is to bring up the track geometry that the ballast is given a packing.

Packing involves breaking the consolidated pattern of the ballast particles, loosening them and then packing them back after holding the track to the correct geometry. Such packing can never achieve the same degree of consolidation as existed prior to commencement of packing operations. Thus certain passage of traffic is necessary to regain the earlier peak level of consolidation.

A maintenance operation, necessary for improvement of the track geometry, is accompanied by a drop in the lateral ballast resistance. If the maintenance operation involves lifting of track the loss of resistance would be still more.

It is for this reason that one of the criteria for design of rolling stock is that they should not exert a lateral force greater than the strength of a track which has just been given maintenance attention.

Such limit on the lateral force exerted by an oscillating vehicle had earlier been given by Blonded as 0.4P+2 (in tonnes), where, P= axle load (in tonnes).

Almost all the World Railways, including Indian Railways, have since adopted the Prudhommes limit, viz.

$\left(1+\frac{P}{3}\right)\rho$ where ρ is normally 0.85.

In case of long welded track, maintaining adequate lateral ballast resistance, particularly during the high temperature periods, is crucial for safety against buckling. Interference with the state of consolidation of the ballast, which a maintenance attention inevitably causes, should, therefore, be kept to a minimum. Through maintenance attention should be completed well before the onset of summer (see provisions of LWR/CWR Manual).

7.1.2.6 Effect of deep screening

As deep screening loosens up ballast in the cushion also, the loss in ballast resistance is substantial. It is for this reason that speed restriction is imposed after deep screening and the same is relaxed in stages to match with the gradual build up of strength under traffic (refer IRPWM).

7.1.3 FAILURES OF SLEEPER AND FASTENING

Sleeper and fastening perform the following functions:

- hold the two rails to the desired gauge
- transmit the load from rail to a wider area over the ballast
- provide a resilient support with ability to absorb most of the high frequency vibrations
- provide, along with ballast, lateral and longitudinal strength to the track
- permit rectification of track geometry
- be amenable to packing and retain it
- resist longitudinal creep of rail
- resist overturning of rail

Lateral strength is contributed by sleepers in two ways:

- development of ballast resistance either by friction between sleeper and ballast at bottom (soffit resistance) and sides (crib resistance) and passive ballast resistance at the shoulders (shoulder resistance) of the sleeper, or by physical keying or wedge action of the sleeper, or a combination of both. Fig.7.2 illustrates the above.

- development of lateral track frame stiffness with torsional resistance of the fastenings.

The aspect of ballast resistance has already been discussed. For ensuring adequate frame stiffness of the track, the fastenings should be maintained in tight condition and elastic fastenings should be kept adequately tensioned.

Longitudinal resistance is provided by

- sleeper-ballast resistance (through friction or keying)
- rail fastening-sleeper grip

The lower of the above two determines the longitudinal resistance, as the weaker of the two will give way first. If fastenings are too loose, rails would creep past the sleepers and lead to jammed joints and possibly buckling. Even if fastenings are tight, rail and sleeper may move as one assembly past the ballast, for instance, on steep gradients, thus leading to jammed joints.

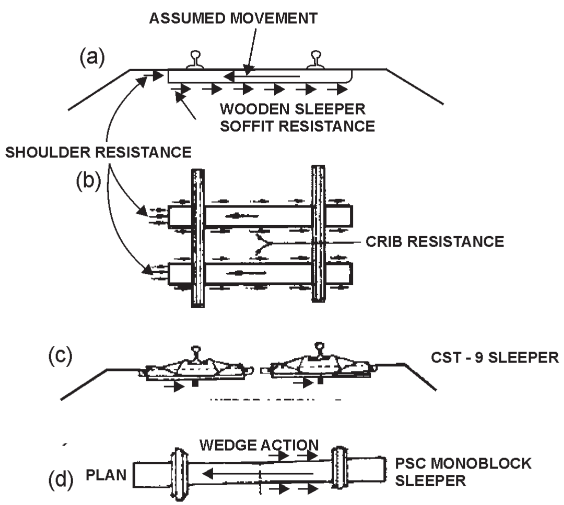


Fig. 7.2 (a) Soffit resistance (b) Crib and shoulder resistance (c) Keying action CST9 sleeper and (d) Wedge action in monoblock PSC sleeper

On long welded rails where build-up of thermal forces occurs, special care is therefore required to be taken in summer to ensure that lateral and longitudinal ballast resistances are not allowed to become low. Prior to onset of summer, distressing of LWR shall be completed if the behaviour of LWR warrants the same. Special precautions during summer on LWR includes, introduction of hot weather patrol when necessary, undertaking track maintenance works strictly within the permitted temperature ranges, etc.

On winter months, special care shall be taken in early detection and replacement of suspected rail defect locations by USFD testing to prevent rail/weld fractures. Behaviour of LWR, as adjudged by the gap measurements at SEJ, shall form the basis for necessity of distressing the LWR and this should be done well before onset of winter.

7.1.4 FAILURES OF RAIL

Two aspects are important :

- rail failure or fracture
- rail wear

The first feature as a contributory cause of derailment is obvious, unless the failure or fracture is as a result of the derailment. This can be established by physical and metallurgical examination of the fractured surface. If the failure or fracture is as a result of the derailment, the entire surface of fracture will show a brittle failure, whereas, if the crack existed prior to derailment, a relatively polished fatigue zone will be evident. Fractured rail pieces resulting in derailment are required to be sent to M&C Directorate of RDSO for examination and analysis.

Locations where incidence of rail fractures may be high include those where corrosion also is predominant e.g. in tunnels, at ashpits, water columns, platform lines etc. It is preferable to test such sections more frequently by ultrasonic rail flaw detector. In tunnels, the rails should be renewed on planned basis and the sound released rails re- used elsewhere.

With regard to the aspect of wear, the wear can be :

- vertical
- lateral
- angular

The permissible limits for the above types of wear are given in Table 7.1. (see Fig. 7.3)

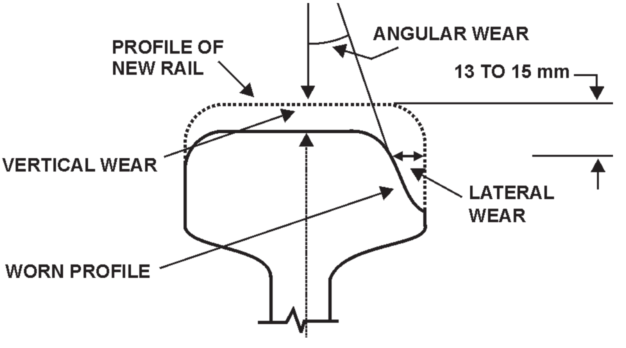


Fig. 7.3 Rail wear

Table 7.1 Limits of Rail Wear

Vertical wear

Gauge	Rail section	Vertical Wear (mm)
B.G.	60kg/m	13.00
	52 kg/m	8.00
	90 R	5.00
M.G.	75R	4.50
	60R	3.00

(N.B. Vertical wear is measured at center line of rail)

Lateral wear

Section	Gauge	Category of Track	Limit of Lateral wear (mm)
Curves	B.G.	Group A & B routes	8
		Group C & D routes	10
	M.G.	Group Q & R routes	9
Straight	B.G.	Group A & B routes	6
		Group C & D routes	8
	M.G.	Group Q routes	6
		Group R routes	8

(**N.B.** Lateral wear is measured at 13 to 15 mm below the rail top table.)

If the vertical wear is excessive, a deep flange may ride over the fish plates or distance or check blocks and thus may damage the track components.

With excessive lateral wear, the play between the wheelset and the track increases which would contribute to increased oscillations and greater angularity of the axle.

Angular wear is the wear pattern which results in gauge face of rail becoming inclined at a slope to the vertical. Its effect on safety would depend on the depth (from rail table top) upto which the slope of rail gauge face (with the horizontal) has becomes smaller than the flange slope. Effective safety depth of flange would be the depth of flange slope below this position. In case the wheel flange is able to mount up the gauge face by this reduced safety depth (under adverse combination of derailing and stabilising forces), further mounting would be easier, as the effective ß in NADAL's formula would be the slope to which the rail gauge face has got worn. Extent to which such wear pattern could affect safety should be analysed in qualitative terms, very carefully.

Angular wear is normally encountered on the outer rail of sharp curves as well as on turnouts. Gauge face lubrication of the outer rail, which is beneficial even otherwise from wear consideration, also increases the margin of safety by reducing the coefficient of flange friction.

Aspects of track geometry will now be taken up.

7.2 TRACK GEOMETRY

In simple terms, the rail tables and the gauge faces of the two rails form the track geometry.

Precisely, track geometry is defined through the geometry of vertical profile as well as the lateral profile of the running surfaces on individual rails. By design, vertical profile is fixed through the track gradients (as reflected on Longitudinal Sections) and through the superelevation or cross level. The lateral profile is fixed by the extent of straight and curved alignments and by track gauge. The profiles of running surfaces form the basic data for construction of railway line. However, during subsequent maintenance, track geometry is measured through track geometry variations parameters, as defined below.

Vertical Profile

i) Unevenness measured as mid chord offset.

ii) Cross level difference/superelevation, as well as its variation with reference to length of track, namely, Twist.

Lateral Profle

i) Alignment measured as mid chord offset.

ii) Slackness or tightness of track gauge, as well as its variation with reference to length of track.

It may be seen that during maintenance, the straightness of the running surfaces are measured as relative measurement, taking the reference of the existing profile itself. For normal speeds of operation (upto 160 km/h), this method of measurement is found to be adequate.

Relative measurement of straightness in vertical plane through mid-chord off-set on a chord of 3.6 m or 7.2 m does not, however, bring out the variations of gradient, which can only be identified by using very long chords. Such defects which are perceptible on long chords are termed as long wave track defects. During normal maintenance, for speeds upto 160 km/h, such measurement on very long chords (of the order of 100 m) is not essential.,However, whenever the track profile is adjusted while executing major track works, like deep screening, track relaying etc., the long wave track defects should be eliminated.

Long wave track defects combined with poor engine-manship can lead to unfavourable forces being exerted over the couplings/ buffers, which can lead to derailment. Therefore, in derailment investigations, the contribution of the track profile measured over long stretches of track by absolute measurement (levelling) should also be examined in conjunction with the engine-manship of the driver for the possible off-loading of the wheels leading to the derailments. Such long wave track defects, as far as high speed passenger rolling stock are concerned, generally lead to unfavourable ride behaviour. These may not lead to unsafe condition unless adversely compounded by other factors. For monitoring long wave track defects on high speed routes on Indian Railways, a chord length of 9.6 m is presently being adopted for unevenness and alignment.

Track geometry can be measured on the spot by using equipments like gauge-cum-level, nylon chord, stepped gauge and scale. Track Recording Cars (TRC) measure the track parameters on a continuous pattern, producing roll charts with marks to locate kilometrage. Use of TRC has enabled categorization of the quality of track geometry and has been providing valuable guidance for track maintenance activities.

IRPWM para 607(2) prescribes the following limits of track tolerances for the guidance of the Engineering officials on the suitability* of standard of maintenance of track for sanctioned speeds above 100 Km/hr. and upto 140 km/hr on BG track.

(i) Alignment defects - (versine measured on a chord of 7.5 metres under floating conditions)
   a) On Straight Track - 5mm; values upto 10mm could

be tolerated at few isolated locations**.

b) On Curves $\pm$ 5mm over the average versine, Values up to $\pm$ 7mm could be tolerated at few isolated locations**. Total change of versine from chord to chord should not exceed 10mm.

(ii) Cross Level Defects - No special tolerance limits. As regards cross levels, the track should be maintained to standards generally superior to that at present available on main line track on which unrestricted speeds upto 100 Km/hr. are permitted.

(iii) Twist- (to be measured on a base of 3.5 m)

(a) On straight and curve track, other than on transitions - 2mm/metre except that at isolated locations**, this may go upto 3.5mm/metre.

(b) On transitions of curves - Local defects should not exceed 1mm/metres, except that at isolated locations** this may go upto 2.1 mm per metre.

(iv) Unevenness rail joint depressions (versine measured on a chord of 3.5m) - 10mm in general and 15mm for isolated locations**.

(v) Gauge variations - No special specifications. The maximum limits for tight and slack gauge should be as indicated in Para 224(2) (e).

(*) Suitability - Suitability refers to good riding quality for passenger comfort and not from stability point of view.

(**) In above `few isolated locations' has been taken as not exceeding 10 per km.

It is noteworthy that safety tolerances have not been laid down for the track geometry as compared to what has been stipulated for rolling stock, not only on the Indian Railways but also on the World Railways. Only US Railroads have laid down Track Safety

Standards. These have been based more or less on an adhoc basis and not on any extensive derailment tests. These tolerances have time and again been found to be wanting and there are frequent modifications to the same.

Such safety track tolerances laid down by Federal Railroad Administration (Governmental body) binding on the various American Railroad Companies (private firms) are not strictly safety tolerances in real terms, but penalty tolerances. If the track defects exceed these tolerances, then the Railroad Company as well as the personnel responsible for track are liable for civil penalty as laid down in the code. To give an idea of the extent of such penalty tolerances, the following is extracted from the Track Safety Standards published by US Department of Transportation.

Track is classified depending on maximum permissible speed as follows :

	The maximum allowable operating speed for freight trains is	The maximum allowable operating speed for passenger trains is
Class 1 track	10 mph (16 km/h)	15 mph (24 km/h)
Class 2 track	25 mph (40 km/h)	30 mph (48 km/h)
Class 3 track	40 mph (64 km/h)	60 mph (96 km/h)
Class 4 track	60 mph (96 km/h)	80 mph (128 km/h)
Class 5 track	80 mph (128 km/h)	90 mph (144 km/h)
Class 6 track	110 mph (176 km/h)	110 mph (176 km/h)

Gauge

Gauge must be within the limits as per the following table:

Class of track	The gauge must be atleast	But not more than
1	4'8" (12 mm tight)	4'10" (38 mm slack)
2 and 3	4'8" (12 mm tight)	4'9¼" (32 mm slack)

4 and 5	4'8" (12 mm tight)	4'9½" (25 mm slack)
6	4'8" (12 mm tight)	4'9¼" (20 mm slack)

Alignment

Alignment may not deviate from uniformity more than the amount prescribed in the following table:

Class of track	**Tangent track** The deviation of the mid-offset from 62 foot (18.9 m) line may not be more than (in inch)	**Curved track** The deviation of the mid-ordinate from 62-footchord (18.9 m) may not be more than (in inch)
1	5 (127 mm)	5 (127 mm)
2	3 (76 mm)	3 (76 mm)
3	1 ¾ (44 mm)	1 ¾ (44 mm)
4	1 ½(38 mm)	1 ½ (38 mm)
5	¾ (20 mm)	5/8 (16 mm)
6	½ (12 mm)	3/8 (10 mm)

Track surface

Each owner of track to which this part applies shall maintain the surface of its track within the limits prescribed in the following table :

Track surface	Class of Track (Fig in inch)					
	1	2	3	4	5	6
The run-off in any 31 feet of rail at the end of a raise may not be more than	3½	3	2	1½	1	½
The deviation from uniform profile on						

either rail at the mid-ordinate of a 62 foot chord may not be more than	3	2¾	2¼	2	1¼	½
Deviation from designed elevation on spirals may not be more than	1¾	1½	1¼	1	¾	½
Variation in cross level on spirals in any 31 feet may not be more than	2	1¾	1¼	1	¾	½
Deviation from zero cross level at any point on tangent or from designed elevation on curves between spirals may not be more than	3	2	1¾	1¼	1	½
The difference in cross level between any two points less than 62 feet apart on tangents and curves between spirals may not be more than	3	2	1 ¾	1 ¼	1	5/8

Such an approach on Indian Railways was not followed mainly because these tolerances are very much slack than what has been laid down as service tolerances as per categories A, B, C and D and may therefore, lead to slackness in upkeep of track.

There is yet another odd against stipulation of safety tolerances for track. some types of vehicles (such as CRT wagons, 4-wheeler tank wagons, etc.) have proved to be very vulnerable to derailment, even on good track where express trains run with comfortable riding quality. Obviously, safety track-tolerances for such sensitive vehicles are tighter than those for other vehicles. Therefore, it is impossible to stipulate safety tolerances to be adopted universally with respect to all vehicles. It will prove to be extremely costly, unmanageable and unwise to adopt the safety track-tolerances suitable for the sensitive vehicles, as the universal standard, considering the undue burden to be faced for maintaining thousands of kilometers of track.

In the background of foregoing, the irregularities in track geometry are discussed in the following paras.

7.2.1 GAUGE

It has been well established that a uniform departure, but within practical limits does not have any adverse effect on safety, stability or comfort. What are the practical limits for safety, however, are difficult to establish for reasons already discussed.

If the gauge is more or less uniform, the extent of its departure from the standard value can be discussed only in general terms. Besides, this parameter has to be considered along with the thickness of flange and the play and clearances at axle box level, as ultimately it is the total effective play between the wheelset and the track which is actually relevant. If this play is excessive the wheelset will be prone to increased oscillations as well as greater angularity.

If the gauge is too tight, the wheelset would strain the track fastenings and, in the event of failure of fastenings, the rail may either move outwards or may over-turn or in the event of the track holding, the wheel may lift off the rail.As per ORE C-9 Report, " Gauge has very definite effect on the period of nosing. Lateral force at axle bearing increases markedly with the flange distance (play) and running speed. It also increases the angularity of the approach of wheel."

Whatever tolerances are available in the Indian Railways Permanent Way Manual and also those stipulated for speeds higher than 100 kmph on B.G. and 75 km/h on M.G. (see RDSOs Report C & M, 1 and C 138), are only good riding tolerances and not safety tolerances.

A variety of tolerances have been laid down from time to time, for instance, by the 19th Meeting of the IRCA, Recommendations of the Efficiency Bureau Study No. 4/64 on Mid-section Derailments on Central Railway, Railway Boards letter dated 10-11-1964, etc. However, as already mentioned, these have not been fixed based on any derailment tests and as such have obvious limitations so far as investigation of derailments is concerned.

7.2.2 CROSS LEVELS

Just like the gauge parameter, a uniform defect in cross levels but within practical limits does not have any adverse effect on safety, stability or comfort.

Here again, what are the practical limits for safety are not known. The tolerances which have been laid down from time to time for this parameter are for good riding and are not safety tolerances. This parameter too, therefore, has to be discussed in general terms.

7.2.3 TWIST

This is a crucial parameter. Though a number of studies have been carried out on some of the World Railways and by ORE, it has been found next to impossible to lay down safety tolerances for various types of rolling stock and speeds, the two variables which particularly determine such tolerances.

The only safety tolerances available for twist parameter on the Indian Railways is as per RDSO letter No. CRA/ 501 of 29.04.1983. This letter specifies the slow-down tolerances for unevenness and twist for 4-wheeler wagons (Table 7.2). These tolerances are based on investigation into running of 4-wheeler wagons. Rolling stock with a superior suspension would have a larger tolerance than the 4-wheeler wagon.

Table 7.2

Speed (Kmph)	Peak value of unevenness on 3.6 m (mm)	Peak value of twist on 3.6 m (mm)
75	14	13
60	16	15
45	22	22
30	24	25
15	33	30

As per the above, the tolerance for twist for a 4-wheeler wagon at the slow speed of 30 kmph is approximately 7 mm/ m. UIC studies (ORE B/55) have suggested, for a 4-wheeler goods wagon at slow speed, a safety tolerance of 7% is for 5 m base of measurement. If within this 5 m, the base of measurement is 3 m, the tolerance would be 10% (i.e. 1 in 100).

In other words, 1 in 144 (or 7%) safety tolerance would be for a vehicle with 5m wheel-base. For a smaller wheel-base a steeper twist can be tolerated e.g. 10% (1 in 100) for a 3 m wheel-base.

For sectional derailments, where speed are higher, each case has to be discussed on its own merits against background of type and condition of vehicle, its speed etc.

It is, however, of utmost importance that twist is calculated over the wheel base of the vehicle which derailed first, as that is the only twist which had actually been effective.

The twist should not be calculated over the interval of measurement of cross-levels, which are recorded at every sleeper or 3 m. The cross level measurements only give the track record and from this record the steepest effective twist, which the derailing vehicle encountered, should be found by moving a tracing paper (with the wheel-base plotted) over the cross level record and evaluating the twist at various positions of the wheel-base (see **Fig.** 7.4). The steepest effective twist can then be known.

Following illustration will clarify the premise.

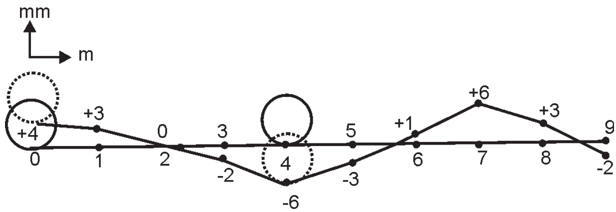


Fig.7.4 Evaluation of twist effective for a particular wheel-base.

Illustration : Say, wheel base of vehicle which derailed first = 4 m. Record of measurement (when the left rail is considered as base):-

Station	*Cross level (mm)*	*Twist over base of measurement (mm/m)*	*Effective twist over 4m i.e. the wheel-base (mm/m)*
0	+ 4		
		 1	$\frac{10}{4}$ = 2.5
1 m	+ 3		
		 3	
2 m	0		
		 2	
3 m	- 2		1.5
		 4	
4 m	- 6		
		 3	
5 m	- 3		
		 4	0.25
6 m	+ 1		
		 5	2.0
7 m	+ 6		
		 3	2.25
8 m	+ 3		
		 5	0.25
9 m	- 2		
Max. value :		**5 mm/m**	**2.5 mm/m**

It will be seen that though the maximum twist as calculated from the record of measurement is 5 mm/m (base is 1 m), the maximum effective twist which the vehicle actually encounters is only 2.5 mm/m (base is 4m).

7.2.4 VARIATION IN ALIGNMENT

Here too a number of tolerances are available on the World Railways as well as on Indian Railways but all are for good riding comfort. Each case, therefore, has to be discussed on its own merits.

Alignment defect results in unbalanced dynamic lateral forces. This unbalanced lateral force, in combination with cross level defects, has much larger cumulative effect on safety, compared to either of the defect alone.

Lateral forces may also be generated in a cyclic pattern, due to lateral oscillations. These forces may be severe in case of hunting of vehicle resulting from cyclic track irregularity, high speed etc. This aspect should be analysed carefully in Derailment Investigation. These large lateral forces are capable of inducing misalignment in track, particularly where lateral track resistance is low. These misalignments could grow with traffic, ultimately leading to derailment.

7.2.5 BUCKLING/ DISTORTION IN TRACK

When the track alignment gets affected due to excessive thermal forces in rails, which cannot be contained by the inherent strength of track, the track is said to have buckled. On the other hand, if the track gets out of alignment due to excessive flange force, which the track cannot resist, the track is said to have distorted. Serious controversies arise at the accident sites on this aspect, mainly because the evidence of track such as closed expansion gaps and poor lateral strength get destroyed in the accident. The following important particulars must be recorded correctly at site:

a) Alignment of track in the rear of point of derailment (offsets)

b) Creep in rear and advance of the site of accident

c) Location of expansion gaps w.r.t. joint sleepers

d) Condition of sleeper packing

e) Rail temperature while measuring expansion gaps

f) Length of track between reference points (If the measured length is more than fixed length then track is buckled).

g) Ballast cushion (especially clean cushion)

h) Behaviour of formation at site (Whether yielding or not)

i) Whether any gang worked at site leading to weakening of track

j) Sleeper spacing (irregular or not) and expansion gaps in switch expansion joints in long welded rails.

k) A thorough probe on vehicles defects which can lead to excessive flange forces.

7.2.6 UNEVENNESS AND LOW JOINTS

These cause wheel offloading. But what are the safety tolerances, is a question which has not been satisfactorily answered.

More often than not, it may be the resonance effect resulting from unevenness occurring in a repetitive wave-like form, which combined with inadequate damping, may be the cause of offloading to unsafe levels, (see Chapter 1, Vehicle Oscillations). If the defect is an isolated one, even a comparatively large defect can be tolerated.

As is evident from field trials reported in RDSO Report C & M-1 (Feasibility Studies for 120 kmph operation) even a larger unevenness at a joint does not produce appreciable offloading of wheel. In fact, the unevenness value for high speeds has to be restricted more from consideration of deterioration of the joint and its fittings, joint packing etc, rather than from consideration of wheel off-loading.

We will now discuss some of the special track features.

7.3 CURVES

Curves, by their very nature, are more derailment-prone than straights. For one, on a sharp curve, axle angularity tends to be high and this angularity persists throughout the curve negotiation.

On the transition portion, the inherent twist due to cant gradient reduces the margin by which irregularities in twist parameter could be allowed to occur, thus needing more frequent attention (see Chapter 1, Effect of Track or Vehicle Twist on Wheel Off-loading).

The outer rail suffers angular wear, which may result in reduced flange safety depth and reduced effective flange slope.

Besides, vehicle design and its condition have considerable relevance to safe curve negotiation; for any hindrance to rotation or rotation and sliding of bogie or truck designed for such movement on curve would result in appreciably high flange forces, which might cause track distortion or lead to flange climbing derailment through high Y/Q ratio.

In a curve, large flange forces may arise due to unsatisfactory curving characteristics, unsatisfactory riding of the vehicle, misalignment in track or misalignment of vehicle/ bogie wheel base.

Majority of railway vehicle have unsatisfactory curving characteristics. Ideally, vehicle should negotiate a curve almost without any flange contact, steering being achieved by axles assuming radial position under the effect of steering forces generated on the wheel trade. However, most of the railway vehicles have a very low YAW flexibility and, as a result, steering around a curve is achieved by large flange force on the leading outer wheel, under the condition of a large angle of attack. This effectively reduces the limiting safe value of (Y/Q) on the leading outer wheel. Further, greater angle of attack is generated on the outer wheel of leading axle under conditions of cant excess (and vice-versa for cant deficiency), as cant deficiency results in the trailing axle moving outward, thereby reducing the angle of attack.

Variation of versine in a curve can be evaluated in terms of variation from the designed versine or station to station variation. The former is related to unbalanced centrifugal force, affecting safety and curve negotiability (as discussed in the para above). The latter relates to rate of change of lateral acceleration, affecting the comfort level. Generally on Railways, including the Indian Railways, the guiding factor is the rate of change of lateral acceleration and, hence, station to station variation of versine is stipulated for maintenance. Calculations show that this variation is important only at higher speeds. It is irrelevant for speeds lower than 60 kmph and important for speeds higher than 90 kmph.

On the Indian Railways, criteria for re-alignment of curve is stipulated in IRPWM para 421. Therein, service limit of Station to station variation of versine has been stipulated. Thus, the criteria is based on comfort requirement and the same is mentioned in the para also.

As lateral forces, in general, are higher in a curve as compared to a straight, loss of lateral strength of track should be guarded against.

A vehicle entering a curve at a speed higher than the maximum permissible may cause distortion of the track or lead to mounting of the wheel over outer rail. Mounting over the inner rail is also possible if there is sudden braking causing the vehicles to bunch, or if there is a jerky bogie rotation.

Another seemingly strange derailment has been often reported viz. of a wagon when just starting to move after having stopped on a sharp curve, particularly with high superelevation.

When the train starts on a sharp curve, a net component of the tractive force is generated towards the centre of the curve at the coupling level, due to string lining of the wagons. This causes off-loading of the outer wheels. The amount of off-loading being higher for sharper curves and higher acceleration. Further, when a wagon stands on high cant, the outer wheels get off-loaded substantially (the whole cant acts as cant excess).

Also, when the wagon starts moving, even at such crawling speed, the rail guiding force acting on the outer leading wheel will be significant as this will be the force required to turn the vehicle (bogie) along a curved path. The Y/Q ratio for the outer leading wheel, thus, becomes quite adverse and the wheel mounts the rail, or so to say, just rolls off the track.

Derailments ocuring under the conditions as above, should be investigated keeping in mind the above factors and analysing the extent of sudden acceleration and the features of rail – wheel profile.

7.3.1 CHECK RAILS ON CURVES

Check rails are sometimes provided as one of the measures against derailment proneness on sharp curves.

Other advantages claimed for check rails are:

- reduction of lateral and angular wear of outer rail
- increase in stiffness of track frame and, thus, in the lateral strength of track against distortion.

Check rail, however, is not a solution which can be applied indiscriminately. A popular misconception is that the check rail shares the curving forces with the outer rail. In fact, for any given wheelset only if the flange thickness, wheel gauge and check rail clearance are exactly matching-a rare possibility- will the curving forces be shared by both the outer rail and check rail.

The above parameters are linked by the following expression: Check rail clearance = Track gauge* -(wheel gauge+flange thickness) + allowance for angularity of axle

(***N.B**. : This is the gauge actually provided i.e. including any gauge widening).

Though track gauge and wheel gauge may be taken, more or less as fixed or of known values, flange thickness is certainly not. Flange thickness for various wheel sets can vary over a wide range. This presents a problem, how to calculate the check rail clearance. For this, normally the predominant average flange thickness is taken viz. somewhere in between a new flange thickness and a thin flange.

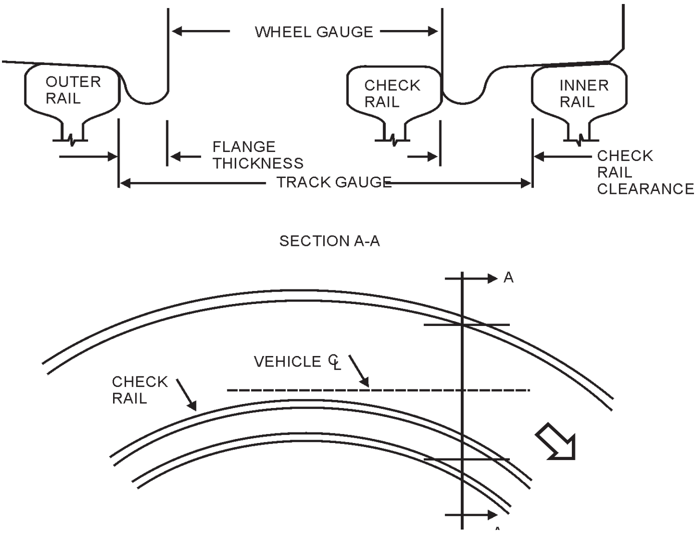


Fig. 7.5 Check rail on curve

Taking an example on B.G.	
new flange thickness	= 28.5 mm
thickness of thin flange	= 16 mm
Assume average flange thickness	= 22 mm
Track gauge (assuming 6 mm gauge widening)	= 1676+6
	= 1682 mm
Wheel gauge	= 1600 mm
Check Rail clearance	= 1682 - (1600+22)
	= 1682-1622
	= 60 mm

Allowing 4 mm for angularity of axle, Check rail clearance works out to= 64 mm, say 65 mm.

Now, when a wheel with flange thickness more than 22 mm negotiates the curve, only the outer rail will bear the guiding forces and the check rail will not come into play at all. When flange thickness is less than 22 mm, only the check rail will guide the wheelset and outer rail will develop no flange force.

In case the check rail is intended only as a safeguard against derailment it need not be called upon to perform the guiding function. In that case check rail clearance can be increased to such a limit that only when the outer wheel mounts the rail should the check rail come into play and help in forcing the wheel back to the track.

For this the only criterion is minimum bearing of inner wheel on the rail, which may be taken as 40 mm.

7.4 POINTS AND CROSSINGS

Considering the discontinuities inherent in a turnout, it forms a weak link in the track structure.

Care in laying and maintenance of turnouts is thus crucial. Some of the aspects are highlighted below:

(i) Gauge at approach to the switch portion of a turnout should be given particular attention and should be maintained to finer tolerances, so that the vehicle sets into more or less a stable motion before negotiating the discontinuity in the form of switch portion.

(ii) The gauge opposite to nose of crossing needs equal care. If it is tighter than the nominal value, there would be danger of a new wheel hitting the crossing nose (see Chapter 3, Expression 3-2).

(iii) Similarly, check rail clearance opposite the nose of crossing should be checked frequently, as an excessive clearance might cause a new wheel flange to hit the nose of crossing.

(iv) Turn-in curve following a turnout, especially on running loops, should be given the same attention as that for main line curve, i.e. the frequency and thoroughness of inspection and rectification should be the same.

It is a good practice to provide the same track structure for turn-in curves as that for main line curves, or even to provide extra ballast shoulder say 150 mm on the outside of turn-in curves.

(v) Angular wear at the tongue rail and the lead curve should not be allowed to become large so as to affect safety.

7.5 GIRDER BRIDGE (UNBALLASTED) AND LEVEL CROSSING APPROACHES

Girder bridges and level crossings provide far more rigid supports than ballasted soil formation. The sudden change in track modulus at their approaches sets up oscillations in the vehicle, which apart from increasing the wheel off-loading, increases the dynamic augment at the approaches as well as on the bridge. This increase rate of track deterioration.

Such approaches, therefore, need more frequent attention.

7.6 SAFETY AT WORKSITES

One of the major causes of derailment/accident on track account is inadequate care taken at workspots. Few instances are listed along with the required precautions as follows:

a) Sinkage of track on diversions/new formations/supports of temporary arrangement, especially during rains, leads to derailment. The track geometry should be maintained to adequate standards for passage of traffic at stipulated speeds. Where abnormal behavior of formation/supports is expected, track geometry and effectiveness of packing/ support should be checked as often as warranted even during night time.

b) Derailment takes place due to failure to pack the track after lifting the same here and there for adjusting the levels during maintenance, if speed of train is not restricted. The gang staff should be counselled in this respect.

c) At deep screening and relaying spots, if the top-table and gauge face of rail are smeared with ballast particles, the enhanced friction at flange contact area can encourage mounting of the wheel, especially with empty wagon, sharp flange, angularity of axle, difference in buffer heights and bad enginemanship. This kind of derailment can take place even at very slow speeds. Before the passage of each train at such workspots, the running surfaces of rail should be cleaned. It is also desirable to hand-grease the gauge face.

d) Imposition of speed restriction with duly displayed SR boards needs care. Non-provision of suitable SR boards as required and/or sudden display of STOP (red) signal by gang staff/stationary watchman can lead to unsafe conditions. The positions of SR and Caution Boards should be correctly decided, duly ensuring proper visibility during day and night. Full details should be furnished in the caution orders issued to the drivers, indicating the correct kilometrage of the SR and Caution Boards. The description of the work requiring speed restriction should be given in a brief and simple manner in the caution order so that the driver fully understands the implication. Working of materials lorry on the track requires all the stipulated precautions being observed.

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CHAPTER 8

OPERATING FEATURES

Train operation has a crucial bearing on safety. Skill of a driver in safe operation of train is critical. Apparently simple functions like observing aspects of signal and following speed restrictions need a lot of skill. The challenge for the driver is that the train is to be run at the maximum possible speed, in order to achieve punctuality. Features, such as track gradient and curve, speed restriction on track, frequent train stoppage and acceleration etc. should be handled deftly by the driver, otherwise, safety may be affected.

8.1 Slack

Slack is the free play provided in draw-gear and in some draft-gear. It is required for movement around curves and grades.

Total slack consists of Free slack and Spring slack. Free Slack (or loose slack) is the clearance within the draw-gear, which can run-in or run-out, without compressing / stretching the draft-gear. Its value is upto 1 inch for a wagon. Spring Slack is the additional longitudinal movement that can occur after free slack movement is finished, and when draft-gear is compressed or rebounds directing all slack in opposite direction. Its value is upto 5 inches, when draft-gear is fully pressed. Thus, total slack for 40 vehicles is about 20 feet.

‘Run-In’ is the rapid change of the train’s coupler-slack to buff (compressed). It may happen when rear section of a train travelling at a faster speed bumps against the front portion of the train travelling at a slower speed, due to sudden braking.’ Run-Out’ is the rapid change of a train’s coupler-slack to draft (stretched). It may happen when rear section of a train travelling at slower speed stretches against the front portion of the train travelling at faster speed, due to sudden acceleration.

‘ Slack Action’ is the movement of one part of a coupled train at a speed different from another part of the train, causing run-in or run-out.

One of the key factors in limiting longitudinal coupler forces is the ability to control the slack action within the train. Relatively low forces will result if the coupler slack is taken-up one wagon at a time; however it is not possible with longer trains on undulating grades.

It is important to prevent the relative movement of large groups of wagons that behave as single mass. High impact forces result when two blocks of bunched (or stretched) wagons separated by a number of stretched (or bunched) wagons move together (or apart) due to differences in speed along the train. When all slack between the two separate blocks of wagons is used up, the resultant run-in (or runout) can easily cause longitudinal coupler forces in excess of the permissible limit.

Maximum coupler forces arise while negotiating undulating graded section. Its severity can be mitigated by skillfull driving. Driver should reduce power slightly just before approaching top of the hump, to avoid subsequent excess speed and use of severe braking. Similarly, power should be increased slightly just before approaching the dip.

8.2 Train Brake Application

Following are the different system of brake available in a locomotive:

Pneumatic Train Brake – It applies throughout the train, including the locomotive. It is controlled by A9 valve.

Pneumatic Loco Brake – It applies only on the locomotive. It is controlled by SA-9 valve.

Regenerative Brake – It is the Electrical or Dynamic Brake. Braking is achieved by making the traction motor work as a generator. This is done by moving the Master Controller towards Braking (from Traction mode). It applies only on the locomotive.

Parking Brake

Application and release of train brakes has been found to be a key factor in many of the recorded train partings and knuckle failures. A train brake application usually results in compressive coupler forces. However, much of the severity of the resultant run-in depends on the force and slack distribution within the train prior to the brake application. The wagons may or may not be uniformly loaded and, hence, wagons may brake differently to each other.

A longer time is required for release or build up of brake cylinder pressure on the tail end vehicles. As a result, head end vehicles tend to decelerate faster than the tail end vehicles. Greater longitudinal dynamic stability is achieved when heavier vehicles are placed closer to the locomotive and the lighter ones are placed the farthest, as the likelihood of run-in gets reduced.

Use of brake on locomotive only (Locomotive brake or Dynamic brake) can result in a severe slack action. Run-in is highly possible if application is at an inappropriate time (with respect to track geometry and train speed) and, if released at an inappropriate time, it can result in a run-out.

8.3 Wheel Off-loading due to Braking and Tractive Forces

Effect of acceleration (or deceleration) of a vehicle on vertical wheel load can be analysed by considering a pseudo force acting longitudinally at the centre of mass, in a direction opposite that of the acceleration. Magnitude of this pseudo force would be proportional to the magnitude of acceleration, being equal to the product of mass and magnitude of acceleration. In case of severe buff or draw action on a vehicle, longitudinal forces would be predominantly transferred at the buffer/ coupler level, compared to that generated at the rail-wheel contact. Thus, effectively, a couple would be generated by equal and opposite longitudinal forces acting at the centre of mass and buffer/ coupler levels. This mechanism is illustrated in Figure 8.1.

The above couple results in generation of additional vertical forces at the wheel. In case of braking, rear wheels get off-loaded, whereas, tractive forces off-load the front wheel.

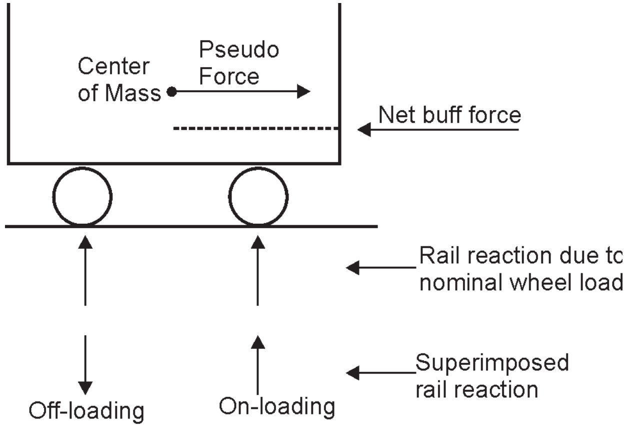

Figure 8.1 Off-loading due to Buffing Force

8.4 Effect of Curvature

Buff/ tractive force on a vehicle (which act from both front and rear ends) has a resultant in the lateral direction on a curve due to string lining effect. Magnitude of this resultant force increases with increase in curvature and magnitude of the buff/ tractive force. Therefore, special care is required to be observed while negotiating sharp curves, particularly curves of four degree or sharper.

Uncontrolled run-in or run-out of slack in a sharp curve may result in excessive lateral forces and derailment. Retardation by locomotive or dynamic brake must be avoided, if possible, in a sharp curve. Speed of the train should be reduced to the permissible speed before entering a curve. Similarly, extreme care should be taken while starting a train in a sharp curve. Only sufficient throttle should be used to start the train. Any advance of throttle must be made one notch at a time, to avoid excessive inward lateral forces.

8.5 Marshalling of the train

A train is more prone to derailment at the junction of a set of loaded and empty wagons owing to difference in the buffer heights and consequent eccentric transmission of buffing forces, which causes wheel off-loading (see Chapter 3, Buffers).

If an empty wagon is marshalled between two loaded wagons, and subjected to sudden brake application, the empty wagon owing to inertia of loaded wagons is gripped hard, which makes it possible for the empty wagon to travel in an off-loaded condition without possibly even making contact with the rails. A relatively small lateral flange force during this stage would increase the derailment proneness considerably.

8.6 Movement of 3-axled bogie on sags and humps

On a sag, there is tendency for the middle axle of a 3-axled bogie to offload, whereas, on a hump, the outer axles tend to offload.

On humps, off-loading would further increase with speed due to the action of centrifugal force.

8.7 Wheel slip on diamond crossing

In diesel or electric rolling stock in which axles are powered by axle-hung traction motors, the torque at starting of the motor produces a yaw or nosing action in the wheelset, owing to eccentricity of motor masses. In other words, the wheel slips laterally.

This effect is more marked in the motor coach of suburban

EMU stock, which has lighter axle load compared to that of electric locomotive.

If the motor coach of an EMU rake, therefore, stops on a diamond crossing and then starts, there is a clear possibility of the wheelset taking two roads on the crossing owing to lateral slip of the wheels, as the diamond crossing already suffers from disadvantage of very small length of check rail guidance opposite the crossing nose.

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CHAPTER 9

SUMMARY AND CASE STUDIES

Derailments account for a very large share of accidents on the Indian Railways. In the year 2009- 10, total number of consequential train accidents on the Indian Railways was 165, which is 0.17 accidents per million train kilometers. Out of this, derailments accounted for 80 (48.5 %) accidents. Other major category of accident contributing to the number was accident at unmanned level crossing, 65 (39.4 %). In the same year, derailments have resulted in 105 casualties (out of a total of 635). Thus, derailments are a major source of accident and loss of life/ injury/ property. (Data source – Railway Board/ Safety Directorate Website).

9.1 SITE INVESTIGATION

Importance of a systematic and thorough investigation at the site of a derailment need not be emphasized, as the ability to arrive at the most probable cause or causes of derailment depends largely on it.

It is necessary to proceed to the site as quickly as possible, not only for reasons of protection of track, rescue, first aid and restoration, but also for collection of all possible evidence before it is tampered with or destroyed willfully or otherwise.

Causes leading to derailment generally disappear within the very course of derailment. A defective track gets damaged and disturbed. The defective parts in rolling stock get displaced or get broken. Sometimes, it is difficult to establish whether a certain breakage is the cause or result of the accident, unless there are clear indications that the breakage is an old one. Sometimes, the officials first reaching the site of accident are not objective in their approach and see through departmental bias. Further, the officials of a department may be ignorant of weaknesses of a sister department. The result is that derailments are clouded by controversies and treated as mysterious events. But the fact is that derailments are a scientific phenomenon and causes can be scientifically established, provided relevant evidence is available and necessary expertise is developed to analyse that evidence.

9.2 VARIOUS CAUSES OF DERAILMENT

Derailments may be attributed to one or more of the following causes:-

9.2.1 OBSTRUCTION OR DISCONTINUITY OF TRACK

Obstruction may be in the form of a boulder falling on track , a heavy component of a vehicle falling on track or a heavy article (like a piece of rail or PSC sleeper) kept on track as an act of sabotage. Discontinuity in track may be caused by an act of sabotage or caused by formation failures, usually caused by natural calamities such as floods/ breaches etc.

Normally, such causes are obvious and no disputes arise in establishing cause of the accident.

9.2.2 OVER SPEED

Marginally excessive speeds on straights should not cause an accident, unless a train is coasting a down grade. But excessive speed on turnouts/sharp curves may lead to derailments due to excessive flange forces, as centrifugal forces are proportional to the square of speed. Also, excessive speed tends to augment lateral and vertical oscillations, leading to excessive flange forces/ wheel off loadings, and may also gives rise to resonant parasitic motions of rolling, pitching and bouncing.

Over speed has to be established with the data from speedometer. There are other evidence in the form of train timings being maintained by train passing staff and train crew. Nowadays, reliable train timing can also be obtained from analysis of data logger report.

9.2.3 OVER TURNING OF VEHICLE

Vehicles can overturn due to heavy wind pressure and also due to centrifugal force effect in sharp curves. An adverse combination of unbalanced centrifugal force with wind pressure would be even more dangerous.

This explains as to why on MG and NG, the mid-section derailments are more than that on B.G. H/G ratio (height of C.G. of vehicle/track gauge) is higher on MG and NG vehicles, as compared to B.G. vehicles.

9.2.4 INJUDICIOUS CONTROL OF TRAIN

Sudden notching up of diesel/electric locomotives, or sudden application of brakes may result in jerks which may cause derailments, particularly when the trains are moving on sharp curve and coasting down a gradient or when couplings are loose. For detailed analysis of the mechanism, please refer Chapter 8 on 'Operating Features'.

9.2.5 UNEVEN LOADING

This leads to unbalanced load on wheels, resulting in some of them running light. This is caused by factors such as shifted consignments and excessive loading on one side of the wagon etc. Certain wagons are more prone and sensitive to irregular loading. Similarly, certain consignments (such as steel coils) are more prone to shifting. Steel coil consignment are to be secured as per standard drawing.

Position of load is an important item to be observed in derailment investigation. Uneven loading also gets correlated with uneven buffer height and spring compression.

9.2.6 DEFECTS OF SIGNALLING AND INTERLOCKING

Derailments can occur on points & crossings if the interlocking is defective and signals can be taken off without tongue rail housing properly. Wheels can take different routes if there is a split in the switch.

There have been a large number of derailments on the above account. Some of these derailments have also occurred due to manual interference, wherein signal maintainers have deliberately by-passed the interlocking in order to avoid detention of trains resulting from failure of the system. Careful interpretation of data from data loggers would present a very strong evidence in analysis of such accidents.

9.2.7 DEFECTS IN TRACK

Various defects of track and their role in derailment has already been discussed in Chapter 7 on ' Track Defects'.

9.2.8 VEHICLE DEFECTS

Common defects of vehicles have already been discussed in Chapter 3. Specific defect of a vehicle would depend on whether it is a wagon/ coach/ locomotive and its special construction. These have already been discussed in the respective Chapters.

9.3 CASE STUDIES

The case studies selected include accidents to mail express/ passenger trains as well as goods trains and have been taken out from inquiries done by CRS as well as by Railway Administration. They have been chosen mainly to highlight the cause of derailment due to various factors discussed above. They are only illustrative and by no means exhaustive. It is hoped that a serious study of the above accidents would not only create an awareness of the complexities of a derailment phenomenon, but also motivate an analytical and scientific approach towards derailment investigation.

9.3.1 CASE STUDY NO.1

Title

Derailment of A-14 Up Ambernath-Bombay V.T. suburban train at km.41/12-10 between Diva and Mumbra stations on the Mumbai V.T.- Kalyan suburban section of Bombay Division of Central Railway at about 08.31 hrs. on 15.05.1992.

Description of the accident

At about 08.31 hrs. on a clear morning of 15.5.92, the South bound A-14 Up Ambernath-Mumbai V.T. suburban train derailed on the Up local line at Km.41/12 between Diva and Mumbra stations. The train consisted of 9 electrical multiple unit coaches (3 units of 3 coaches each, coupled together), of which 5 coaches went off the rails. While the first 4 coaches remained on rail, the 5th to 9th coaches derailed. In the 5th coach the bolster of the leading trolley at the left hand side (in the facing direction) had fallen down due to its bolster hanger and its fittings broken and fallen while on the run. This was dragged till it got obstructed and stuck up in the V rail of the diamond (connecting down local to down through) and the left hand side wheels following it rode over it and jumped off the track which resulted in the derailment of the 4 rear coaches at km.41/10-12. It may be mentioned that the bolster bottom spring plank started working out at km.44/6-5 gradually (as seen from the hit marks on the track made by it) and the spring plank gradually developed instability and ultimately fell down after passing Diva. The grazing and rubbing marks are available on the rail table about 165 m in the rear from the place where the bolster spring plank got entangled. After the derailment, several undergear parts of the bogies such as springs, pins, hanger blocks and shock absorbers had worked out and lay at various locations.

Cause of the accident

The obstruction caused to the movement of the above local by the bolster assembly of the motor coach No. 70234 (5th from Mumbai end) having fallen from the coach due to its broken/ missing hangers. The accident could have been averted if: (a) The material used for the hanger was made of class I steel to IS

1875/78 and the manufacturing process was as per RDSOs specifications; and (b) The hanger had been replaced during its POH in March/April, 1991 at MTN shops.

Accordingly, this accident falls under the category of Failure of Equipment-Mechanical (under gear of coach) combined with Failure of Railway Staff.

Reasons for arriving at the cause of the accident

The condition of permanent way in rear was reasonably good. From the joint observations of the track in rear of the point of derailment, fresh grazing marks and hit marks were noticed on OHE jumpers and impedance bond cables. Grazing marks were also noticed on some of the fish plates in the immediate rear of the point of derailment.

At the site of derailment viz. km. 41/10-12, the first grazing marks were seen at about 165m from the point where the bolster spring plank had dropped and got bent/ twisted under traffic. At about 28.4 m from the first rubbing marks, a pin 9 cm long was found between the turn out point and check rails. From this point till a point at 156.70 m there were several hit/grazing marks and a spindle was seen lying at about 22cm outside the gauge face of the turnouts. A broken hanger was lying at 165 m from the 1st grazing marks and check rail broken at about 185 m distance.

It became apparent from the remarks given above that some metallic part under slung on the left of the coach (bolster assembly) had got dislodged from its intact position at Km.44/6-5 and had made contact with the various track/ line components on the left rail of the up local line. It also became clear that the process of derailment had started about 165 m beyond the V of the diamond crossing connecting the Up and Down local lines.

It was noticed that on the 5th coach, out of the 16 hangers from 2 bolster assemblies there were only 5 nos. in position, one of the inner right hand side bolster plank on leading trolley and 4 nos. on the trailing trolley. Out of 11 nos. of hangers, 6 have been obtained in unbroken condition, 4 nos. in broken pieces and one could not be traced out from site, inspite of vigorous search for the same.

The sequence of events was perhaps as follows:

Out of 8 hangers in the leading bolster assembly of motor coach No. 70234, one might have been broken/ missing even at the time of starting the train A-14 Up at Ambernath. With only 7 hangers in service, one more hanger fell out at Km.46/1, after passing the Dombivili station at Km.48/8. It should be reiterated that this train gets heavily crowded at Dombivili due to morning peak hour rush and there is every possibility of uneven/excessive load on this coach which, after travelling a short distance of 2 kms., must have resulted in one more hanger working out and falling out from the bolster assembly.

In the present case, with two hangers having fallen out by the time the train had reached Km. 46/1, the bolster started travelling in a tilted position with the various configurations of track geometry and terrain and by the time it reached km.44/6-5, it had assumed a fully inclined position and had started hitting the track components. The result of the hit marks was also felt by some of the passengers who had given their evidence. After the train had approached the diamond crossing on the Up local line (the crossing is from down local to down through line), the culprit bolster assembly, which by this time was being suspended precariously hit the V rail and due to restricted space, got entangled with this and left spring plank came off and along with it the springs, dash pots and finally the bolster assembly. The spring plank got mauled over by the wheel in the trailing coaches and due to wedging action rendered by it, the wheel took a crossing turn and went on the wrong track (first the 5th one i.e. motor coach which in its trail dragged on the rest of the train), resulting in blocking the down local line as well.

Important recommendations

1. Reduction of maximum permissible speed of DC EMU local from 90 to 80 kmph.

2. POH schedule of EMU rakes, currently at 18 months interval to be restored to 12 months, as was existing prior to 1988.

3. 100% replacement of hangers of assemblies of EMU rakes during POH. While doing so it should be ensured that the new hangers are as per stipulated specification.

4. Quality Control Inspection should be given the due importance and it should be ensured that all rakes leaving the shops after POH would have Nil defect, certified by the Quality Control Inspector.

5. Hangers should be procured only from approved firms

6. A suitable safety strap must be designed for bolster assembly in EMU coaches, which would prevent the falling of bolster assembly, in case of broken hangers/pins.

9.3.2 CASE STUDY NO. 2

Title

Derailment of 3 NHJ Nainpur-Jabalpur passenger train between Shikara and Sukrimangela stations on the Nainpur-JBP N.G. section of Nagpur Division of S.E.Rly on 26.02.89.

Description

At about 15.15 hrs on a warm clear afternoon of 26.02.89, while 3 NHJ Up Nainpur-Jabalpur train was running between Shikara and Sukrimangela, it suddenly derailed at km.1180/2-4 on the outside of a 14^0 curve. The driver found vacuum dropping and train coming to a halt. While the loco did not derail, the first coach after the locomotive derailed of its rear trolley. The next 3 coaches (2nd to 4th) capsized and came to rest on their right sides with the wheels pointing towards the track, about 3 to 6 m from the center of track but remained coupled together. The 5th coach derailed of all wheels of the front trolley, but the wheels of the rear trolley remained on the rail. 6th to 8th coaches remained on rail. The couplings gave way between 1st & 2nd and 4th & 5th coaches only.

Cause of the Accident

Most probably due to a combination of the following two factors :-

(i) Instability caused by the eccentric loading of passengers on the roof and on the right side foot boards of the NG coaches due to heavy over crowding.

(ii) Excessive speed on the 14^0 curve at Km. 1180.

Reasons for arriving at the cause of the accident

(a) There was no evidence of discontinuity of train due to sabotage or any unauthorized interference with the track.

(b) Observations of the under gear of the derailed and un derailed, coaches made soon after the accident did not point towards the possibility of any under gear component having dropped down from the coaches on the run and causing an obstruction. There were certain firewood bundles kept on the roof, which had been carried by passengers, which had also fallen by the side of track. But careful examination of fire wood bundles as well as other eyewitness accounts did not indicate that wheels have ploughed into the bundles viz. there was no evidence of firewood forming an obstruction to wheel movement. The dropping of firewood bundles was attributed due to derailment of coaches.

(c) There was no evidence of rail breakage or joint fracture.

(d) General maintenance of track was quite satisfactory. It was remarked by CRS in his report that it was even better than Broad Gauge section viz. ballasting, full complement of fittings etc. The track geometry parameters measured did not indicate any abnormal variations.

(e) The locomotive did not derail. Further, the loco was fairly new, having been commissioned a year earlier to this accident. The loco measurement indicated that there was no deficiency beyond permissible limit.

(f) Measurements were taken on the affected coaches regarding wheel dia, flange thickness, wheel gauge, trolley frame, load deflection characteristics of the springs. All the parameters were within allowable tolerances laid down in IRCA part IV, except spring deflection of two coaches. These were considered by CRS in detail and found to have no contribution towards the cause of derailment.

(g) Speed of train : The speedometer on the engine (an imported one) had become defective. From the evidences tendered by railway and public witnesses, it became apparent that the speed of train while negotiating the 14^0 curve was excessive. Many of the public (who were regular travelers on this train) stated that train was proceeding at much higher speed than on normal days. From the working time table, it was seen that there was a permanent speed restriction of 25 kmph on 14^0 curve. It was assessed by CRS that the speed was about 40 to 45 kmph (assessment based on time distance basis).

According to para 405 (c) of IRPWM 1986, safe speed on NG on curves is given by formula v = 3.65 (R – 6)1/2, $V=3.65\sqrt{R-6}$ where R is radius in metres. For 14^0, this comes to 39.8 km/h.

However, calculations had been made theoretically to determine the speed at which an NG coach will overturn on the outside of a 14^0 curve, when loaded with eccentric loading of passengers on the roof as well as right foot boards of the coach, taking into consideration tilt of coach due to springs, eccentric shift in C.G. of coach due to centrifugal force and stabilizing movement due to weight of coach. These were done with the help of RDSO officials. As per this, an NG coach loaded with eccentric loading by passengers traveling on the roof and on the right foot board is liable to overturn on the outside of a 14^0 curve, when speed exceeds 37 kmph. (The dynamic augment assumed is 15% in this case). Similar calculations were also made by the S.E.Rly., without considering the spring deflection, but taking into account the actual tilt of the coach as observed at site. Safe speed works out 39.7 kmph, with 50% dynamic augment and 35.5 kmph at 100% augment. Hence, the above calculations clearly prove that N.G. coach is definitely susceptible to overturning on 14^0 curve when eccentrically loaded with passengers on the roof and on foot boards towards the outside of the curve, if the speeds are above 35 to 39 km/h.

(h) From the various evidences there was no sudden application of brakes by the driver.

(i) Did the coaches overturn?

It is well known that N.G. coaches are inherently unstable owing to the narrowness of the guage and considerable overhang of the coach body beyond the wheel gauge. Any factor such as passengers traveling on roof/ foot boards combined with over speeding could definitely make the coach more unstable. In addition the following factors noticed at site were also taken note of:-

a) absence of any wheel mark on the sleepers at or near the point of jump.

b) Hit marks on wooden sleepers by axle boxes of coaches which were clearly visible and later verified by actual measurements. There was no evidence to establish the occurrence of any wind/storm, leading to overturning of the coaches.

(j) It was also pointed out by CRS that the train did not derail in the earlier 14^0 curves negotiated, mainly due to their being reverse curves on rising gradients, when the driver would have been more cautious. This was the first curve on a continuously down grade up to 1 in 80, where most probably the driver must have become less vigilant.

Important Recommendations

I. Augmentation of services on Sundays & market days to reduce roof/foot board traveling.

II. Springs of coaches (N.G.) to be subjected to scrag tests more regularly to ensure safety standards.

III. Speedometers on N.G. locomotives to be kept in order.

IV. Speed restriction of 25 km/h mentioned in Rly.Boards letter no. 79/Safety (A&R)/1/13 dt.14/ 7/ 80 on N.G. trains when efforts to prevent roof traveling on N.G. coaches fail, should be implemented.

9.3.3 CASE STUDY NO. 3 (This summarizes 3 accidents of identical nature.)

Title

1) Derailment of 13 wagons of Itarsi-Uijjain goods train between Budni & Midghat section of Central Railway (Bhopal division) on 15.02.92.

2) Derailment of one wagon of KSAG Up spl. Goods train between Asvali & Ghoti stations on Bhusaval division of Central Railway on 08.05.92.

3) Derailment and capsizing of 12 wagons (10 BFRs & 2 BRHs) of KSAG spl. Goods train on Mund Bridge between Paras-Nagijhan stations on Bhusawal division of Central Railway on 15.12.93.

Description of accidents

In all the above three cases, the wagons which derailed and capsized contained M.S. sheet rolls/bundles which got shifted due to improper packing/lashing causing imbalance in the under gear suspension. In all the above cases, the steel plant loading point in-charge (commercial and C&W) were held responsible.

The following instructions contained in DG/RDSO/LKOs circular No. MW/ACT/BG of 11.03.75, regarding packing conditions were not adhered to:

Method of packing and loading of steel sheet rolls

Loading and packing of sheet rolls has been depicted in Drawing SK 74600 (Annexure-IV). Main principles of packing rolls is that it should be tied with at least three steel bands with protection angles at the edges to avoid its opening. Also, if more than one in number is loaded, common steel bands should be used as shown in the sketch. The rolls should invariably be loaded on wooden cradles of the shape given in this drawing. Proper wedges of size should be used to avoid the movement. Rolls can be loaded either two or more in one cradle as depicted, keeping in view that the load is more or less uniformally distributed over the wagon floor.

Preparation of wooden cradle/frame work and method of loading

First of all, wooden longitudes are jointed to the end-cross members with the help of vee-blocks. It should be observed that longitudes are placed in line with the end stanchions as far as possible. Now the cross members are fixed to form the framework. After loading and tying of the rolls, wedges are fixed in position with the help of nails. Finally, the end and side connecting pieces are fixed to make the wooden structure of the cradle rigid. Thus the longitudinal movement of the rolls is prevented by wedges and the sideways movement by the side connecting members. Cradles of different sizes may be formed at site depending upon the number of rolls packed together.

9.3.4 CASE STUDY NO. 4

Title

Derailment of 2615, Grand Trunk Express between Jiron and Lalitpur stations on Central Railway, Jhansi Division on 11.05.92.

Description

On 11.05.92, GT Express stopped at 15.35 hrs at Km.1036 between Jiron and Lalitpur on Jhansi division of Central Railway. The guard of the train reported from site that buffers of last but four coaches were entangled with each other but all coaches were on rail. Subsequently, decision was taken to move the train in the same condition to Lalitpur at slow speed in order to clear the section. While doing so, one of the affected coaches with entangled buffer (last but two) derailed by trailing pair of wheels of leading trolley at Km. 1037/6. Inspection of the track on the section immediately after the incident revealed that the track at km. 1034/22-24 was buckled.

Cause

The SA Grade Inquiry Committee after examination of the site, affected coaches, examination of staff and evidence on record concluded that the cause for the incidence of entanglement of coach buffers was buckling of the track.

Reasons for arriving at the cause of the accident

(a) Both the driving crew stated that while approaching km. 1034, they observed abnormal condition of track ahead. The speed of the train was 70 to 75 kmph. (This was also confirmed by the speed recorder chart). The driver started controlling the train speed. He, however, felt sideways jerk as the engine passed km. 1034/20-30, when the train speed was 50 kmph. The train finally came to a halt with the engine at km.1036/0.

(b) Buffers of last but one coach were entangled, but the coaches were on rail. On last but fourth coach, leading buffers had dropped, with fresh shear marks on buffer bolt threads. One buffer bolt was missing.

(c) Engine with front 17 coaches were totally unaffected. Subsequent examination of affected coaches revealed that there was no abnormality or defect in the coaches that could have led to the entanglement of the buffers.

(d) The track at km.1034/22-24 before the approach of train was having sun kink distortion. During the passage of train, when the last 5 coaches were passing over this stretch, the track buckled in S shape curve which resulted in severe lurch to last but four coaches and consequent entanglement of buffers.

(e) Shape of the buckled track is given in **Fig.** 9.1

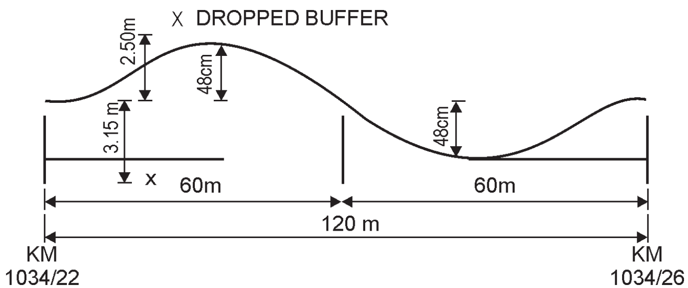


Figure 9.1

As can be seen from the above, the S curve spread overe 120m length with maximum versine of 48cm on 60m chord

(f) Track structure was 52 kg rails (1972) laid on MBC sleepers in March, 92. The actual sleeper renewal at affected spot was done on 05.02.92 to 06.02.92 and at average rail temperature of 30^0 C, The rail temperature at time of buckling was 61^0 C. The gaps at SEJs after buckling were:

	Bombay end	Delhi end
Left Hand	65 mm	78 mm
Right Hand	49 mm	70 mm

(g) Reasons for buckling

Through sleeper renewal of track at affected location was done on 5/6 February, 92 at the average rail temperature of 30^0 C, which was much lower than the distressing temperature range of 37^0 C to 42^0 C. On the date of unusual occurrence, the rail temperature was 61^0 C.

Thus, the difference of temperatures between actual rail temperature and average laying temperature was as high as 31^0 C. This fact, coupled with inadequate shoulder ballast width of 20 cms only (against required 35 cms) resulted in sun kinks and subsequent buckling during passage of train.

(h) Subsequent derailment of last but two coaches was due to buffer entanglement, which generated adverse ride thrusts on the affected coaches.

Recommendations

(i) There is quite a large deficiency of shoulder ballast on LWR/CWR stretches which need to be made good urgently. This was due to poor turnaround of ballast hoppers from depots.

(ii) Heavy arrears of distressing of 272 nos. CWRs/ LWRs (more than 5 years) need to be tackled on a war footing.

(iii) Elimination of buffer rails.

(iv) Conversion of long CWR into shorter lengths of 3 km.

9.3.5 CASE STUDY NO. 5

Title

Derailment of Engine and 11 coaches of 8029 Dn Kurla-Howrah express between Mhasavad and Shirsoli stations of Manmad-Bhusaval section on 25Kv AC Electrified BG double line of Bhusaval Division on Central Railway at about 15.41 Hrs. on 03.12.1998.

Description of accident

8029 Dn Kurla-Howrah Express of 03.12.98 with 18 coaches hauled by electric loco WAM-4 left Pachora station at 15.18 hrs and derailed at 15.41 hrs. between Mhasavad and Shirsoli at Km. 405/15-23.

The accident had resulted in death of 9 persons, Grievous injuries to 13 and simple injuries to 30. When Driver checked the train formation in rear, he observed that train engine had derailed by all wheels of rear trolley and following eleven coaches had completely derailed while remaining 7 coaches, namely, 12th to 18th from engine were on track. Ist coach had derailed by all wheels but coupling with the loco was intact. The coaches had drifted towards left side by varying degrees and the coach no.5th & 6th were thrown by about 20m from center of track. Though the train was running at speed of 97 Kmph, because of large number of coaches derailed, the drag of engine was limited to 230 m only.

The track had 52kg 72 UTS rail and was laid in 1988. It had steel trough sleepers to 1540 density with 300mm machine crushed ballast of which 125 mm was, reportedly, clear and rest was caked. Distressing of this CWR was last done in bits and pieces in March, 94 to September, 97. The last USFD was done in September, 98 in the vicinity of the accident site.

Cause of the accident

Sudden failure of AT weld on continuous welded rails.

Reasons for arriving at the cause of the accident

In the locomotive a few defects of substantial nature in its under gear were noticed. Similarly, first coach also had large number of parameters beyond permissible limit. Though some of these parameters might be beyond permissible limit because of effect of derailment but definitely not all of them. The train was traveling on down grade and, therefore, all the buffers must have been in buffing mode. It was observed that plunger plate of left rear buffer of the locomotive was missing.

In case left rear buffer plunger plate of loco had given way prior to failed weld location in km. 405/15-17, then buffing action of right buffer would induce anticlockwise rotation to the loco which would result in higher lateral force on left rail by wheels of leading trolley. The left rail joint at 405/5-1 was already fatigued with rail in substantial tension and passage of leading trolley resulted in complete fracture of joint and consequent multiple fracture of rail, discontinuity and misalignment of left rail causing derailment of rear trolley of loco by all wheels. The lateral shift of rear trolley of loco towards left further resulted in locking of right side rear buffer of loco and right leading buffer of first coach due to slowing of loco and momentum of the following coach.

In case, it is considered that the first coach derailed first and shifted laterally, then locking of buffer could have also taken place with braking of loco but in this case the coach buffer would be left of loco buffer. This was not so at site. Moreover, such situation is further ruled out as lateral shift of derailed rear trolley of locomotive was more than similar shift of derailed leading trolley of first coach.

Thus, process of derailment in the extant accident was initiated most likely on account of complete snapping of fatigued AT weld in tension at Km.405/15-17 due to combination of increased vertical load and lateral force by wheels of leading trolley of locomotive of train, coupled with multiple rail fractures of landing rail at AT joint, misalignment caused to landing rail and its ultimate overturning with head towards cess of Dn line. This resulted in derailment of trailing trolley of loco by all wheels, further leading to derailment of 11 consecutive coaches behind the loco. The first coach NE 4836 had derailed by its all wheels and had also knocked off three OHE masts at 405/17, 405/19 & 405/21. The derailed coaches had drifted upto 20 m towards lefts of Dn line and there was snapping of coupling between the first and second coaches from the engine.

Important recommendations

(i) Distressing of LWR/CWR should be carried out at one temperature irrespective of length and in the shortest possible time not exceeding a week in any case.

(ii) The present prescription provided that AT Welds made by the conventional process should be first tested with 80^0

hand probing flange after passage of 80 GMT and, thereafter, passage of 40 GMT each in accordance to para 7.8.2.1 of the manual of USFD, 1998. The para 7.8.2.2 specifies that such 80^0 hand probing for flange in case of SKV welds is not needed after initial testing. It is suggested that 80^0 hand probing for the flanges of all AT welds should be made mandatory after passage of 2/3rd of designated life of the parent rail or after 10 years of life and, thereafter, on same scale as for the parent rail.

(iii) 8 door ICF coaches should be withdrawn immediately from the passengers services in regular trains.

(iv) For expeditious rescue and relief operations two sets each of cold cutting equipments and compressors with adequate length of lead should be provided in the Tool Van of ARMEs.

9.3.6 CASE STUDY NO.6

Title

Derailment of 59 Up Howrah-Gauhati Kamrup Express between Jamirghata and Gourmalda stations of Malda division of Eastern Railway at 03/15 hrs on 19.07.85.

Description of Accident

On 19. 07. 85, 59 Up Howrah-Gauhati Kamrup Express, while on run between Jamirghata and Gourmalda stations on New Farakka-Malda Town single line Non-electrified Broad Gauge section of Malda Division of EasternRailway, derailed at km. 281/11-13 at about03.15. The Express hauled by a WDM-2 diesel electric locomotive No. 17900 consisted of 16 bogie coaches of which the front 8 coaches from the train engine went off the rails. The rear truck of the locomotive derailed of all the 6 wheels. The first coach behind the locomotive derailed of all wheels. The next 2 coaches capsized on the left and came to rest down below the embankment at a distance of upto 20 m from the track. The 4th coach got derailed of all wheels and thrown out of the track to come to rest across the slope at a steep angle with the initial alignment. The 5th and 6th coaches also derailed of all wheels and rested across the slope in a tilted condition. The 7th coach, which was the dining car, also derailed of all wheels and rammed against the rear of the 6th coach, as a result of which the front portion of the dining car telescoped and got lifted up from the front trolley. In the 8th coach, all the wheels of front trolley only derailed. The 9th to 16th coaches remained on rail. There was a parting of the train between the 1st and 2nd coaches behind the engine and the two coaches lay apart by about 52.73 m. The total length of dragging of the first coach from the point of drop was about 275 m. The coaches that derailed or capsized, did so wholly on the left. However, the left wheels of the rear truck of the locomotive derailed on the right of the left rail which stood upright while the right side wheels of this truck rested on the web of the right rail which had tilted heavily with the rail head turning to the right. The front truck wheels of the locomotive were on rail.

Cause of the accident

The accident occurred on account of deficiency of the leading trolley right trailing outer axle box spring in coach No. NR VPU 7218 (1st coach after train engine).

Reasons for arriving at the cause of the accident

a) The speed of train at the time of accident was 70 kmph, as ascertained from the VDO speed recorder chart of the loco, which was below the sectional speed of 100 kmph. Hence there was no element of over speeding.

b) There was no obstruction at site, as per evidences as well as statements of the crew concerned.

c) Condition of track geometry, as revealed by Joint track measurement, did not indicate very high standard of maintenance. However, they were not serious enough to warrant any consideration either. The results of TRC charts of April 85 (3 months prior to accident) indicated unevenness & twist to be in C category. Portable accelerometer recording done (20 days prior to accident) indicated there was no peak exceeding 0.15 g (either in vertical or lateral mode) on km. 281/282. Drivers of the trains prior to accident indicated that there was no lurch at this location, rather they stated that riding was good. On the whole, the track geometry parameters were such that at 70 kmph, they were considered adequate.

d) There was no rail fracture on either rail after the accident.

e) Even on the distorted track, there was no occurrence of dropping of wheel inside the track as a result of spread of gauge. Lateral displacement of spikes by cutting into the sleepers also was not significant. Hence possibility of spread gauge was ruled out.

f) There was no evidence of missing fastenings on wooden sleepers even though on the outside of the curve, only one spike had been used in most cases. By and large, the fastenings available were considered adequate.

g) From various evidences made available, it was clear that there could be no possibility of sabotage.

h) Rolling stock

(i) Loco was not overdue any schedule and the measurements in loco under gear did not reveal any serious shortcoming.

(ii) Eight coaches, involved in the accident, were not overdue any schedule. Since they had suffered severe damages, check of their axle boxes carried out revealed that they were in good condition and none showed signs of seizure. Many of the under gear fittings as well as trolleys got separated or wrenched out as a result of accident. All the various components could, however, be accounted for and no serious discrepancy could be located except in case of NR VPU 7218 which was the first coach (BEML) behind the train engine. In this coach the leading bogie (right trailing) outer axle box spring, a primary spring, was found deficient. In the same trolley, the right side vertical shock absorber bottom plate and the right side trailing roller were found to be deficient. Close examination of these locations indicated that these deficiencies had been existent for quite some time. The leading axle of the trailing bogie had got bent apparently as a result of the accident. Right leading wheel of the rear bogie of this coach had distinct dent marks on the wheel flange. There were 3 such dents, thesize of the dents being 62mm/18mm deep, 25mm/10mm deep. These dent marks, particularly the deepest one, were clearly indicative of the wheel flange having hit against the top of rail end. The corresponding dent marks at the rear end of rail were also noticed. It was jointly observed by DEN & DME that the bottom base of axle boxes spring seat (primary) in VPU 7218 was a plain base without any lug as required in accordance with the modified arrangement shown of Maintenance Manual for BEML coaches issued by RDSO/LKO in April, 71. It was also admitted that the above deficiency of axle box spring had caused the derailment of the VPU, making it the first vehicle to derail. Loading of VPU was checked. It had a total number of 324 packages with a gross weight of 10390 kg. The front half of VPU had 5110 kg, while the rear half had 5280 kg. Hence there was, no uneven loading of the VPU.

i) There was ample circumstantial evidence to indicate that the first coach to derail was the VPU next to engine, as seen by deep and wide furrows made on the left hand side of the track by passage of derailing vehicles.

j) The main question arises How could the axle box spring be deficient? The missing spring could not be traced within a distance of 3 to 4 kms behind the site of accident. It was rather baffling as to whether the coach had started from Howrah without the spring or had it fallen en route?

Examination of TXR records at C&W depot HWH revealed that the above VPU was examined not with the rake of 59 UP in the pit line on 18/7/85, but on the old parcel siding, where facility for detailed examination was inadequate. It was not known whether the spring was in broken condition at HWH itself , causing itself to work out on route. From site observation, the spring seat assembly indicated that spring was missing for quite some time. The condition of vertical shock absorber on the same right trolley indicated that it was not attended for a long time. This indicated poor standard of C&W maintenance though it could not be conclusively proved that the VPU coach left Howrah with the deficient spring. But it was definitely not there for 3 kms in rear of derailment as per evidence at site. The reason as to why it worked out during run was mainly due to deficiency of lug needed to properly seat the spring. This provision of lug had not been carried out in the POH last done at N.Rly. Alambagh shops on 12/4/85, who were held responsible, though in a secondary capacity.

Important Recommendation

Immediate steps should be taken by all zonal Railways to check up BEML coaches in service as regards provision of this lug in spring seat assembly. This had been accepted by Railway Board and instructions issued accordingly.

9.3.7 CASE STUDY NO. 7

Title

Derailment of 301 Up Asansol Varanasi Passenger at Km.352/29 - 353/3 between Keshwari block Hut and Chaube stations on Gomoh-Gaya Grand chord double line broad guage electrified section of Dhanbad Division of Eastern Railway on 17-04-2002.

Description of the accident

On 17.04.2002, the train no. 301 Up Asansol Varanasi Passenger was being hauled by WAM-4 loco No. 21286 up to Km 264/19 between Pradhan khunta and Dokra Block Hut where the Brake Pressure pipe of loco broke due to cow run over. The train was brought to Dhanbad (km. 270.88) on train brakes and loco no. WAG- 5H 23350 was attached in front of the earlier loco. The train consisted of Loco no. WAG-5H 23350 and dead loco No. WAM-4 21286 + 12 coaches. The train was running 139 minutes behind schedule. At 13.05 hrs, the train derailed between Keshwari block Hut and Chaube station at Km. 352/29-353/3 of Gomoh-Gaya grand chord electrified section. Before derailment, the train had negotiated a 1.55^0 left hand curve from Km. 352/11 to 352/27. Thereafter, it was moving on a straight track on a rising gradient of 1 in 200. The height of the bank is 1 to 1.5m. It is a plain country. The formation has moorum soil. The track structure consists of 60 kg MM rails of 1985 laid on PSC-14 sleepers (M+7) of 1985 and wear is 6 to 7%.

The train was running at a speed of 65 kmph at the time of derailment and there was no speed restriction at that site and the speed of the train was within the permissible limit.

The accident resulted in two casualties : one grievous injury and two simple injuries.

Cause of the accident

The cause of the accident was established as buckling of track.

Reasons for arriving at the cause of the accident

The speed of the train was within permissible limit. There was no report of any natural calamity in the area and also no obstruction was reported on the track. None of the witnesses suspected any sabotage. The driver who was running the train has a clean record and during cross examination also it was found that his reflexes and knowledge were sufficient for a good loco driver and, so, bad engine-manship was not expected.

Pantograph of the train engine no. 23350 was in lowered condition. Middle wheel of Kalka end trolley was out of trolley and the equalizer was worked out. Horn was missing and only one horn bolt was available in thread. CBC and buffer interlocked with next engine. Engine was with all wheels derailed and tilted.

This can lead to following possibilities :

1. The equalizer could have worked out and caused derailment.

2. The buffer were interlocked and could have caused the derailment.

In this case, the loco had traveled more than 100m in the derailed condition. Therefore, in all likelihood, the equalizer must have worked out after the derailment. As regards interlocking of buffers, the buffer heights recorded are within permissible limits and none of the witnesses pointed out towards a drooping buffer. Therefore, possibility of buffers getting interlocked and causing the derailment is ruled out. The track can only contribute to derailment due to following reasons:

1. Bad Track Geometry
2. Rail fracture
3. Buckling

Tamping of the track was done just 5 days before the accident. The possibility of buckling was analysed and the circumstances which point towards buckling are :

i. Bad fittings Crushed grooved rubber pads and corroded liners which had reduced toe load on rails and, thus, allowed the long welded rail to creep causing locked up stresses.

i . Scanty ballast - Ballast available was less than requirement.

iii. Distressing of continuous welded rail was overdue.

iv. The train Driver has also stated that he saw that the track had buckled.

v. The creep in Long Welded Rails was abnormal due to bad fittings

The factors which negate buckling are :

i. It was a concrete sleeper track where chances of buckling are rare.

ii . Rail temperature was around 54-55^{0}C at the time of accident, whereas distressing temperature was 44^{0}C in 1999.

iii. Missing fittings were hardly 1% ,which indicates that there was no laxity in maintenance of track by the Gang.

iv. No work was in progress to disturb the track.

v. The formation is good.

vi. The gap in SEJs was within the permissible limits.

However, abnormal creep in Long Welded Rails and scanty ballast lead to conclude that the derailment could have occurred due to buckling of track.

Important Recommendations

1. Engineering staff/field officers should be trained in maintenance of Long Welded Rails and thermit welds through short courses in Divisions and Headquarter.
2. Immediate replacement of fittings like crushed grooved rubber pads and corroded liners should be undertaken. Ballasting should be undertaken at locations wherever it is scanty. Till these are complied with, suitable speed restrictions should be imposed wherever condition warrants.
3. Foot-plate books or register should be got printed by the Railway and issued to all Assistant Engineers and Permanent Way Inspector.
4. Motor Trolley should be restored to Assistant Engineer/ Gujhandi immediately. Without motor trolley he cannot cover his section as per schedules laid down.

9.3.8 CASE STUDY NO. 8

Title

Derailment of 3007 Dn Howrah-Sri Ganganagar Toofan Express (Electric Engine Hauled) at Km. 1344.00- 1344.133 between Idgah and Agra cantt stations (single line electrified section) of Jhansi Division of Central Railway at 19.10 hours on 11.08.1996.

Description of the accident

3007 Dn Howrah-Sri Ganganagar Toofan Express left Jamuna Bridge at 15.20 hrs. and reached Agra Fort at 15.35 hrs. The train left Agra Fort, after long detention due to OHE disturbance, at 18.05 hrs. and reached Idgah at 18.15 hrs. Idgah-Agra Cantt is a small single line (Electrified) section, 2.10 km long with one level crossing No.496 (km.1344.402) in between and 600m long 5^0 curve.

The train had left Idagh at 18.50 hrs. for Agra Cantt after a halt of 35'. It was running late by 4.00 Hr. and 8 minutes. As stated by the driver, the train could not pick up speed due to slipping of engine wheels, on wet rails as it was drizzling, and the train was moving on a rising gradient. He looked back from the engine on the inside of curve when he saw some passengers jumping out of the coach. Seeing this, he stopped the train and got down from the engine and went behind. He found that front SLR No.7952 ER had derailed.

The train had stopped after a drag of 38.2 m. In the process of derailment, the engine remained on track, leading axle of trailing trolley of the first coach derailed on straight portion of track after passing the curve with left side wheel inside and the right side wheel outside the track. Front trolley and trailing axle of trailing trolley remained on rail. Second to fifth coaches did not derail. Sixth coach derailed by only leading left wheel of trailing trolley inside the track, other wheels remaining on rail. Seventh coach derailed by two left side wheels of leading trolley, inside the track, other wheels remaining on the rail. These two coaches derailed in the transition portion of curve and in the process, two left side rails got tilted. Rest of the Five rear coaches remained on track (circular portion of curve). In all out of 12 coaches, 3 coaches viz 1st, 6th and 7th from engine derailed. There was no parting between any coach and all the coupling were intact.

There was no incident of death. One passenger received grievous injuries.

Cause of the accident

A combination of following factors:

i) Track in bad state of maintenance with large no. of missing fittings and unsatisfactory track geometry.

ii) Non functioning sanders of the engine and an unqualified goods drivers not able to manoeuvre the engine properly.

Reasons for arriving at the cause of the accident

There is a permanent speed restriction of 30 kmph on this line. The estimated speed of the train at the time of derailment was 5 kmph. At the time of derailment, there was no indication of any convulsion of nature. Based on the evidences, it was found that there was no possibility of sabotage and no obstruction on the track was there.

It is an old track laid more than 25 year back with wooden sleepers. Whatever ballast is available, has all caked up and no clean ballast is available, thus the drain- age was very poor. The accumulation of garbage heap on the East side along the track has further blocked the drain-age. This area is theft prone area. Thus, the theft of P.Way fittings is a common phenomenon in this area. In the section, the deficiency of keys and other fittings has built up over a period of time. As only two pairs of trains run on this lines, it has been considered an unimportant line in comparison to the main line and very less attention has been paid to its maintenance. The joint observations of track taken after the derailment show huge deficiency of track fittings. In fact, according to these observations, keys of both the rails were found missing right from 1st coach to 8th coach length.

Thus, poor maintenance condition of track and poor drainage, mainly due to misuse of track by outsiders as a dumping and shit ground, has been a contributory factor in causing the derailment of the train.

The detailed examination of coaches suggested that none of the coach have contributed in the derailment. It was found that the Driver was not qualified to work an Express Train and the sanders of the engine, which could control the wheel slips, were not functioning. The driver could not manouevre the engine properly in the difficult situation that he faced due to slippage of engine wheels. In his anxiety to raise the speed, he repeatedly tried to increase the notches causing shuttling motion in the train. During one such motion, the leading pair of wheels of trailing trolley of 1st coach jumped off the rails when the train was in buffing mode. The derailment of one wheel of 6th coach and two wheels of 7th coach followed suit. No mounting marks on the right rail were found. Unsatisfactory condition of track contributed in this process. Absence of any mounting mark on the rail table and derailment of only few wheels that too of 1st, 6th and 7th coaches, other coaches remaining intact on rails, is a sure indication of the bad engineman-ship on the part of the driver. However, driver was not held responsible, firstly because he was not qualified to work an express train, still he was booked on this train secondly he could not control the wheel slippage due to non functioning of the sanders.

Important Recommendations

(i) Swift and deterrent action should be taken against the habitual absentee gangmen to check large scale absenteeism in P.Way gangs.

(ii) It may be desirable to appoint some safaiwalas in the P.Way gangs of this and similar other sections passing through densely populated areas for re-moving night soil from the track in order to facilitate track maintenance.

(iii) Concept of Zero missing fittings gangs should be revived with all seriousness to ensure complete fittings in the track.

(iv) Suitable instructions should be issued to all concerned that sanders of the engines must remain in working order, particularly in rainy season, with adequate sand. This may be listed as a safety equipment.

(v) A driver not qualified to drive a particular train should not be booked on such a train.

9.3.9 CASE STUDY NO. 9

Title

Derailment of Tr. No. MSP-17 Down EMU suburban local ex. Chennai Central Sullurupeeteta at Km. 2/44 between Basin Bridge Junction and Korukkupet stations on Madras-Gudur Broad Gauge Double line electrified section of Chennai Division of Southern Railway at 19.49 hrs on 09.02.04.

Description of accident

MSP 17 down suburban local from Chennai Central to Sullurupetta started BBQ at 19.48 hours with starter signal No.37 showing green aspect and the guard giving starting bell. Immediately after passing Km.2/44, within a minute, the motorman experienced sudden jerk (slight pulling affect). He immediately opened DMH, applied emergency brakes and the train stopped. When he looked backward he found that derailment had taken place and the formation is infringing the up main line (KOK-BBQ). He switched on the flasher light and informed the matter to SM/BQ.

Cause of accident

Due to freak unintended operation of point contactor unit of point No. 42 A resulting in operation of above point from normal to reverse while the EMU train was in motion.

Reasons for arriving at the cause of the accident

An accident of this nature could occur due to singular or combination of factors important of them being

- Defects in EMU stock
- Defect/deficiencies in Permanent Way
- Sabotage/train wrecking
- Over speeding/sudden braking/bad Engineman-ship
- Defect/deficiencies in signaling system

1. Examination of entire rake after accident did not reveal

any defect/ deficiency so as to cause the above derailment. In view of the above, any defects/deficiency in the EMU stock as a cause of derailment is ruled out.

2. Though the condition of scissors cross-over is not satisfactory, the poor track condition as a cause of derailment is ruled out due to absence of wheel-mounting marks on rails switches and wheels of all the coaches other than coach at 4th position on rails with first three coaches in right direction and rear four coaches on turn out side.

3. The driver of the EMU train has stated that he did not find any outsider near scissor cross-over tampering with the track. During inquiry also, no one pointed out towards sabotage as the cause of the derailment. In view of the above, sabotage as the cause of the derailment is ruled out.

4. The motor coaches are not fitted with speed recording device and, therefore, the speed of the train at the time of derailment cannot be precisely ascertained. The trials jointly carried out by the officers of various departments have revealed that the running time from the time the train occupies the first circuit (37T) immediately passing the starter signal no. 37 upto the point location as 21.562 seconds. Therefore, it could be interpreted from the above that the speed of above train would not have exceeded 28 Kmph. In view of the above, over-speeding/sudden braking/ bad engineman-ship as a cause of accident is ruled out.

5. Scissors cross over No. 41 & 42 connect up and down main line. In this accident, the first 3 coaches travelled on the intended route with point no. 42A in Normal position, the 4th coach derailed by taking two routes and all rear four coaches neatly traveled on the un-intended route with Point No.42A in reverse position. This establishes the fact that Point No. 42A have moved to reverse position while train is in motion. Further, the fact that Point No. 42A is unlocked, set to reverse and locked in reverse under the wheels indicates that the point No. 42A have been commanded electrically to go to reverse position.

Further, the Points No. 42B (the other end of the cross over) being in fully damaged condition with its position to normal also establishes that only one near end of the cross over (point no. 42A) alone received the command to move to reverse position when the train was in motion. Further investigations have led to the following conclusion:

(i) Point operating command did not originate from RRI.

(ii) Point machine did not operate on its own.

(iii) Point machine operated in between the wheels with the spurious command of the point contactor unit.

(iv) The spurious command is confined only to 42A end of the cross over

(v) The point machine was not fully guarded against its response to spurious command due to signaling circuitry deficiencies.

(vi) The spurious operation of the point contactor unit is due to external interference/freak operation.

In nutshell, the point No. 42A has been operated while the train is under motion due to the accidental switch over of the point contactor unit due to external interference/ freak operation. The following are the various possibilities for such undesirable operation.

1. Defects in interlocking
2. External interference
3. Effects of electro-magnetic induction
4. Deficiencies in point operating methodology.

On perusal of the dump of the data logger, it is seen that no attempt has been made by the operator to command point No. 42 A to reverse. As per the data logger the point control relay was not activated at all, indicating that the interlocking did not permit initiation of point operation. Therefore, any possibility of defect in interlocking is ruled out as the cause of the accident.

Further going by the situation, there was no signal failure at that point of time. The said MSP-17 was not detained. No physical work was in progress in BB yard, the ESM did not requisition for the relay room key. There was also no failure/ provocation to open the relay room. By circumstantial evidence the ESM was available in ESMs room. Therefore, there is no reason to suspect human intervention. Also, based on the evidences, malfunctioning of the point contactor unit due to vibration is ruled out.

Electromagnetic induction caused by the H.T. lines as a cause of the accident is ruled out.

The detailed examination revealed that there were deficiencies in signaling circuits.

On most of the Railways for the last 15 years, 110 Volt D.C. Power supply is made available only for few seconds along with the command for point operation, so as to minimize the probabilities of spurious operation of point machine. In this case, 110 Volt D.C. Power supply is omnipresent up to the terminals of point contactor unit.

Therefore, the continuous presence of 110V in the entire cabling system renders the point operation vulnerable to spurious operation during the occurrence of earth faults of the cable, mal-operation of relays etc. Thus, the safety level in signaling goes down substantially.

Also, it was found that the incoming power supply of 110 Volt DC and the outgoing power supply to the Point Machine are terminated in the same box in close proximity of few centimeters.

Any accidental connection between the incoming and outgoing power supply for reasons like lizards, ingress of rain water etc., have the potentiality to operate the machines even without the change-over of the Point Contactor relay as 110 V DC power supply is continuously available.

Going by the probability, the unsatisfactory feature is the presence of 110 V D.C. power supply to prompt spurious operation of the point contactor unit to reverse appears to be freak incident. This might have occurred either due to mechanical or electrical disturbance. Because of the freak operation of the point contractor unit, the readily available 110 Volt D.C. Power supply was extended to the point machine of points No. 42A. Hence only A end of the cross over alone got operated.

Important Recommendations

1. The definition of track locking features including the methodology for its implementation should be incorporated in Signal Engineering Manual in the right spirit of SEM para 21.8.2. and 21.12.3 to prevent operation of points even with accidental operation of points contactor unit.
2. Continuous presence of 110 V power supply in the cabling system being hazardous, the point control circuits should be standardized for ensuring availability of power supply for point operation only for a very short duration in conjunction with point operating command.
3. In terms of para 22.5.7 of Signaling Manual, Railway Board should review deployment of magnetic stick relays and issue necessary instructions for their elimination in external circuits viz., outdoor installations in 25 KV AC electrified areas.
4. Point control circuitry shall have over protection in terms of SEM 21.12.2 & 21.12.3.

9.3.10 Case Study No. 10

Title

Derailment of K – 27 DN CSTM – KYN EMU Local at Km. 0/ 0-1 in CSTM Yard in Mumbai CST – Kalyan Electrified BG Multiple Lines Secion of Mumbai Division of Central Railway at 08.39 Hrs on 21.12.2011.

Description of accident

The EMU Local train, while coming out from platform, derailed.

Both the bogies of the coach 7th from driving cab derailed of all wheels and the front bogie of the coach 8th from driving cab derailed of all wheels.

Site of accident – Point of Mount of derailment was located on a right hand curve of 5.4 degree. Track for 165 m in rear of POM the on a right hand curve of 6.35 degree and 95 m ahead another right hand curve of 3.29 degree is located. Point of Mount was located 1.24 m ahead of the end point of check rail provided on the curve. Point of Drop was located 5.27 m ahead of Point of Mount. Plain track was having 52 kg rail on 1540 per Km density PSC sleepers.

Cause of the Accident

In the Enquiry, cause of the accident was ascertained as defect in permanent way and rolling stock.

Reasons for arriving at the cause of the accident

1. Speed of train and bad driving

Speed recording facility was not available in the accident involved EMU rake. As such, speed could not be calculated using speedometer chart.

As per the statement of the motorman of the train, speed at the time of accident was 14/15kmph. The Point of Mount was on track circuit No.317. Speed at the time of accident, calculated based on the data logger track down condition report, was 23 kmph.

In view of these evidences, speed of the train at the time of accident was 23 kmph.

From the above, speed of the train was within the permissible speed of 25 kmph at the location. Further, the derailment had taken place only after 1 minute of the start of the train from CSTM station. Therefore, there was no reason for sudden increase/ decrease of speed.

2. First vehicle to derail

Sequence of wheel drop marks and damages to track were as follows:-

One wheel mounted on outer rail (left rail) on 5.4 degree right hand curve and wheel dropped outside on sleeper. Continuous wheel traveling marks were observed on sleepers on LH outer side and RH inner side for a distance of 115.20m from Point of Drop. Train stopped after traveling a distance of 115.20 m front point of drop. RH wheels, which dropped inside, damaged and dragged the tongue rail and lead rail of points and crossing (2 no.) encountered in the process.

Both the trolleys of non driving trailer coach no. 310C, 7th from front driving cab, had clerailed on the curve and all RH wheels were inside of inner rail. All LH wheels were in lifted condition. Complete coach was tilted towards right. Trailing trolley of motor coach No.372B, 8th from front driving cab was on track and leading trolley was completely derailed, leading left wheel was inside LH rail and RH wheel was outside the RH rail and trailing left wheel was outside LH rail and RH wheel was inside RH rail.

Analysing this derailed position of coach no. 372B, as the vehicle was traveling on right hand curve and left wheels were traveling on outer rail of curve, such type of derailment cannot occur due to reasons of off loading and mounting of wheels on outer rail. Such a derailed position could be due to forces generated from derailment of the adjoining vehicle and its dragging after derailment. In view of above facts, it is clear that motor coach no. 372B, 8th from front driving cab was not the first vehicle to derail. As only two coaches had derailed, it was concluded that non driving trailer coach no. 310C, 7th from front driving cab, was the first to derail.

3. Defects in Rolling Stock

The following defects were found in the non driving trailer coach No.310C

(i) No oil was found in leading trolley pivot housing. On periphery of top brass of centre pivot, dent marks of size

5x10mm, 7x20mm, 4x20mm, 5x5 mm were found at 4 places. (1 was at BB end, 2 at RH side and1 was at KYN end).

(ii) Centre pivot pin of trailing trolley was found 20mm inclined towards RH side from vertical. On top brass of centre pivot on RH side scratch mark of 40mm length was found. On upper collar of centre pivot on RH side more rubbing marks than on LH side were found.

(iii) On the trailing trolley, both lateral dampers which connect bolster with bogie were found in disconnected position.

It was concluded in the Inquiry Report that trailing bogie of coach No.310C was the first to derail. It was reasoned that as centre pivot of trailing trolley was 20mm inclined toward RH side, it restricted the free rotational movement of trolley. Further, as both lateral dampers were disconnected, lateral forces generated due to motion of vehicle on curve were not damped. As a result, when both axles of trailing trolley came out of check rail, wheel climbed on the outer rail and derailed.

4. Defects in track

Cross level variation between station no.'0' and '1' was 27mm i.e. twist was 9mm/meter. Versine on 6m chord on station No. '0' and station No.'1' was 14 mm and 9 mm respectively. Effect of these track parameters on derailment was analysed in the Inquiry Report, as below.

RDSO has laid down limit of twist in track for CRT wagons, as 6.94 mm/m and 8.33 mm/m for a speed of 30 kmph and 15 kmph, respectively. In the present case, speed was 23 kmph and twist in track was 9mm/m. Though above limits of twist are prescribed for a four wheeler CRT wagon, existence of excessive twist of 9mm/m on curve would cause off loading and mounting of wheel on outer rail of curve, which will lead to derailment.

As per para 421 of IRPWM, for speed below 80 kmph and upto 50 kmph, limit of station to station versine variation is 40mm or 25% of average versine on circular curve, whichever is more.

In the present case, versine variation between station No.'0' and '1' is 55.4mm (converting measured versine on 6 m chord of 14 mm and 9 mm to an equivalent versine on 20 m chord of 155.4 mm and 100mm), which is beyond permissible limit mentioned on Para 421 of IRPWM. Though the above limit mentioned in IRPWM is for a speed of 50 kmph and above and it is for riding comfort, abrupt change in versine would create high lateral flange forces and wheel would have a tendency to mount on outer rail on curve and derail.

In view of above facts, above mentioned defects in track were considered to have contributed towards derailment.

Further discussion

Axle box spring heights measurement for the trailing trolley of derailed trailer coach No. 310C were:

TTLL (L) = 235 mm	TTLR (L) = 238 mm
TTLL (T) = 238 mm	TTLR (T) = 245 mm
TTTL (L) = 237 mm	TTTR (L) = 246 mm
TTTL (T) = 236 mm	TTTR (T) = 245 mm

Axle box crown clearance for the same trolley were:

TTLL = 27 mm	TTLR = 39 mm
TTTL = 32 mm	TTTR = 39 mm

Maximum variation among the diagonally opposite springs (TTLL and TTTR) was 11 mm. Variation in axle box crown clearances is also large. As per Maintenance Instructions, axle box springs on the same bogie should be from the same group, such that variation in spring height and, hence, wheel off-loading is limited. In the present case, a large variation in spring height would have caused large off-loading and contributed to wheel mounting the rail.

9.3.11 Case Study No. 11

Title

Derailment of 6 wagons of CONRAJ/DDL Goods train at

Godhra in Vadodara Division of Western Railway on 08th Nov, 2007

Description of the accident

On 8th Nov 2007, CONRAJ/DDL (BLCA/ BLCB wagons) derailed in Godhra yard (Rajdhani Route of WR) at Point No. 107/ 108, while passing over cross-over between up and down main lines in a sharp curves (3.5^0). The derailment occurred at about 00.55 hrs. 5th wagon (from engine) derailed by rear bogie and 6th,7th,8th,9th and 10th wagon derailed by all wheels. Train parting took place between the 5th and 6th wagon. Total length of drag subsequent to derailment was 157 m.

The Point of Mount (POM) was observed on the left hand tongue rail of point no 107 (a right hand turnout) at a distance of 1.24 m from the Actual Toe of Switch (ATS) and point of Drop (POD) was observed at 4.76 m from POM.

Cause of the accident

In the Joint Enquiry conducted at JA Grade level, cause of the accident was found as ' The overall length of the cross-over 107/ 108, laid between two curves, was less than the standard requirement by 3.708 m. Slack available in slack-less draw bar of BLC wagons is 1.5" as against 7.5" in center buffer coupler. The derailment took place on a sharp curve due to limited slack available.'

Reasons for arriving at the cause of the accident

P. Way : Gauge was within the maintenance limit of -6 mm and +15 mm (upto 5 degree curve) and upto +20 mm (above 5 degree curve), as per IRPWM Para No. 224 (V). Maximum track twist was 8 mm over 3 m base. As per 'Slow Down Tolerances' for a four wheeler, laid down by RDSO, Lr. No. CRA 501, dt. 29.04.1983, permissible track twist for 15kmph is 30mm on a chord of 3.6m. Hence, twist was considered to be very much within permissible limit.

Wheel diameter difference : Difference in wheel diameters on the same axle of 2 mm on leading axle and 3 mm on trailing axle of trailing trolley of wagon No CN 100402 (5th from engine) was observed. The difference was right wheel diameter larger than left wheel diameter by 5 mm for both leading as well as trailing axles of the leading trolley of wagon 6th from engine. For the same wagon, left wheel diameter was larger than right wheel diameter by 7mm for both leading as well as trailing axles of its trailing trolley.

The Enquiry Committee observed that the difference in wheel diameters on the same axle is stipulated only for newly profiled wheels, as 0.5 mm (IRCA Pt. III, Rule No. 2.8.14.2). These limits do not form a part of train examination. CME-CCG letter No. T5/42/1117 dt. 20.08.85 stipulates a limit of 10 mm after service for condemnation.

Clearance : Longitudinal and lateral clearances between side frame and adaptor and lateral clearance between side frame and bolster for the leading trolley of wagon 6th from engine were scrutinized and found to be within permissible limits of wear.

Sharp flange : Sharp flange was observed in Joint Observation on the leading left and trailing left wheels of the leading trolley of the 6th wagon from engine. The Enquiry Committee ruled out derailment on account of sharp flange in view of the pattern of wear, being not a case of worn root and thin flange, and derailment not being on account of splitting of point. They observed that the dragging of 157 m might have resulted in damage at periphery of the wheels.

Speed of train : It was found from analysis of speedometer record that the speed of train just before derailment was 11- 12 kmph and, suddenly, it became 15 kmph. The speed dropped to 10 kmph in about 6 seconds and, thereafter, remained almost constant for the period in which a dragging of 158 m took place. The Committee reasoned that the sudden increase in speed at the time derailment might have occurred due to slip of engine wheel, resulting from jerk caused by derailment.

Other items : Total load and pattern of loading was found to be normal. Movement of train was studied and it was found that it was made to make a cup shaped movement, by negotiating 2 additional cross-overs, in the process of receipt of the train in the yard. Nothing was mentioned in the Station Working Rule (SWR) about the permissibility of this movement.

History of derailment of CONRAJ : History of derailment in Godhra yard for the last four years was looked into. It was found that a total of four derailments had taken place, three being of BLC rakes and the other being of a diesel locomotive. All the derailments of BLC rakes had occurred during negotiation of points and crossing on curves. One of the derailments of BLC wagons had occurred at the same cross-over (point no. 107/ 108) about 2 months back.

Curve negotiability of BLC wagons : The committee concluded that curve negotiability of BLC wagons was poorer than other stocks on account of its special feature of Slack-less Draw Bar arrangement as the total slack available in this arrangement was only 1 ½" as compared to 7 ½" in the Center Buffer Coupler. Further, the cross-over of point no. 107/ 108 was found to be of smaller length than the theoretical requirement (by about 3 m), which would have caused increased sharpness of the curve of point no. 107.

Recommendations –

The committee made the following recommendations

* Godhra yard, which was having a large number of turnouts taking off from sharp curves, should be re-modelled as long term solution
* Non standard length of cross-overs should be corrected, wherever possible, as a short term improvement
* Movement of BLC rakes should be restricted over deficient layouts

Further Discussion

As per field data on the Indian Railways, BLC wagons have been found to be very prone to derailment while entering a loop through 1 in 8 ½ turnout. A large buff force is likely to be present while the train enters a loop on account of Slack-less Draw Bar system. Under a large buff force, the Slack-less Draw Bar is likely to have increased lateral rigidity, on account of frictional forces in the coupling system. This increased lateral rigidity, in combination with a large angular run, would increase derailment proneness of the wagon in the sharp curve.

In the present case, the wagon was made to negotiate a very sharp curve. A large longitudinal force would also have been present as the train was entering into a loop and the speed variation as indicated by speedometer records.

5th and 6th wagon from engine were BLCA wagon. They have CBC coupler on one end and slack-less draw bar on the other end having heights 1105 mm and 845 mm, respectively. Under condition of buffing, vertical eccentricity would have caused off-loading of rear trolley of the 5th wagon and front trolly of the 6th wagon.

For the leading trolley of wagon 6th from engine, the wheel diameter difference (right hand wheel larger than left hand wheel) would have increased the tendency of angular run in the sharp right hand turnout. Sharp flange was observed, at the Joint Observation stage, in the left leading as well as trailing wheels of the same trolley. Sharpness of flange generates higher frictional force due to increased biting action of flange on rail gauge face, thereby, increasing the chance of derailment.

9.3.12 Case Study No. 12

Title

Derailment of train No. 17416 Express at NVU at Km.481/400-100 of Hubli Division, SW Railway on 02.08.2012 at 18.49 Hrs.

Description of the accident

On 02.08.12 at 18.49 Hrs., Train No. 17416 (Loco No. WDM3D/ 11153/ GTL) derailed of all three pairs of wheel of the front trolley of the locomotive. The derailment took place at an estimated speed of 90 kmph, on the outside of a curve of 3 degree, with M+7 PSC sleepers and 52 Kg/ 90 UTS rail. The weather condition at the time of the accident was that of a drizzle. The derailed wheels travelled for a distance of 510 m before coming to stop. Ahead of the Point of Drop, 12 joggle fish plated welds got freshly fractured due to impact of derailment.

Cause of the accident

In the Joint Enquiry conducted at JA Grade level, cause of the accident was found as 'due to mounting on outer rail of 3 degree left hand curve by the right hand trailing wheel of front bogie of Locomotive No.11153 WDM3D at KM 481/640, which in turn was caused by difference in the diameter on the same axle resulting in the angular running in combination with defective spring and failure of Side bearers in service'.

Reasons for arriving at the cause of the accident

As per Joint observations, there was no mark of wheel mounting the rail. The Point of Drop was seen on outside insert of PSC sleeper on outer rail at km 481.640. As per hit marks on ERCs and insert of sleepers, the derailed wheel travelled in a floating condition from Km 481.640 to 481.416 in curved portion of track and, there after, several sleepers had the marks of wheel hitting in straight portion of track from Km 481.416 to 481.130. Thus, The derailed wheel travelled a distance of 510 mtrs.before coming to stop. Ahead of point of drop, 12 joggle fishplated welds had got freshly fractured right through at or near the weld due to impact of derailment.

Permanent Way : Track parameters, under loaded condition, in the immediate vicinity of rear of Point of Drop were examined, which were as follows:

Station No.	Sleeper No.	Cross Level	Remarks
0	0	130 SE	Point of Drop
	1	130 SE	
	2	130 SE	
	3	130 SE	
1	4	130 SE	
	5	130 SE	
	6	130 SE	
	7	130 SE	
	8	130 SE	
2	9	130 SE	
	10	130 SE	
	11	128 SE	
	12	128 SE	
3	13	124 SE	

a) Gauge – In the immediate vicinity of rear of Point of Drop, gauge was varying from + Neat to + 10mm. This was within the permissible tolerance of -6mm to + 15mm.

b) Twist – Cross level in the immediate vicinity of rear of Point of Drop was varying from minimum 124mm. The maximum twist in the immediate vicinity of rear of Point of Drop for the wheel base worked out to (130 – 124 / 3.81 =) 2.36 mm. per m. This was well within the limit of 3.5mm. per m prescribed as riding criteria (not safety criteria) for 120 Kmph speed based on chord of 3.6 mtrs.

c) Wear – As per joint readings of worn out rail profile, the vertical and lateral wear observed were 4mm and 7mm, which were within the permitted tolerances of 8mm and 10mm, respectively. Angular wear observed was approximately 25 deg, for which no limits are prescribed. However, as per the Joint

Observation recorded, there was no rubbing mark on guage face of outer rail and greasing mark was noticed on outer rail which indicated regular greasing of gauge face of outer rail.

d) Versine – From the Joint readings recorded, versine variation over 90 m in the rear of Point of Drop was varying from minimum 02mm to 18mm, which was well within the permissible limit of 21.5mm (15mm or 20% of the average versine, i.e 0.25 x 86 = 21.5mm, whichever is more) prescribed for local attention of the curve. As per Correction slip no. 96 dated 28.02.2004 of IRPW Manual, the limits prescribed were for riding comfort only and not safety tolerances.

As per latest OMS recording results on 27.07.12, there was no peak recorded at KM 481.

Therefore, contribution of Gauge, Twist, Versine and Wear as cause of/contribution to derailment was ruled out.

Rolling stock - Locomotive

a) Wheel diameter – As per last check on 28.07.2012 and from the Joint observations at site, the loco wheel diameters were as follows:

Wheelset	As per last check on 28.07.12		As per joint observation after derailment		
	Left wheel dia in mm	Right wheel dia in mm	Left wheel dia in mm	Right wheel dia in mm	Difference between Left and right
1	1078	1078	1070	1069.5	0.5
2	1078	1078	1071.5	1067	4.5
3	1078	1078	1071.5	1068.5	3.0

There was a large variation in the readings of wheel dia between the last schedule on 28.07.2012 and Joint observations after the accident, within a short period of 06 days. This raised a doubt about correct recording of this important parameter during the last schedule. The larger diameter wheel would have tried to travel a longer distance than the smaller diameter wheel in a given number of revolutions, resulting in the wheelset running persistently angular. Persistently angular run of wheel increases the derailment proneness appreciably. From the above readings, it was observed that the inner wheels had larger diameter, resulting in increased angularity in the curve.

b) Spring height - Readings of Side Bearer condition, Centre Pivot condition, and Under Frame, etc. were not recorded on the plea that all. Loco readings were already given/taken at EMD shed as per Accident Manual. However, as springs and side bearer play a major role having bearing on safety, especially while negotiating curve, few readings were taken at Diesel Loco Shed/ Guntakal on 08.08.2012, after bringing the matter to the notice at higher levels.

Joint readings of spring Free height for outer coil spring were taken but other measurements (height under working load, free height of inner spring) were not taken. Free height of outer spring varied between 549 mm and 563 mm.

As per stipulations for Outer spring of Primary stage, Free height is 552 mm (with permissible variation of + 9 mm) and working height is 447 mm (permissible variation + 4 mm, - 6 mm and working load 4288 kg). The springs are to be grouped and colour coded as A (white) and B (red) based on the working height range of 441 - 446.5 mm and 447 – 451 mm, respectively. Spring of same working height (group) are to be used on the same bogie.

Variation in the free height of outer springs indicated a likely variation in the loaded spring height of a larger magnitude than the permitted.

c) Side Bearers – Measured free heights were as below

SL No.	Right (mm)	Left (mm)
01	159, 159.5	158,157
02	161, 159	161,159
03	158, 159	161, 160
04	163, 161	162, 160

As per Maintenance Instructions of the Loco, free height of side bearers is 165 mm. They are to be replaced in case of crack of 50mm length or more on rubber surface, bond failure of 40mm length or more and permanent set of 4mm or more. Thus, the permanent set for several of the side bearers exceeded the limit. It was observed by the Inquiry Committee that the permanent set data indicated that the side bearers had lost their resilience.

Engineman-ship : The locomotive was found to have sufficient brake power. Brake application and releasing was proper. As per the speedometer readings, the maximum speed was 90 kmph at 18.41.13 hours, before dropping progressively to zero at 18.41.48 hours. Hence, contribution of speed factor towards cause of derailment was ruled out.

9.3.13 Case Study No. 13

Title

Derailment of train No. 18426 DURG-PURI Express at 20.47 hrs. of 13.02.2012 at km. 125/2-3 between NAWAPARA ROAD (NPD) AND LAKHANA stations on RAIPUR-TITLAGARH B.G. single line non - electrified section of Sambalpur Division of East Coast Railway.

Description of the accident

Train No. 18426 Durg-Puri Intercity Express derailed at 20:47 hours of 13.02.2012 at Km.125/2-3 between Nawapara Rd and

Lakhana stations on Raipur-Titlagarh section. The train derailed by 9 coaches, serially 7th from engine onwards. No POM was observed while POD was observed at 125/ 2-3. At 125/ 2-3, a weld failure and multiple rail fracture had taken place. 4.45 m of RH rail had broken into 15 pieces and it was lying along the track.

Cause of the Accident

In the Enquiry, cause of the accident was ascertained as due to discontinuity in right hand rail caused by breakage of rail in a fatigue manner initiated from gauge face corner covering about 40% of rail head area at the identified location of OBSR.

Reasons for arriving at the cause of the accident

1. Defects/deficiencies in coaches

The locomotives and leading 06 coaches had not derailed. However, the train had parted between coach No. 5 & 6 from the front and the leading portion, after moving some distance, had finally stopped, with locomotive at Km 125/12-13.

Observations/ measurements recorded for the 7th coach No. ECo 94201CN, which was the first coach to derail were as below

Several denting and sliding marks noticed on entire wheel flange (beyond wheel contact area). Height of Wearing Piece: L1-40, R1-42, L2-42 (indentation 60x65x27 mm) R-2-44. Thickness of Wearing Plate : L1-9, R1-9,L2-10, R2-10.

Further, as per Joint Measurement of coaches, the following was observed

(i) Buffer height

As per Maintenance Manual	
Empty condition	Max – 1105mm Min – 1090 mm
Loaded condition	Min – 1030mm

Hence the maximum buffer height difference between two consecutive coaches could be 75mm (1105mm – 1030mm).

As per the Joint Measurement, buffer height of 6th coach RHS was 1070mm (as against a minimum 1090mm) and 7th coach LHS was 1125mm (as against a maximum of 1105mm). These readings were found beyond permissible limits as mentioned above. However, the difference between the buffer heights of these two coaches was 55mm which is less than 75mm, hence it is permissible. Buffer entanglement between coach 6th and 7th had taken place as 7th coach had derailed and was getting dragged by 6th coach which was on rail.

(ii) Buffer length

Following are the prescribed length of buffer

Max – 635mm

Min – 584 mm

As per the Joint Measurement, buffer length of coaches was mostly within limit. Marginal deviation could be due to after effect of derailment.

(iii) Wheel gauge

As per Maintenance Manual, following are the prescribed limit

1600mm + 2mm

1600mm – 1mm

As per the Joint Measurement, maximum deviation recorded was 2.5mm (L4 and R4 of coach no. 06078) . Even though few wheel sets had gauge distance marginally beyond prescribed limits, it could not be a contributory factor for the derailment.

(iv) Spring Height

As per Maintenance Manual, following are the prescribed limit

Non AC ICF coach –

Free height	-	360mm
Height under load (2000 kg)	-	279mm to 295mm

AC ICF coach –

Free height	-	375mm

Height under load (2800 kg) - 264mm to 282mm

As per the Joint Measurement, spring height of coaches was mostly within limit. Marginal deviation could be due to after effect of derailment.

(v) Wheel flange wear

None of the wheel had excess/abnormal flange wear.

In view of the above evidences, possibility of coach defect as a contributory factor was ruled out in the Inquiry.

Further discussion

Axle box spring height – As per Coaching Maintenance Manual, axle box spring free height are 375 mm and 360 mm for AC and Non AC coach bogies. The loaded springs are to be grouped as per height as Gr A (264-269) mm, Gr B (270-275) mm, Gr C (276-282) mm for AC coach and Gr A (279-284) mm, Gr B (285-289) mm, Gr C (290-295) mm for Non AC coach. Also, as per the Manual (RDSO C - 8419 (rev1) Procedure of Maintenance for Helical Spring for Coaching), springs of the same group are to be provided on a coach. There were large variations in the loaded spring height of the axle box springs in the derailed coaches, e.g. the maximum and minimum loaded spring heights for coach no. ECo 94201 (7th from engine, Non AC) were 296 mm and 269 mm. It was discussed in the Inquiry Report that the springs were generally within the limits of loaded height. However, the issue of these springs falling in different category was not taken into account.

Measurements of crown clearance for coaches also indicated a large variation. For example, crown clearance for coach no. ECo 94201 (7th from engine, Non AC) varied between 47 – 96 mm, whereas, asper Coaching Maintenance Manual, its limit should be within 20 mm (loaded) to 47 mm (tare). Variation in crown clearance could be caused by spring defect, uneven distribution of load, or some other defect of spring assembly/ bogie. Reasons for such large variations were not analysed.

Also, buffer height on 7th coach (LHS), as described above, was more than the prescribed limit under tare condition. Reason for the same was not ascertained in the Inquiry Report. In addition, conclusions regarding vertical eccentricity of buffer height were drawn without loaded buffer height being available.

Excessive variation in spring height, associated with large crown clearance, would have caused unequal load on wheels. This , combined with vertical buffer eccentricity would have caused off-loading of some wheels. Its role in derailment was not analysed.

2. Defect/deficiency in the P.Way

Salient features of oral and material evidence were as below:

i. Crew of the previous goods train did not experience any jerk or any other abnormality in track. Crew of the derailed train also did not observe any irregularity in track till they experienced jerk while passing through Km.125.

ii. There was no Point of Mount. The Point of Drop was just at the location of first rail breakage on RHS rail between Km. 125/2-3.

iii. Right hand rail had broken into 15 pieces at Km. 125/2-3. Total length of broken rail was 4.45 m. The first location of rail breakage had a transverse crack. An AT weld was approx. 2.61 mtr ahead in the direction of movement from the first point of rail breakage. AT weld (and its vicinity) had multiple fracture. Joggle fish plates provided at the weld had wheel hitting marks.

iv. All the coaches had derailed towards right side. Left hand rail was intact and had no damage.

v. The track in rear was intact and found to have no irregularity.

vi. The location where the first rail breakage took place (between Km.125/2-3) was identified as OBSR during USFD testing on 07.01.2012. Advice was given to protect the location by joggle fish plate. USFD Test Register had the record. The location was marked with ‘red arrow symbol’. However, the location was not protected, in violation of the provisions of USFD Manual.

vii. The crew experienced jerk, which corresponded to the time when 7th coach from engine (front most derailed coach) would have been located at 125/ 2-3, the location of rail fracture.

The broken rail pieces were sent to RDSO for detailed metallurgical and chemical test. Extract of the RDSO Report is as below

"Rail had broken into multiple pieces at two locations in sudden manner and other three locations in fatigue manner. These locations had identification no. 13/12 /1, 2, 3, 4 & 5.

(i) Location 13/12 /1- Rail had broken into 7 pieces in a sudden manner. All fracture surfaces were crystalline in nature. Wheel drop was noticed on rail table.

(ii) Location 13/12 /2- This location included an AT weld and adjoining portion and had broken in 5 pieces. Fracture surfaces were crystalline in nature showing sudden failure. Rail was broken transversally in Heat Affected Zone (HAZ) from head to the transition of web-flange, and then longitudinally through flange. One piece from rail head had broken from bolt hole. Wheel drop marks were observed on rail table.

Macro examination did not reveal any internal welding defects.

(iii) Location 13/12 /3- One piece of these rails was counter to location no. 13/12 /2. Rail had broken transversely in fatigue manner, fatigue having covered about 7 % of fractured area. Battering was noticed on the fracture face.

(iv) Location 13/12 /4- One piece of the rail was counter to location no. 13/12 /3. Rail had broken in fatigue mode, initiated from gauge face side. Fatigue had covered about 30% at the fractured area.

(v) Location 13/12 /5- One piece of this location was counter to the location no. 13/12 /4. Visual examination revealed that rail had broken in fatigue manner, fatigue having covered about 40% of the rail head area. The transverse fissure in the head was inclined at an angle approx. 5 deg with vertical (Figure 9.2). This defect could have been detected at the time of last Ultrasonic testing as an OBS category.

Conclusion - Breakage of rail in fatigue manner initiated from gauge face corner covering about 40% of rail head area may be attributable to presence of shear cracks, present on rail table towards gauge face side. The other breakages are the subsequent effect of this failure."

Other important items brought to light in the Inquiry 'D' Marked Rail - The section is laid with 52 kg/90UTS rail rolled in the year 1996/97. All the rails belong to 'D' marked rail.'D' marking of rails were done at Bhilai Steel plant for rails manufactured prior to year 1999. This marking signifies that dispensation with regard to any one or more parameters of IRS T-12 were granted while inspecting the rails. In this specification dispensation was granted upto 31.12.1999 for 'Hydrogen content' End straightness' and 'On line ultrasonic testing'. The dispensation on 'Hydrogen content' was permitted in 52kg rails only and no dispensation was given for 60kg rails.

As per A & C Slip No. 7 of January, 2012, " Digital Double Rail Tester is to be used for testing 'D' Marked rails at reduced interval to be decided by CTE of Zonal Railway".

Number of Rail fracture and weld failure in Raipur - Titlagarh Section – There had been a very large number of rail fracture/ weld failure on this section in the recent years.

Important Recommendations

1. On the basis of USFD Test Report, protection of OBSR and OBSW locations must be protected as per provisions

of the Manual on Ultrasonic Testing of Rails and Welds.

2. Proper documentation of USFD Test Report and maintenance of Master Register for defects detected and rail/ weld fracture as per provisions of the Manual on Ultrasonic Testing of Rails and Welds to be ensured.

3. A close monitoring system should be introduced to continuously review the 'Action Taken' on the Report of USFD tests at AEN and Sr DEN level.

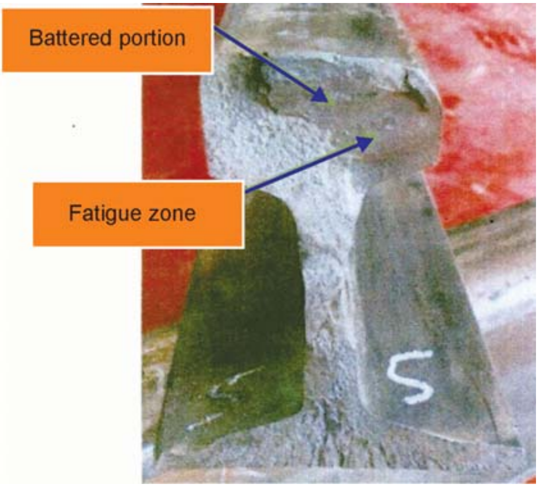


Fig. 9.2 Fractured rail face

9.3.14 Case Study No. 14

Title

Derailment of 13226 UP DANAPUR JAYNAGAR INTERCITY EXPRESS in BACHHAWARA station yard of SONPUR DIVISION of EAST CENTRAL RAILWAY on 12.09.2012 at 11.12.20 Hrs.

Description of accident

Train No.13226 Up Danapur Jaynagar Intercity Express comprising of 10 coaches worked by Diesel Locomotive no. 16317

WDM3A/KGP arrived Bachhawara station on line no 4 (Up loop) at 11.08 hrs on 12.09.2012 for its scheduled stoppage. The train started after 2 minutes on proper signals, cleared the loop line and while moving on the mainline towards Samastipur Junction (SPJ), the rear trolley of the last coach (GSLR no EC 92725) took a wrong route towards Hajipur through Point no 23. At that point of time, the speed of the train was about 30 kmph. Due to pull of the train engine, wheels of the rear trolley (which had taken wrong route) mounted on to the rails, jumped off and got dragged towards the track to SPJ, resulting in the coach body hitting against two OHE masts coming in the path. The cross pull due to derailment of the rear trolley and obstruction encountered by the moving coach caused derailment of the front trolley also and resulted in entanglement of front buffer of the GSLR coach with the adjoining coach. During the movement of the coach in derailed condition encountering obstructions by OHE masts, passengers sitting on left side of the coach got hurt on hands and fingers causing grievous injury to 04 of them.

Cause of accident

The derailment of 13226Up Danapur Jaynagar Intercity Express in Bachhwara station yard on 12.09.2012 was caused on account of the train taking two routes at Point no 23 due to manipulation done by railway staff of the station by changing the position of the Point while the train was on run.

Reasons for arriving at the cause of the accident

SWRD of Bachhwara Jn showing the Track circuits and Signals is shown in **Fig.** 9.3. (a) and (b) Output of the Data Logger at Bachhwara for the relevant period and interpretation of the data is placed in **Table** 9.1 and 9.2

Wrong setting and locking of the Points due to interlocking failure - In the Inquiry Report, Data logger recording was analysed for 04 previous trains viz 18181Up, 55525 Pgr, 53041Up and 13019Up (1st train before 13226 Up to 4th train) which took the same route via PF line 4 on 12.09.2012 as the ill-fated train. Operation of various relays was found to be in correct sequence for the three trains viz 55525 pgr, 53041 Up and 13019 Up. However, for the 1st previous train 18181 Up, manipulation of the track circuit 21BT was noted as the track circuit A2T, which was ahead, was showing occupied before the previous track circuit 21BT.

As per the Joint Note of Sr. Supervisors, after the accident the track circuit 21BT in the Point zone was locally made red i.e. occupied and operation of Point 23 was attempted, which was not successful. This indicates that there was proper interlocking in place, preventing operation of Point 23 when the Point track circuit zone 21BT was in occupied position. In view of this, wrong setting and locking of the Points due to interlocking failure was ruled out.

Mismanagement in operation of Points during the run of the train - 13226 Up was dispatched to Sathajagat station ex Bachhwara station from PF line no.4. The train engine and all its 9 front coaches took the correct route, whereas the front trolley of the rear most coach EC GSLR 92725 took the correct route but the rear trolley took the track towards vidyapatinagar from point no. 23.

For movement of a train from PF Line no 4 of Bachhawara station towards Sathajagat, point 23A has to be set and locked in normal position, and for movement towards vidyapatinagar it has to take reverse position. This accident was caused as during the movement of 13226 Up, Point 23A was initially set to normal position and, while rear trolley of the EC GSLR 92725 was approaching the point 23, it took a reverse position, causing two route for the coach. However, after the accident, the Point no 23 was found to be in normal position by the team of multidisciplinary senior supervisors as per observations recorded in their Joint Note. This points out to a situation where position of Point no.23 changed from normal to reverse and again from reverse to normal during the movement of the train over it. This type of changeover of position of point 23 cannot happen in normal course of working. As per the circumstantial evidence, it was suspected that there was mismanagement in operation of Point no 23 during the run of the train.

Analysis of Datalogger recording of the station revealed the

SWR Diagram - 1

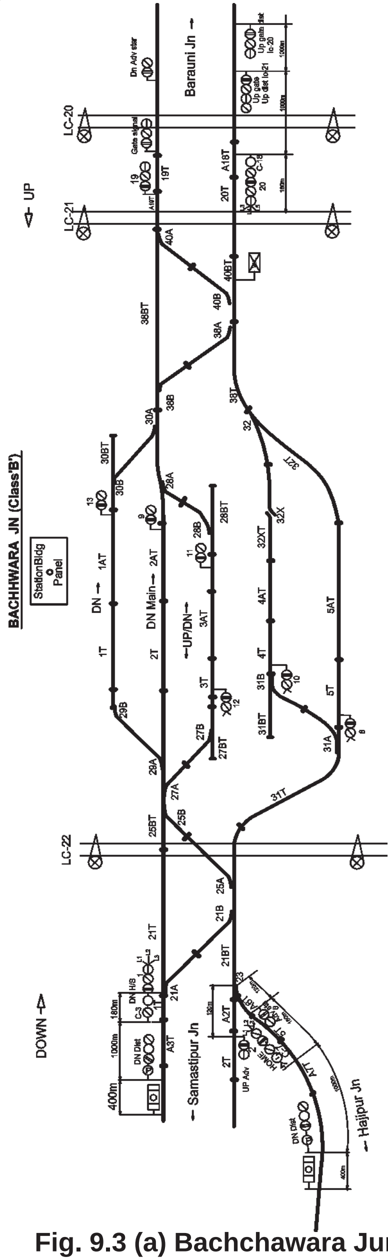


Fig. 9.3 (a) Bachchawara Junction

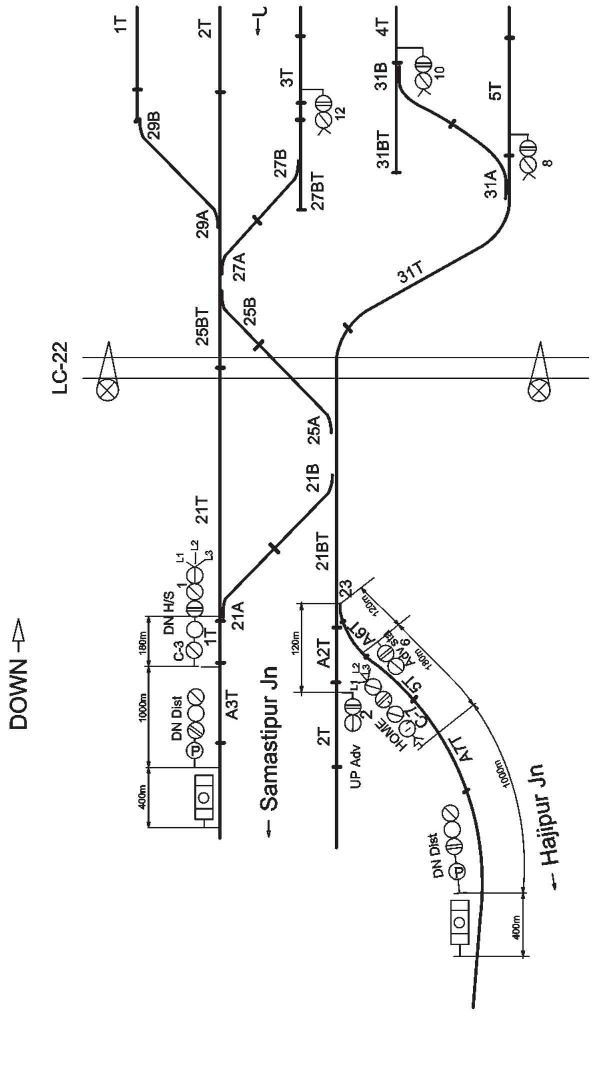


Fig. 9.3 (b) Bachchawara Junction

Table 9.1

Digital Report - Bachchawara

SN	Signal (Relay) name	Signal (Relay) Status	Time	Nomenclature	Remarks
1191	S-10 HR	Up	11.12.40.078	Sig. no. 10 yellow aspect controlling relay	Control extended to home sig. no. 10 for yellow aspect
1192	10/G/ECR	Drop	11.12.40.125	Sig. No. 10 lamp checking relay	Sig. no. 10 blank momentarily
1193	S-10 HPR	Up	11.12.40.187	Repeater of HR	Repeater of S-10 HR
1194	SA NR	Drop	11.12.40.281	Normal relay	Points are free to set to reverse
1195	10G/ECR	Up	11.12.40.859	Sig. 10 Lamp Checking Relay	S- 10 is not Blank
1196	32RWLR	Up	11.13.03.313	Reverse Point Locking Relay	Point no. 32 is free to be operated from Reverse to Normal
1197	32 NWR	Up	11.13.04.375	Normal Point operation Relay	When UP its front contacts are used in normal operation
1198	32 RWKPR	Drop	11.13.04.406	Reverse Point Indication Relay	Reverse point Indication Relay drops i.e. point is no more in Reverse

1199	32 WJR	Up	11.13.04.484	Point Operation Time Limiting Relay	Overload protection Relay. Drops and cut off the power supply to point in case point remains in overload above specific time (Say 10 sec.)
1200	32 XR	Up	11.13.04.656	Special Relay	Siding control relay picks up, drops with E-type lock contacts.
1201	32XR	Drop	11.13.08.016	Normal Point operation Relay	
1202	32 NWR	Drop	11.13.08.016	Reverse Point Indication Relay	Normal point operation is over so NWR drops.
1203	32 NWKR	Up	11.13.08.031	Normal Point Indication Relay	Point is set and locked in Normal
1204	32 WJR	Drop	11.13.08.156	Point Operation Time Limiting Relay	Point operation from reverse to normal is over, so concern red WJR drops.
1205	32 NWKPR	UP	11.13.08.172	Normal Point Indication Repeater Relay	Repeater of NWKR
1206	31 BTTPR	Drop	11.13.46.047	Track Proving Relay	Train occupies the track circuit

1207	S-10 HPR	Drop	11.13.46.062	Sig. no. 10 controlling Repeater Relay	Sig. Put back to danger by as train occupied the Signal controlling track
1208	S-10 HR	Drop	11.13.46.062	Sig. no. 10 Yellow aspect controlling relay	circuit i.e. 31 BT
1209 1210	31 BT TSR 31T TSR	Drop Drop	11.13.46.062 11.13.57.781	Track Stick Relay for Track Circuit No. 31 B	This relay pickup and Drops simultaneously with Track Relay when the track is shunted.
1211	31T TPR	Drop	11.13.57.813	Track Proving Relay	Train occupies the track circuit No. and 31
1212	S-10 UYR-I	UP	11.13.57.938	Route section Approach Lock Relay No.1	This Relay picks up when train advances on proper signal and occupies the tracks sequentially.
1213	UP TOLR	Drop	11.14.03.656	Train On Line Relay	This Relay is related to Block Instrument. Train entered into Block section.
1214	UPBSR	Drop	11.14.03.688	Block Instrument Relay	Block Instrument Relay
1215	S-10 UCR	Drop	11.14.06.703	Route Checking Relay	Drop means route is locked for train movement.
1216	04T TPR	UP	11.14.29.984	Track Proving Relay	Track Circuit No. 04 is non-occupied

1217	04/04 ATPR	UP	11.14.30.094	Track Proving Relay	Track Circuit No. 04/04A is non-occupied
1218	31 BT TPR	UP	11.14.38.656	Track Proving Relay	Track Circuit No. 31 B is non-occupied
1219	3H/D YR	Drop	11.14.44.375	Slot Relay	Slot Relay
1220	31T TPR	UP	11.14.50.734	Track Proving Relay	Track Circuit No. 31 is non-occupied
1221	31T TSR	UP	11.14.51.047	Track Stick Relay for Track Circuit No. 31	This relay pickup and drops simultaneously with track relay when the track is shunted
1222	A2 T TPR	Drop	11.14.53.875	Track Proving Relay	Track Circuit No. A2 is occupied by train.
1223	21BT TPR	Drop	11.14.54.953	Track Proving Relay	Track Circuit No. 21B is occupied
1224	21BT TP2R	Drop	11.14.54.953		
1225	S-10 UYR-2	UP	11.14.54.953		
1226	21BT TPR	UP	11.14.57.313	Track Proving Relay	Track Circuit No. 21 is non-occupied
1227	21BT TP2R	UP	11.14.57.547		
1228	21N/RWL-ZR	Drop	11.14.57.594	Special Relay for Point	Some relays are locally designed as

				Operation	per their local requirement
1229	25 N/R WL -ZR	Drop	11.14.57.625		
1230	31 N/R WL -ZR	Drop	11.14.57.656		
1231	31 RWLR	UP	11.14.57.656	Reverse point locking relay	Point No. 31 is free to be operated from Reverse to Normal
1232	21 NWLR	UP	11.14.57.656	Normal point locking relay	Point No. 21 is free to be operated from Normal to Reverse.
1233	S-19 DECPR	Drop	11.14.57.656	Green Aspect Lamp Checking Relay	Green Aspect of Sig. No. 19 is no more Green.
1234	23 NWLR	UP	11.14.57.656	Normal point locking relay	Point No. 23 is free to be operated from Normal to Reverse.
1235	31BT TSR	UP	11.14.57.672	Track Stick Relay for Track Circuit No. 31	This relay pickup and drops simultaneously with Track Relay when the track is shunted.
1236	25 RWLR	UP	11.14.57.687	Reverse point locking relay	Point No. 25 is free to be operated from Reverse to Normal.

1237	25 NWLR	UP	11.14.57.687	Normal point locking relay	Point No. 25 is free to be operated from Normal to Reverse.
1238	S-10 ASPR	UP	11.14.57.703	Repeater Approach Lock Stick Relay	Route for Sig. No.1 0 is released.
1239	S-10 UYR-1	Drop	11.14.58.313	Route section approach lock relay-1	This relay drops when previously set route is released.
1240	S-10 UYR-2	Drop	11.14.58.375	Route section approach lock relay-2	This relay drops when previously set route is released.
1241	UP LCR	UP	11. 15.00.312	Line Clear Relay (Block Instrument Relay)	Relay related to block instrument
1242	23 RWLR	UP	11.15.08.984	Reverse point locking relay	Point No. 23 is free to be operated from Reverse to Normal.
1243	23 RWR	UP	1l.15.09.172	Reverse point operation Relay	When up its Front contacts are used for Reverse operation
1244	23 WJR	UP	1l.15.09.266	Point operation time limiting relay	Overload protection Relay. Drops and cut off the power supply to point in case point remains in overload for specific time (say 10 sec.)
1245	23 XR	UP	1l.15.09.438	Special Relay.	Siding Control Relay picks up

1246	23 XR	Drop	1l.15.12.234		Drops with E-type lock contacts.
1247	23RWR	Drop	1l.15.12.234	Reverse point operation relay	Reverse point operation is over so RWR Drops.
1248	23 WJR	Drop	1l.15.12.375	Point operation time limiting relay	Overload protection relay drops and cut off the power supply to point in case point remains in overload for specific time (say 10 sec.)
1249	23 RWKPR	UP	11.15.12.375	Reverse point indication relay	Reverse point Indication Relay up. i.e. point is set in Reverse.
1250	25RWR	UP	1l.15.12.375	Reverse point operation Relay	When up its Front contacts are used for Reverse operation
1251	25 NWKPR	Drop	11.15.12.422	Normal point indication repeater relay	Normal point Indication Relay drops. i.e. point is no more in Normal.
1252	25WJR	UP	1l.15.12.516	Point operation time limiting relay	Overload protection Relay. Drops and cut off the power supply to point in case point remains in overload for specific time (say 10 sec.)
1253	25 XR	UP	1l.15.12.641	Special Relay.	Siding Control Relay picks up Drops with E-type lock contacts.

1254	29 RWLR	UP	11.15.13.516	Reverse point locking relay	Point No. 29 is free to be operated from Reverse to Normal.
1255	29 RWR	UP	11.15.13.656	Reverse point operation Relay	When up its Front contacts are used
1256	29 NWKP2R	Drop	11.15.13.719	Normal point indication repeater relay	Normal point Indication Relay drops. i.e. point is no more in Normal.
1257	29WJR	UP	11.15.13.812	Point operation time limiting relay	Overload protection Relay. Drops and cut off the power supply to point in case point remains in overload for specific time (say 10 sec.)
1258	29XR	UP	11.15.13.984	Special Relay.	Siding Control Relay picks up and Drops with E-type lock contacts.
1259	25RWR	Drop	11.15.15.469	Reverse point operation Relay	When drop its Back contacts are used for Normal operation
1260	25 XR	Drop	11.15.15.469	Special Relay.	Siding Control Relay picks up Drops with E-type lock contacts
1261	25 RWKPR	UP	11.15.15.609	Reverse point indication repeater relay	Reverse point Indication Relay up. i.e. point is in reverse position.

1262	25WJR	Drop	11.15.15.609	Point operation time limiting relay	Overload protection Relay. Drops and cut off the power supply to point in case point remains in overload for specific time (say 10 sec.)
1263	SNR	Drop	11.15.15.844		
1264	5 RCR	UP	11.15.15.844		
1265	2T TPR	Drop	11.15.16.297	Track Proving Relay	Train occupies the track circuit No. 2T
1266	S-2 DR	Drop	11.15.16.312	Sig. no. 2 Green Aspect controlling relay	Control extended to Sig. No.2 for Green aspect.
1267	S-2 DPR	Drop	11.15.16.328	Repeater of DR	Repeater of DR
1268	S-2 DZR	UP	11.15.16.453	Green aspect power supply checking relay	Green aspect power supply checking relay
1269	S-2 DECPR	Drop	11.15.16.469	Green aspect lamp checking relay	Green aspect of Sig. No.2 is extinguished.
1270	29 RWR	Drop	11.15.17.109	Reverse point operation Relay	When drop its Back contacts are used for Normal operation

1271	S-2 RECR	UP	11.15.17.109	Red aspect lamp checking relay	Red aspect of Sig. No.2 is litting.
1272	29XR	Drop	11.15.17.125	Special Relay.	Siding Control Relay picks up Drops with E-type lock contacts.
1273	29 WJR	Drop	11.15.17.250	Point operation time limiting relay	Overload protection Relay drops and cut off the power supply to point in case point remains in overload for specific time (say 10 sec.)
1274	S-2 SR	Drop	11.15.17.250	Stick relay for Sig. No. -2	Stick relay for Sig. No.2
1275	29 RWKPR	UP	11.15.17.266	Reverse point indication repeater relay	Reverse point Indication Relay up. i.e. point is in reverse position.

Table 9.2

Nomenclature of Symbols

Symbol	Nomenclature	Symbol	Nomenclature
A	Approach	N	Normal, button
B	Block	P	Repeater, Proving
C	Checking	R	Relay, Reverse
D	Green	S	Stick
E	Lamp	T	Track
G	Signal	U	Route
H	yellow	W	Point
I	Indicator	X	For special purpose
J	Timer, Time controlling	Y	Slot
K	Indication	Z	Zone
L	Lock, locking	N	Normal, button
M	Main		

following abnormalities:

i) Track circuit 21BT, which controlled Point (23) zone, was noted to be flickering during the time 13226 Up was on move after its departure ex Bachhwara station. During the period from 11.03.12 to 11.14.57 in about 12 minutes, the track circuit 21BT was shown occupied and unoccupied six times for durations of nearly zero second to 03 seconds.

Earlier also (between 10.26.30 and 10.33.33 hrs.), track circuit 21BT was shown occupied and unoccupied five times for durations of nearly zero second to 05 seconds. This was not the normal working situation and indicated that the track circuit 21BT was not functioning normally due to some failure.

ii) 13226 UP, while moving, occupied track circuits 31BT and 31T at 11.13.46.047 (s.n. 1206) and 11.13.57.813 (s.n. 1211).

The next track circuit on the route was 21BT and the corresponding track relay should have dropped as the train moved from 31T and occupied 21BT. However, track circuit A2T ahead got occupied at 11.14.53.875 (s.n. 1222) before the previous track circuit 21BT, which was shown occupied between 11.14.54.953 (s.n. 1223) and 11.14.57.313 (s.n. 1226). This is not the normal sequence and cannot happen unless some manipulation was resorted to in the operation of relays.

Further, the track circuit 21BT is shown occupied only for 03 seconds, which is not possible for a train length of 10 coaches moving at a speed of 30 kmph max.

iii) Point no.23 was operated from normal to reverse position at 11.15.08.984 (s.n. 1242) and it was finally set to reverse at 11.15.12.375 (s.n. 1249). Probably, this was the time when rear trolley of the derailed coach took the route with point no. 23 in reverse. Point no. 25 and 29 were also set to reverse at 11.15.15.609 (s. n. 1261) and 11.15.17.266 (s.n. 1275).This was perhaps done to receive another train from Vidyapatinagar side.

iv) After the derailment, point no. 23 was again operated back to normal position at 11.15.58.375, probably to restore position as per original movement.

The above anomalies in functioning of interlocking circuit relays make one not only suspect the mismanagement in operation of point no.23 but willful interference gets confirmed.

Reasons for manipulation of the relays - As discussed, the track circuit 21BT was erratically behaving for some time before arrival of 13226 Up. This fact was known to SS/Bachhwara and ESM at the station, as established through interrogation in the Inquiry. To avoid detention and to save themselves from painstakingly lengthy working procedure in the event of failure of track circuit, they jointly decided to go for shortcut by getting the relays operated manually from the relay room, to enable them to dispatch the train. They succeeded in dispatching the train but in a hurry to receive 15210 Dn, which was waiting at the Home

Signal from Vidyapatinagar side, assumed that 13226 Up had cleared the Point zone and made 21BT track free manually and reversed the Point 23 before complete clearance of the train from the Point zone.

Case Study No.15

Title

Derailment of Train No. 13152 (JAT-SDAH Express) on 10/11-03-2011 at Suchipind Station of Firozpur Division of Northern Railway.

Description of the accident

On 10/11/03.2011, Train no.13152 Dn (JAT-SDAH) left AWL station at 00.03 Hrs. This train was being received on line no.2 at Suchipind. At about 00.10 Hrs., while train was midway in the yard, the loco pilot of the train felt a jerk and he stopped the train. The rear trolley of coach no.9 from engine was found to have derailed and was on line 1, and the front trolley, which had not derailed, was on Line no.2. Coach No.10 from engine had derailed by both trolleys. The train behind coach no. 10 and up to the SLR was standing on the route of line no.1. There were damages to track and signaling gears. The train left for SDAH at 04.50 Hrs. on the same day from JUC Station.

Cause of the accident

From the position of the derailed train, Two Road condition got established. The main cause of the accident was the wrong wiring between contact no. D-4 of Relay no.130 NWKR to contact D-5 of Relay no.135 RWKR, which gave feed to a circuit and this operated point No. 135 A, B to set it for Line No. 1 as soon as emergency route release was done by operation of EUUYN.

Apart from this, the operation of EUUYN was done without physically verifying that the operation of train reception was complete to ensure that the track was not occupied. Distance of point No.135 from the station building was small and ASM could have clearly seen the train, even in the night time.

Reasons for arriving at the cause of the accident

Suchipind (SCPD) station is 3.38 km from JUC station and 4.70 km from JRC station, thus, forming apex of the triangle of lines between the three stations i.e. JUC, JRC and Suchipind. There is a permanent speed restriction of 90 kmph from km 4/0 to 4/4 due to sharp curve and 30 kmph from km 3/1 to 2/9 due to negotiation of turnout no. 107 on Main Line in suchipind yard. Suchipind is a B-class, MACL standard III interlocking station, and is having six running lines. Yard layout of the station is shown in **Fig.** 9.4.

Report of Sr.DSTE/FZR: SCPD station has standard III Route Relay Interlocking with 102 routes. The station RRI was taken over by Open Line from Construction on 8.3.2007. He had noted on arrival at site that the Emergency Key of ESM was out. He has further stated that the route set for line no.5 for this train was cancelled. Later on, route was set for line no.2 for this train and again the route for line no.2 was cancelled by emergency button which caused movement of point number 135-B for line No.1, thus, creating two roads situation while the train was crawling on the point no. 135-B.

Sr.DSTE further stated that the construction organization did not test the last 14 routes, including Calling on routes of signal 160 A, B, in the selection table, and this was not signed by officer in-charge. Moreover, normal cancellation by station master was not working and route of line no.1 and 2 did not get released after arrival of train. These defects were neither recorded nor reported in Signal Failure Register.

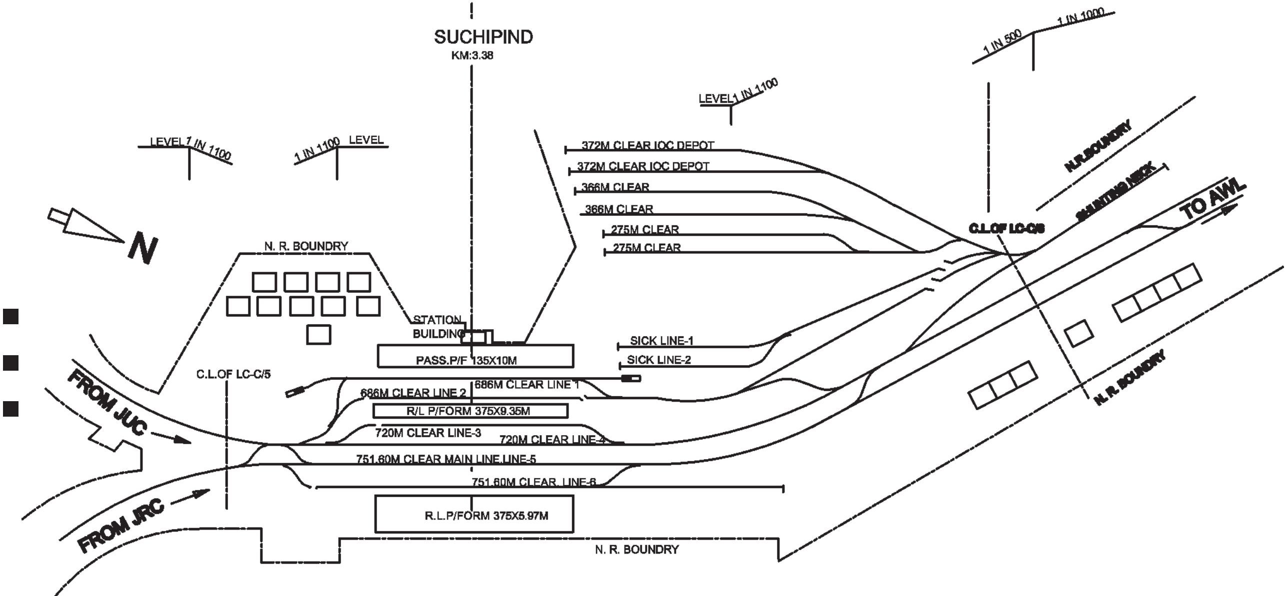
Data Logger Report: This indicated that EUUYNPR & EUUYNR relays were operated at 00.09.52:859 and 00:09:52:875 respectively on 11.03.2011, and 2 seconds after this operation, feed was extended to point no. 135 for its operation from N to R. The cross over point 135 operated at 00:09:58:158. Thus, with this operation of point, Two Road condition was created.

Transcript of the conversation between Dy.CHC and ASM/ SCPD: The recorded voice of the section controller and ASM indicated that, initially, the train was to be received on the Main line. Subsequently, the planning was changed and reception of this train was changed to line no.2, so as to give way to train no. 14036 and for passage of train no. 14002 from JRC. Thereafter, the controller advised the ASM to let the train 13152 pass through. However, the ASM responded by saying that the train will go after a stoppage of 2 minutes. Afterwards, the ASM informed the Controller of the derailment.

Simulation of the derailment: Members of the Enquiry Committee conducted a simulation trial at the Station on 19.04.2011. After doing the wrong wiring, it was observed that the calling on signal was not coming. Also, when line no.2 was in occupied state and emergency route release was operated, point No. 135-B got reversed. This established that the movement of point under wheels, leading to two road condition, took place due to wrong wiring between contact no. D-4 of Relay 130 NWKR to contact no.D-5 of relay 135 RWKR.

Dy.CSTE/Const Report : Sanction for SCPD RRI was granted by CRS on 24,05.2006 and Suchipind-Bhogpur-Sirwal section was inspected by CRS on 31.08.2006 and 1.9.2006. However, original tracings of completion wiring diagram for SCPD were handed over to Open Line only on 14.2.11, after reminders were given to construction organization. Thus, there was a serious delay of about five years in making the records of the works executed in the year 2006. It was stated by Dy.CSTE/Con that complete selection table test and functional test were done before commissioning of SCPD station. However, he could not explain satisfactorily the absence of signature of the authorized official on last two sheets of selection table.

Signal Engineering Manual part – II: As per the provision of para 13.40.2 of Signal Engineering Manual Part-II, a panel station should be periodically tested and system test should be conducted every three year or before. It is apparent that since the installation of the panel interlocking at SCPD station, this panel was not tested as per provision of SEM. Otherwise, the wrong wiring could have been detected well in time this accident could have been avoided.

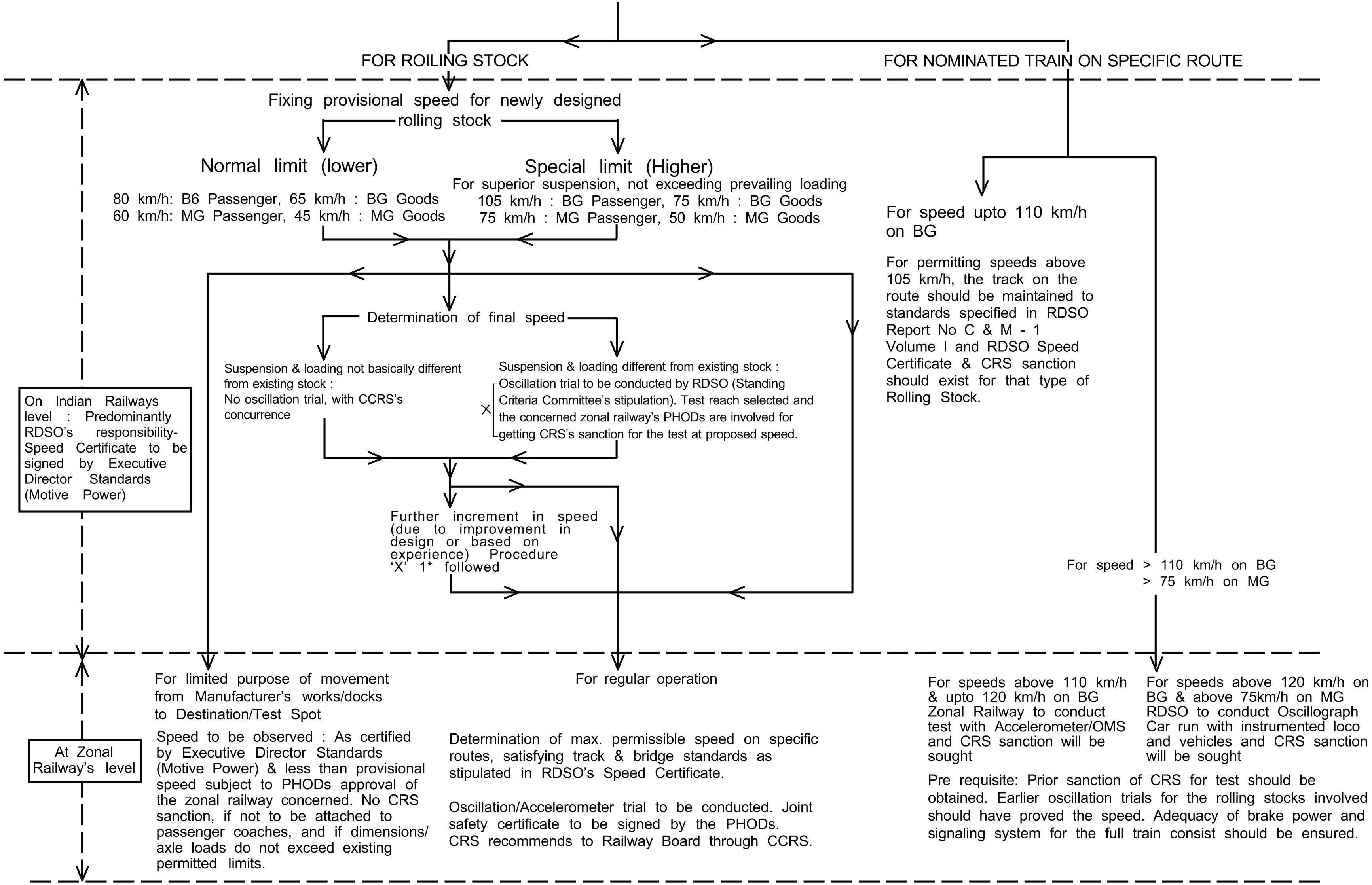


NORTHERN RAILWAY FIROZPUR DIVISION
JUC - PTK SECTION
SUCHIPIND STATION YARD

Fig. 9.4 Suchipind Yard Plan

ANNEXURE - A

CERTIFICATION OF MAXIMUM PERMISSIBLE SPEED



Documents for further reference :

i) Policy Circular No. 6 issued by Rly. Bd. under No. 92/CEDO/SFW/O Pt. dt. 23-12-99 and modifications issued from time to time. (Revised vide letter even no. dated 23-11-2006)

ii) Standing Criteria Committee recommendations for clearing rolling stocks (latest and updated)

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