Basic requirements of Bogies

- STATUTERY (TRACK GAUGE, GROUND CLEARANCE, LATERAL DEFLECTION, SHARPEST TRACK RADIUS)
- RELIABILITY AT CRITICAL OPERATION SPEED
- GUIDE THE VEHICLE ON STRAIGHT TRACK WITH STABILITY
- SMOOTH CURVE NEGOTIATION WITHOUT SKIDDING
- EFFICIENT BRAKING FOR EMERGENCY STOPPING DISTANCE
- ADEQUATE RIDE COMFORT FOR TARE AND GROSS LOAD
- SAFETY AGAINST DERAILMENT Y/Q
- LOW MAINTENANCE COST – NO REPLACEMENT BEFORE IOH/POH
- ECONOMICAL PRODUCTION PROCESS.
VEHICLE DEGREE OF FREEDOM

CAR BODY

X- SHUTTLE
Xr- ROLL

Y- LURCH
Yr- PITCH
Z- BOUNCE
Zr- YAW
Almost all railway vehicles consist of carbody, bogie frames and wheelsets. Therefore an improvement of the model above is to introduce as second suspended mass, i.e. to introduce a second vertical degree of freedom.

The three force equations of the system with respect to the static equilibrium can be written as

\[ m_c \ddot{z}_c + c_2(\dot{z}_c - \dot{z}_b) + k_2(z_c - z_b) = 0 \]  
\[ m_b \ddot{z}_b - c_2(\dot{z}_c - \dot{z}_b) - k_2(z_c - z_b) + c_1(\dot{z}_b - \dot{z}_w) + k_1(z_b - z_w) = 0 \]  
\[ m_w \ddot{z}_w - c_1(\dot{z}_b - \dot{z}_w) - k_1(z_b - z_w) = -Q_{dyn} \]

Using Equation (5-2) one gets the two equations of motion in matrix form as

\[ \begin{bmatrix} m_c & 0 \\ 0 & m_b \end{bmatrix} \begin{bmatrix} \ddot{z}_c \\ \ddot{z}_b \end{bmatrix} + \begin{bmatrix} c_2 & -c_2 \\ -c_2 & c_1 + c_2 \end{bmatrix} \begin{bmatrix} \dot{z}_c \\ \dot{z}_b \end{bmatrix} + \begin{bmatrix} k_2 & -k_2 \\ -k_2 & k_1 + k_2 \end{bmatrix} \begin{bmatrix} z_c \\ z_b \end{bmatrix} = 0 \]

or in short form

\[ M \ddot{x} + C \dot{x} + Kx = F \]
WHY VEHICLE DYNAMICS.

VEHICLE DYNAMIC SIMULATIONS HELPS TO PRE INVESTIGATE THE STATIC, QUASISTATIC AND DYNAMIC BEHAVIOR OF RUNNING VEHICLE SYSTEM IN DIFFERENT DYNAMICS CONDITIONS, VARYING LOAD WITH SPECIFIED GEOMETRY CONDITIONS OF TRACK AND WHEEL-RAIL INTERACTION. TO OPTIMIZE THE SUSPENSION CHARACTERISTICS FOR DESIGNED CRITICAL SPEED OF VEHICLE WITH SAFE & IMPROVED RIDING.

QUASISTATIC: WHEN VEHICLE RUNS WITH CONSTANT SPEED ON IDEAL TRACK WITH CONSTANT CURVE RADIUS, CANT AND WHEEL RAIL FRICTION.

QUASISTATIC LIMIT FOR LATERAL FORCES IN CURVES = 60 KN (V=5.4 KMPH) AS PER UIC-518.
Rigid Bodies with lumped mass

Bushing, Bump and rebound stops
Direction elements Damper and Spring

PWL Characteristics

Initial calculations-Run simulation -Review-results-
Optimise values as per results - Re iteration till desired optimized output
<table>
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<td><strong>LATERAL WHEEL FORCES</strong></td>
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<td>Comfort values</td>
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Carbody bending frequency -10 Hz
Pitching of Bogie -15 Hz separated by \( \sqrt{2} \)
Max. Buffer drop tare to gross \( \leq 75 \text{mm} \)
Tilting coefficient less than 0.4
LATERAL CLEARANCE MEANS POSSIBLE DISPLACEMENT OF A WHEEL UNTIL FLANGE CONTACT IS REACHED. BECAUSE OF THIS LATERAL PLAY BOGIE MOVES IN SINE WAVE MOTION.
Sinusoidal motion of wheel set

- A cylindrical wheel with minor disturbance will take an extreme position and will never return back towards center line on its own.
- Taper on tread of 1:20 gives self centering effect and changed rolling radius works as differential on curves.
Thus, a typical periodic motion is generated with time period \( = \frac{\lambda}{v} \)

where \( v \) is the velocity of the vehicle

Lateral displacement will be \( y = a \sin \omega t \),

\( \omega \) being angular velocity of oscillations being the time when displacement is measured

Thin flange increase Lateral Play
KLINGEL’S FORMULA

Wave Length $\lambda_0$ of a Single wheel

$$\lambda_0 = 2\pi \sqrt{\frac{rG}{2\gamma}}$$

$G$ = Dynamic Gauge
$r$ = Dynamic Wheel Radius
$\gamma$ = Conicity

$$\lambda_0 \propto \frac{1}{\sqrt{\gamma}} ; \quad \text{Frequency} \propto \sqrt{\gamma}$$

Hollow tyre increase conicity of wheel

LARGER CONICITY RESULTS IN LOWER WAVE LENGTH AND HIGH FREQUENCY RESULTING IN STABILITY

--- LOWER CRITICAL SPEED IN BOGIE HUNTING---
Effective conicity evolution due to wear

- Equivalent effective conicity characterising the wheel rail contact geometry
  
- Effective conicity \(=\ \frac{\delta r}{\delta y}\)
- \(\delta r\) – change in rolling radius
- \(\delta y\) – lateral displacement

- High equivalent conicity \(\rightarrow\) more bogie hunting

- New wheel = 0.25 ; Worn wheel = 0.4 to 08
- For Ultra high speed conicity = 0.1 with tread taper 1:40
SECTIONAL PLAN OF WHEEL FLANGE AT LEVEL OF FLANGE TO RAIL CONTACT

Due to lateral clearance between wheel & track, Axle may assume intermediate angular position.

On curves wheel obliquity is accentuated in proportion to ratio of wheel base and radius of curvature.

Sectional plan of wheel flange at level of flange to Rail contact
POSITIVE ANGULARITY (PLAN)

- Point of contact of flange is ahead of tread contact
- Frictional force acts upwards thus adds in wheel climbing the rail
NEGATIVE ANGULARITY (PLAN)

- Flange contact lags (trails) tread contact
- Frictional force acts downwards
CONCLUSIONS FROM ANGULAR MOVEMENTS

- Frictional force acts upwards and acts as a derailing force in case of positive angularity.
- Derailment proneness is higher when wheel makes contact with positive angle of attack.
- Positive angularity is therefore most critical condition of the three possible conditions –
- POSITIVE—ZERO--NEGATIVE
LARGE LATERAL DISPLACEMENT ARE LIMITED BY WHEEL FLANGES RUBBING AGAINST THE SIDE OF RAIL. THE FLANGES ALSO PROVIDES REACTION FORCES TO TURN THE BOGIE AROUND A CURVE TRACK.

Derailing force  =  \( Y \cos \beta + \mu R \)

Stabilising force  =  \( Q \sin \beta \)
Nadal’s Equation

For Safety against wheel climbing:
LHS has to be small than RHS
Y → Low
Q → High
μ → Low
tan β → Large

DERAILMENT CO-EFFICIENT SHOULD BE LESS THAN = 0.8 FOR SAFETY
Salient features – Eurofima FIAT bogie

DIP Y FRAME – TO LOWER CENTER OF GRAVITY

SHORTER WHEEL BASE 2560 mm – FOR BETTER CURVE NEGOTIATION WITHOUT WHEEL SKIDDING

FLEXICOIL SOFTER SECONDARY SUSPENSION :- FOR BETTER RIDE QUALITY IN VERTICAL & LATERAL DIRECTION

CARTRIDGE TAPER ROLLER BEARING :- FOR BETTER LIFE CYCLE AGAINST AXIAL LOADS & EASE OF FITMENT

ANTI ROLL BAR – TO CONTROL ROLE FREQUENCY & DISPLACEMENT

DISC BRAKE ARRGT. – FOR SHORTER EMERGENCY STOPPING DISTANCE

YAW DAMPER – TO SUPRESS HUNTING FORCES

LATERAL & LONGITUDINAL BUMP STOP, CURVE ROLL - TO CONTROL COACH MOVEMENT WITH RESPECT TO BOGIE.

BOGIE BODY CONNECTION – FOR ISOLATION OF NOISE AND VIBRATIONS AND NON DETACHMENT OF SHELL – BOGIE DURING DERAILMENT.
(With Bolster Lifted)

- Primary suspension
- Secondary suspension
- Yaw Damper
- Bogie Bolster
- Cross Section Frame
Wheelset ---

Why condemning limit = 845 mm

- Brake disks dia. = 640 mm.
- Wheel discs Dia = 915 (New), 845 (worn).
- New wheel radius = 915/2 = 457.5 mm
- Brake disc radius = 640/2 = 320 mm
- Ground clearance mandatory = 102 mm
- m=Margin = 457 – (320+102) = 35 mm
- Condemning limit = 915 – (35×2) = 845 mm

- Variation allowed in size of wheel discs
- Wheel disc one axle = 0.5 mm
- Wheel disc one bogie = 5 mm
- Wheel disc one coach = 13 mm

W.I. as RCF MDTS 168
Dynamic balance at 320 rpm
Unbalance moment should be <= 50 gm.
3M self adhesive strip or Glue weights

Chisel Gasket Remover-Loctite 79040
Activator- Loctite 7075
Adhesive Loctite 324
Axle bearings

- CARTRIDGE DOUBLE ROW TAPER ROLLER BEARING
- PRE-ASSEMBLED PRE GREASED SEALED UNIT.
- MAINTENANCE FREE-
- FIRST OVERHAUL = 1.2 MILLION KM.
- SERVICE LIFE = 3.0 MILLION KM

Bearing Mounting Pressure
Timken- 20-25 Tonne
SKF- 28-32 Tonne
Needs to be standardised

Mounted End Play Checking with Dial Indicator
It should be in limit 0.025-0.33 mm

TIMKEN E-48999
Self align double row spherical roller bearings

22326 C3 CK
SELF ALIGNMENT UPTO
2.5 DEG.

Induction Heating 130 -140 DEG. CEN
Prud’homme proposed the limiting lateral force as:

$$Hy \leq 0.85 \ (1+P/3)$$

where $Hy$ is the lateral force & $P$ = axle load (t)

Control arm centre pivot bush for wheel - axle guidance

Existing

Axial –Lateral stiffness = 04 kN/mm
Longitudinal Stiffness = 40 kN/mm

Proposed stiffness for trails by RDSO to avoid shear

Axial –Lateral stiffness = 18 kN/mm
Longitudinal Stiffness = 08 kN/mm

SERVICE LOAD $C_x = 23 \text{ Kn\ aginst stiffness of } 40 \text{ kN}$
NOTE:
1) SERVICE LOAD (RADIAL) = 23kN
2) EXCEPTIONAL RADIAL LOAD = 38kN
3) TORSIONAL ANGLE = 6 DEG

4) MANUFACTURE S INITIAL WITH DATE YEAR TO BE PUNCHED HEREWIT I LETTER HEIGHT
5) APPLICABLE SPECIFICATIONS ARE TS 17.35E
6) FATIGUE TESTING SHOULD BE AS PER NDT5-122
7) TORSIONAL STIFFNESS IS FOR REFERENCE ONLY
Radial-steered bogies on their own allow an increase in operating speeds up to 180Km/h without increasing Rail/wheel forces compared with conventional bogies.

This reduces wear on both the rail and wheels - wheel life is increased by up to six times.
CONCLUSIONS

EXCESSIVE OSCILLATIONS DUE TO

- Slack Gauge
- Thin Flange
- Increased Play in bearing & Journal
- Excessive Lateral and Longitudinal Clearances

Increase Derailment Proneness
The small pieces of metal break out are grinded by tread cleaning device to avoid shelling and spalling.
Primary suspension
High stiffness- Low deflection springs nest

- 04 nos Nested coil springs with top rubber pad, primary vertical dampers, control arm, elastic joints - connecting the cartridge bearing on wheel set to bogie frame.

- Articulated Flexible guidance.

Vertical Stiffness outer = 475 N/mm
Vertical Stiffness inner = 280 N/mm
Combined Stiffness of nest = 755 N/mm

PRI. VERTICAL DAMPERS - 04 NOS.
4250 +/- 640 N @ 0.3 M/sec.

LATERAL STIFFNESS 5 TIMES THE VERTICAL STIFFNESS AND LONGITUDINAL STIFFNESS. 16 TIMES OF VERTICAL STIFFNESS.
Primary Bump stop

AXLE LOAD ON ONE JOURNAL
= 18/2 = 9 TONNE
DEFLECTION = 6.5mm only

NOTE:
1) FATIGUE TESTING SHOULD BE AS PER MDT S-122.
2) MANUFACTURER'S INITIALS, MONTH & YEAR TO BE EMBOSSED HERE#
Primary suspension---

- Spring Design
- Stress $= \frac{G \cdot d^4}{8 \cdot Dm^3 \cdot n}$ -- N/mm.

- $G$ = MODULUS OF RIGIDITY
- $D$ = Bar Dia.
- $Dm$ = Mean Dia.
- $N$ = No. of active turns

- RDSO SPEC WD-01-HLS-94 REV-3
- MATERIAL – IS:3195-92
  - $d<30$ - 60 - Si 7 ,
  - $30<d<60$ - 52Cr4Mo2v
Orientation of Outer & Inner Primary springs to avoid biting of coils and for balanced distribution of load on centering disc.
(With Secondary Spring System Exploded)

- Bogie Bolster
- Miner Pad
- Spring Guide
- Outer Spring
- Inner Spring
- Rubber Spring
SECTION A-A
SCALE 1:2

NOTE:-
2. SPECIAL REQUIREMENTS AS PER MOTS-148.
3. MANUFACTURER'S INITIALS, MONTH AND YEAR TO BE EMBOSSED HERE # IN LETTER HEIGHT OF MIN. 6mm.

DETAIL ORGS STARTING WITH "CL" ARE INTERNAL REFERENCE LISTS ONLY AND ARE NOT FOR ISSUE.
THIS IS A COMPUTER GENERATED DRAWING. ANY MANUAL ALTERATION SHALL AUTOMATICALLY RENDER IT INVALID.
FOR UNTOLERANCED DIMENSIONS REFER MD00096
DATE OF FIRST ISSUE 20/07/2005 CGM BY Rajan
**RESONANCE**

Natural Frequency = \[ \frac{1}{2\pi} \sqrt{\frac{k}{m}} \]

\( k = \) Spring Stiffness  
\( m = \) Mass

If frequency caused by external excitation is equal to natural frequency, resonance occurs. If there is no damping in the system, the amplitude becomes infinite with time.

**High stiffness >>>> High frequency >>>> Poor Ride**
Critical damping is the minimum amount of damping overshooting the equilibrium position when released from displaced position.
SECONDARY SUSPENSION
NEST OF FLEXI-COIL SPRINGS INNER AND OUTER, RUBBER SPRING WITH (MINER )PAD & PRI. VERTICAL, SEC. VERTICAL & LATERAL AND YAW DAMPERS AND ANTI ROLL BAR ETC.

- SEC. VERTICAL DAMPERS -02 NOS.
  3500 +/- 520 N @ 0.2 M/sec.
- LATERAL DAMPERS -01 NOS.
  8000 +/- 1200 N @ 0.3 M/sec.
- YAW DAMPERS -02 NOS.
  11000 +/- 1650 @ 0.1 M/sec.

- Combined Vt. Stiffness Sec. Spg. = 370.6 N/mm
- Combined Lat.. Stiffness Sec. Spg. = 195.6 N/mm
- Lateral flexibility provide better lateral ride.

RATIO PRI AND SEC STIFFNESS
755 N/mm : 370 N/mm
67 % : 33 %
Velocity = \pi \times f \times c

= \pi \times \text{rpm} \times \text{stroke} \times \frac{1}{60}

\text{Act. disp./Force delivered by damper} = \frac{u/F}{60}

= \frac{1}{k} - \frac{1}{wd}

k = \text{series stiffness}

w = 2\pi f

d = \text{Damping Coefficient}

By Fourier Analysis
Secondary Suspension

Helical coil suspension

Vt. Stiff. = 471 N/mm
d = 42 mm; Dm = 242 mm
N = 6.25; F.H. = 400 mm

Fiat Bogie

Conventional Bogie

Flexi-coil suspension

Vt. Stiff. = 241-O, 129-I N/mm (370 N/mm)
d = outer 34 - Inner 26
Dm = outer 246, Inner 138
n = outer 8.3, inner = 6.6
F.H. = outer 707, inner 663 mm
Each flexicoil spring is provided with the following markings:

- The positive directions of the alignment deviations is indicated with an Aluminium band (secured tightly and wound twice around the spring)
- The length of the spring under test load and the value of the alignment deviation (in mm) are printed on a nonferrous metal band.
Proposal to avoid Shifting of Traction Centre

- Perforated stainless steel disc to tacked with bolster and side frame for under pressure plug locking of miner rubber pad to eliminate shifting problem.

- Pre compressed minor pad with load of 20 t before fitment to be fitted with in 04 hors (TS-17477)
ALIGNMENT DEVIATION (COUPLING INSTRUCTIONS)

- The difference between the alignment deviations of the two outer springs not to exceed 4 mm and that of the inner springs 8 mm.

- The outer and inner springs with the greater alignment deviations must be situated in the same spring assembly, that is:

- If A greater than B, C should be greater than D
- A - B = 4 mm max, C - D = 8 mm max
SPRING TESTING AND DAMPER TESTING MACHINES SHOULD BE INSTALLED IN ALL WORKSHOP FOR POH SO THAT EFFECT OF PERMANENT SET MAY BE COUNTER
Rolling of carbody for quasistatic curving. Centrifugal force $mv^2/R$, gravitational force $mg$, roll angle $\varphi_c$ and cant angle $\varphi_t$.

(a) No cant leads to rolling towards the outer side of the curve.
(b) Full compensation of track plane acceleration, $a_y = 0$, gives no rolling at all.
(c) Cant is not sufficient for compensating the track plane acceleration. The carbody rolls towards the outer side of the curve, as in (a).
CENTRIFUGAL FORCE \( F \) = MASS \( M \) x ACCELERATION \( a \) 
\[ F = \left( \frac{W}{g} \right) \times \left( \frac{V^2}{R} \right) \]
WHERE 
\( V \) = SPEED METER/SEC.
\( R \) = RADIUS OF CURVE
\( W \) = WT. OF VEHICLE IN T.

TAN \( \phi \) = SUPER ELEVATION / GUAGE
\[ = \frac{e}{G} \]
\[ = \frac{CENTRIFUGAL \ \FORCE \ \/ \ \WEIGHT}{F / W} \]
HENCE \[ \frac{e}{G} = \frac{F}{W} \]
\[ e = \frac{G \times F}{W} ; \quad \frac{G \times W \times V^2}{W \times g \times R} = \frac{G \times V^2}{g \times R} \]

Super elevation \( e \) = \( \frac{G \times V^2}{127 \times R} \)

WHERE 
\( e \) = Super elevation in mm.
\( R \) = Radius of curve in Meters
\( V \) = velocity in KMPH
\( G \) = Dynamic Guage in mm (1750 mm in BG)

Degree of curve = 1750/ \( R \) in meter
175 M Curve = 10 degree
ANTI-ROLL BAR:

- Anti-roll bar used to control excessive rolling motion and to control roll frequency. Low roll freq. can lead to nausea associated with sea sickness.

- Tilting co-efficient as per UIC-515-1 & 4 should be less than 0.4 at high speed on the sharpest curve with max. permitted cant deficiency for keeping the vehicle within dynamic moving gauge and for passenger comfort.

- UIC-515-4, Wind pr. 600 n/m^2, Lateral force=43.2 kN, Tilting Momentum=108 kN
BOGIE FRAME

- Y-DIP SIDE FRAMES OF MATERIAL S355J2W+N EN10025 Part-5 in place of ST5 2.3
- TWO SIDE FRAMES CONNECTED BY TWO BRAKE BEAM ASSEMBLY
  (CROSS TUBES- DIN.1630-ST52.4 OD=168.3 THK=14.2 MM)
- WHICH SUPPORTS:
  - CONTROL ARM BRACKETS
  - SUPPORT BRAKE SUPPORT,
  - PRIMARY SPRING POTS
  - ANCHOR LINK BRACKETS
  - CROSS SECTION FRAME FOR LATERAL
  - AND LOGITUDINAL BUMP STOPS ETC.
- Surface protection Garnet Ballast Sa 2.5DIN 8501
- Adhesion promoting Etch primer if ballsating not possible.
- Epoxy zinc phosphate primer RDSO spec M&C/PCN/100/2013
- Visco elastic aqueous synthetic resin Anti Stone Chipping Paint RCF MDTS 22283. for corrosion preention.
### FIAT Bogie Materials

#### Description | Thk | Material
---|---|---
Top plate | 10 | S355J2W+N
Bottom plate | 10 |
Web | 10 |

#### Description | Thk | Material
---|---|---
Top plate | 12 | S355J2W+N
Bottom plate | 12 |
Web | 12 |

### Brake Support
- Steel cast (GS20Mn5V)

### Pin Bracket
- Steel cast (GS20Mn5V)

### Brake Beam
- Seam Less OD168, Thk. 14mm (St52)

### Brake Support
- Steel cast (GS20Mn5V)

### IRS R-19 PT-II

### Axle
- (R7T, UIC811)

### Brake Disc
- Grey Cast Iron

### Control Arm Support (Forged)
- Mat. S355J2W

### Outer/Inner Spring
- Mat. 51CrMoV4/52SiCrNi5

### Control Arm (SGCI)
- Mat. SG400/18

### Front Cover (SGCI)
- Mat. SG400/15
Cross Section with Lateral and longitudinal stoppers.

Lateral gap = 25 +/- 5mm ;
Longitudinal Gap = 8 +5/-2mm

1. Lateral bump stop
2. Longitudinal bump stop
3. Support-frame

(Without Bolster)
NOTE:
1. FATIGUE TESTING SHOULD BE AS PER MIL-STD-122.
2. MANUFACTURER'S INITIALS WITH MONTH & YEAR TO BE PUNCHED HERE # IN LETTER
HEIGHT OF 6mm & SAME SHOULD BE EMBOSSED ON RUBBER PART AT SUITABLE PLACE.
Traction center with traction levers

- Traction Lever
- Traction Center
- Curve Roll
- Anti Roll Assly
Traction and braking forces:

- BODY-BOGIE BOLSTER CENTER POST - TRACTION CENTRE-TRACTION LEVER/LONGITUDINAL BUMP STOP-BOGIE FRAME-CONTROL ARM-AXLES.
CURVE ROLL ON COACH END SIDE TO RESTRICT EXCESS ROTATION OF BOGIE WITH RESPECT TO COACH

STOPPER BRACKET NEAR FOOT STEP ARRGT. ON UNDER FRAME FOR RESTRICTED MOVEMENT OF CURVE ROLL
Important clearances in dynamic situation

With shim

Without shim

Vertical stopper gap

Gap between bolster and mounting frame
Bogie Body Connection

- 14mm shims are essentially required in new coach, in tare condition, to maintain CBC height and sole bar bottom level from level.

- As per LHB document maximum permissible limit for shims, for buffer height adjustment, is 64 mm (including mandatory shims of 14mm).
Bogie Body Connection

✔ Provides rigid connection between body and bogie
✔ Capable to transmit 0.25g acceleration in lateral and longitudinal in normal operation and 5g in emergency condition
# Force Transmission path

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<tr>
<th>Vertical Forces</th>
<th>Lateral Forces</th>
<th>Traction and braking forces</th>
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<tr>
<td>- Miner Pad</td>
<td>- Miner pad</td>
<td>- Traction center</td>
</tr>
<tr>
<td>- Sec. Suspension</td>
<td>- Sec. Springs</td>
<td>- Traction lever</td>
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<td>- Bogie Frame</td>
<td>- Lateral Bump Stop</td>
<td>- Longitudinal bump stop</td>
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<tr>
<td>- Primary Springs</td>
<td>- Bogie frame</td>
<td>- Bogie frame</td>
</tr>
<tr>
<td>- Ball joint control arm</td>
<td>- Ball joint control arm</td>
<td>- Control arm</td>
</tr>
<tr>
<td>- Axle</td>
<td>- Axles</td>
<td>- Axle</td>
</tr>
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![Force Transmission path diagram](image)
# Force Transmission Route

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<td>Axle</td>
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</tbody>
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![Diagram of force transmission route]
Brake disc- Converts kinetic energy into heat energy

SPECIFICATION

Dimension -
640 mm x 110 mm; Brake Radius 247 mm
Type -
Axle shaft mounted, concentrically split,
Material -
Grey cast iron Friction ring
Wear (allowed) -upto 96 mm width.
Weight - 126 kg

• ORGANIC PADS: Typically contain Asbestos free
Synthetic rubber with nonferrous metals, organic
mineral fibers, abrasives, lubricants and property modifiers such as glass, rubber, kevlar and carbon
Brake Disc

- Internally vented (Apprx. Weight 145 Kg)
- Good heat dissipation and high braking performance
- No maintenance required
- Wear checks/limits:
  - No hair, incipient crack or through cracks permitted on Hub/connected flange:
  - Incipient cracks on Disc:
    - <80mm, apart by 50mm min. permitted propagating between two edges
    - <50mm, apart by 50mm min permitted propagating from the edges
  - Scoring up to 1mm allowed
  - Concave wear up to 2mm allowed
  - Residual thickness up to 14mm of disc friction surface
  - Slanting wear up to 2mm allowed
  - Permitted for breakage of 4 fins max. alternatively
Improvements in brake disc from MARK-I ----MARK-III

Reinforced cross section of lug by +70%

Reinforced Outer Cooling Fins by +32%

Sliding Blocks and additional „Shock Support“
Brake caliper & Brake pad

- Suitable for UIC type 200 x 2 brake pads, thickness 35mm
- Caliper ratio -2.17 (2.48 for special coaches-power car and DD)
- Brake radius - 247 mm
- Weight - 67 kg (with brake pad)
- Wear limit - 28mm max.

Approved 35 mm Non asbestos pads for LHB

JURID 877 of M/s Federl Mogul (Honeywell) Germany
BECORIT 984 of M/s Becorit, Germany
BK 7000 of M/s Bremskerl, Germany

Friction co-eff.-0.35; Conforms to requirements in UIC 541-3 OR RDSO specification: CG/2013/CG-01
DISC BRAKE SYSTEM FOR 200 KMPH.

- Electro Pneumatic assist – Control panel with electro Magnet valve advantage in achieving uniform braking & EBD
- Steel discs (instead of grey cast iron) per axle to keep the temperature of the brake discs and the pads under control
- Flexible sintered pads - provides iso-pressure even in case of wear of pads and brake energy upto 40 MJ/disc
- Life cost cycle of CS Disc + sintered 185% in comparison to than GCI disc + organic pads

Emergency Braking Distance

<table>
<thead>
<tr>
<th>Initial speed</th>
<th>Pneumatic</th>
<th>With EP</th>
</tr>
</thead>
<tbody>
<tr>
<td>160 km/h</td>
<td>1173 m</td>
<td>1017 m</td>
</tr>
<tr>
<td>180 km/h</td>
<td>1451 m</td>
<td>1275 m</td>
</tr>
<tr>
<td>190 km/h</td>
<td>1600 m</td>
<td>1415 m</td>
</tr>
<tr>
<td>200 km/h</td>
<td>1757 m</td>
<td>1562 m</td>
</tr>
</tbody>
</table>
AMDBS – Schematic Layout

- Speed Sensor
- Phonic Wheel
- WSP-Electronic
- U_{\text{Batt.}}
- Vehicle-Bus
- Dump Valve
- Electric
- Pneumatic
- Brake Pad
- Brake Disc
DETAILED VIEW OF EARTHING ARRGT.

Shell body to bogie frame earthing cable

Bogie frame to axle earthing cable

Resistor earthing cable

Copper bush earthing arrangement on axle end.
DETAILED VIEW OF EARTHING ARRGT.

Shell body to bogie frame earthing cable

Bogie frame to axle earthing cable

Resistor earthing cable

Copper bush earthing arrangement on axle end.
Suspension of LHB GS Coaches

LS1- With 100 seater, chair car spring, SBC underslung

LS2- Chair car springs with 32 mm shim, Suitable for 16 T pay load
Under-slung water tanks 2x685 ltrs removed
Transverse luggage rack shifted upward

LS-3 variant LHB GS/EOG coach
Shalimar springs with 32 mm shim
Suitable for 18 T pay load

LS-4 variant LHB GS/EOG coach
Shalimar springs with stiffer secondary inner springs
with 32 mm shim Suitable for 22.6 T pay load

LS-5 WITH AIR SPRING IN SECONDARY 140 KN CAPCITY.
Secondary Air Suspension

- Maintain constant height at varying load
  - Fewer variants required to be stocked for various coaches
  - Buffer height adjustments easy
  - Helps in maintaining level of coach under non-uniform loading
- Less failures as compared to helical springs

- Air spring as per rdso Spec cK-509 for Conv. and cK-508 for FIAT bogies
Advantage of Air Spring over Helical Springs.

- Constant Buffer Height at varying pay loads.
The spring consists of an air bellow fitted between two plates:

- Air pressure creates an air gap between the plates which provides cushioning.

There is an inner emergency rubber spring also:

- Comes in operation upon deflation of bellow or during overload.

**Table 1: Part list for ck509 airspring**

| Part | Parts name                      | Part number      | Weight (Kg) | Quantity per unit | per
<table>
<thead>
<tr>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>O-ring</td>
<td>GB3452.1 7.5 × 5.3</td>
<td>0.1</td>
<td>2</td>
<td></td>
</tr>
<tr>
<td>2</td>
<td>Air inlet</td>
<td>C.KH0604000301</td>
<td>5</td>
<td>1</td>
<td></td>
</tr>
<tr>
<td>3</td>
<td>Hexagon socket head cap screws</td>
<td>GB/T 5788-2000 M8 × 16</td>
<td>0.2</td>
<td>4</td>
<td></td>
</tr>
<tr>
<td>4</td>
<td>O-ring</td>
<td>GB3452.1 87.5 × 3.55</td>
<td>0.05</td>
<td>1</td>
<td></td>
</tr>
<tr>
<td>5</td>
<td>Top plate</td>
<td>C.KH0604000300</td>
<td>42.5</td>
<td>1</td>
<td></td>
</tr>
<tr>
<td>6</td>
<td>Sliding plate</td>
<td>C.KH0604000500</td>
<td>1.25</td>
<td>1</td>
<td></td>
</tr>
<tr>
<td>7</td>
<td>Hexagon socket head cap screws</td>
<td>GB/T 70.3 M8 × 20</td>
<td>0.3</td>
<td>6</td>
<td></td>
</tr>
<tr>
<td>8</td>
<td>Emergency spring</td>
<td>C.KH060400200</td>
<td>90.5</td>
<td>1</td>
<td></td>
</tr>
<tr>
<td>9</td>
<td>Air spring bellow</td>
<td>C.KH060400100</td>
<td>11.25</td>
<td>1</td>
<td></td>
</tr>
</tbody>
</table>
FIAT-SG with 30 kW permanent magnet alternator
FIAT BOGIE WITH AIR SPRING AND MULTI DISC BRAKES FOR 250 KMPH
Wheel spalling: - spalling occurs as a result of fine thermal cracks joining to produce the loss of small piece of tread material.

Wheel shelling: - shelling is due to result of stress generated by rolling contact fatigue and leading to material flow and damage at wheel surface.

Rolling contact stresses are major factor controlling both shelling and spalling.

Fatigue process: dependent on magnitude and range of multi axial alternating stress component at or near the tread surface. The presence of compressive normal stress on plane having greatest range of shear stress would tend to inhibit crack nucleation and propagation. ----- require information on the full thermodynamic cycle of complex multiaxial stress experienced by tread surface and sub surface elements during both wheel rotation and major breaking cycles.

Stress in rolling contact: Elastic contact pressure between wheel and rail has magnitude proportional the cube root of the wheel loads. The associated half width of contact patch would be about 6.5mm. Orthogonal sub surface shear stress cycle changes with addition of traction as would occur in a braked wheel. Retarding force and traction at the contact increase, the max. Range of shear stress move towards the surface, and tensile stresses begin to develop at the trailing edge of the wheel contact until friction ratio exceed at least 0.25.

Impact effect: it can effect both crack initiation and crack propagation modes. It can arise from rail joints. Wheel flats serves as sites for formation of additional flats and resulting shelling. Dipped joints or corrugation impose repeated defects on wheel as it traverses and contact pressure can be increased by factor of 3 or 4.

Martensite formation: Thermally induced metallurgical transformation of region in the tread surface can contribute to cracking. Eventually spalls are formed by un tempered martensitic, which is very brittle -- cracks