PAPER No. 377

A Study of the Effect of Suspension Parameters of Ride Index of a Railway Vehicle and Results of Trials on the Pakistan Western Railway

By

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1. Introduction

1.1. Railway being a transportation industry, the basic preoccupation of railway management has always been to offer its customers the finest in transportation. The demand for providing better travelling facilities has been aggravated on account of competition offered by road transportation especially in lower class passenger traffic. Great volume of passenger traffic is conveyed by road vehicles, often overcrowded and in poor mechanical condition, at fares with which railway cannot compete. Air transportation shares major portion of upper class passenger traffic. Faced with these two major competitors, the railway administration was compelled to consider some changes, with travel comfort as a vital factor, to make railway journey attractive at a competitive price. Among many factors which make railway journey comfortable, riding characteristics of a railway vehicle is by and large the most important one. It, therefore, became imperative to re-examine the design of coaching stock bogies in the Pakistan Western Railway so as to get at a suspension system which would have good riding characteristics on our track.

1.2. A detailed study of this question was undertaken in collaboration with the German Firm of carriage manufacturers, Messrs. Linke-Hofmann-Busch and this paper mainly describes the trials conducted and the results obtained therefrom.

2. Riding Comfort

2.1. As the objective of the trials is to improve riding comfort, it will first be necessary to get a clear conception of what this term means and the various design factors on which it depends. Comfort may be regarded as the sum total of all those measures which tend to maintain and improve the well

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being of a passenger and to reduce his fatigue during a journey. It is mainly dependent on the following factors :—

(*a*) Car body vibrations,
(*b*) running noises,
(*c*) dust nuisance,
(*d*) temperature control,
(*e*) lighting, and
(*f*) general aesthetics.

2.2. The first factor is the most important of all and the scope of this article is confined to the study of riding comfort with respect to car body vibration and the evaluation of design of a bogie in order to minimize travel fatigue due to vibrations.

2.3. The running quality of a vehicle depends on its 'transmissibility ratio' *i.e.* on the capacity of the running gear (the bogies) to transfer shocks and impacts acting in the vertical and horizontal directions into bearable type of vibrations. Transmissibility ratio is a function of the ratio of the frequency of the exciting forces and the natural frequency of the suspension gear of the car body.

2.4. The exciting forces that give rise to vertical and lateral oscillations are induced by :—

(*i*) the rail joints,
(*ii*) wheel eccentricity,
(*iii*) rail surface irregularities,
(*iv*) the snaking action of the wheel pair,
(*v*) track gauge variations,
(*vi*) bad alignment of track, etc.,

and its frequency is governed by the speed of the vehicle.

2.5. The natural frequency of the suspension gear of the car body is dependent upon the following factors :—

(*i*) specific deflection of the springs,
(*ii*) vehicle mass,
(*iii*) inertia of the system, and
(*iv*) the link arrangement.

3. Modes of Car body movements

3.1. The fundamental modes of car body movements consist of oscillations along the three principal axes *viz.*, bouncing along the Z-axis, fore and aft movements along the longitudinal axis (x-axis) and lateral oscillation along lateral axis (y-axis). The rotational movement about the principal axes are known as nosing, rolling and pitching respectively. Because of the complexity of vehicle dynamics these oscillations are generally coupled up, and

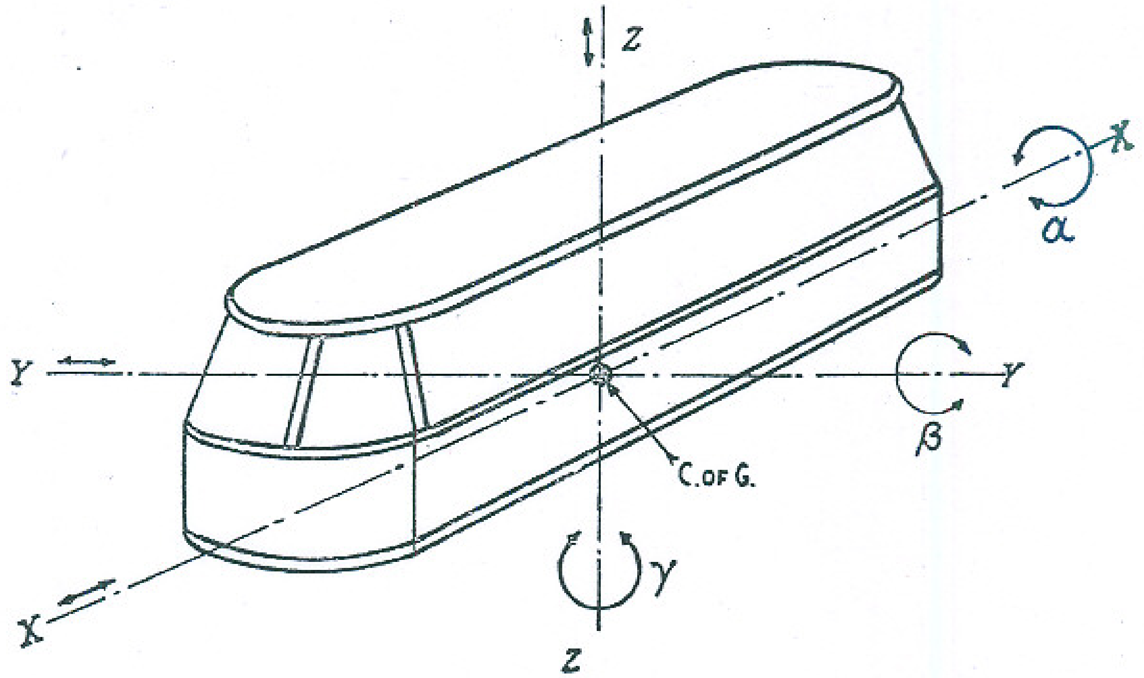

FIG: 1

combinations of various movements shown in the following figures are classified as under :—

Principal Oscillations

(*i*) Z—Z=Bouncing	=Nosing γ	
(*ii*) Y—Y=Lateral	=Pitching β	
(*iii*) X—X=Shuttle	=Rolling α	

Combinations

(*i*) Lateral	(Y) +Nosing (γ)=$\gamma.\gamma$	Hunting	
(*ii*) Lateral	(Y) +Rolling (α)=$\gamma.\alpha$	Swaying	
(*iii*) Bouncing	(Z) +Rolling (α)=$z.\alpha$	Shunning	
(*iv*) Bouncing	(Z) +Pitching (β)=$z.\beta$	Galloping	
(*v*) Shuttle	(X) +Pitching (β)=$x.\beta$	Rocking	
(*vi*) Shuttle	(X) +Nosing (γ)=$x.\gamma$	Jerking.	

3.2. From the above it will be seen that consideration of all such oscillations at the time of designing of the vehicle involve solution of numerous simultaneous equations and is a cumbersome procedure and cannot be done without the aid of digital computer. Generally, all that is being done at present with paper, pencil, and slide rule is to determine the natural frequencies of bouncing, nosing, lateral and rolling oscillations and that of swaying. After determining these natural frequencies, the spring characateristics and swing link proportions are so chosen that resonance does not occur at operational speeds and that the vehicle is not unduly sensitive to vertical and lateral impacts. Knowledge of natural frequencies also helps in determining the damper characteristics for reducing amplitude of oscillations within acceptable limits at resonance, without adversely affecting the riding quality of the vehicle at speeds above that of the resonance speed.

4. Oscillations in the Lateral Plane

4.1. Introduction

Since this particular movement of the railway vehicle has a predominant effect on human beings, it has been a subject of a study by research workers on running of railway vehicles, for quite some time in the past. Specialised investigations of such oscillations date back to Max Maria Von Weber in the year 1854. All such efforts and theoretical investigations are reported by Heuman in his work "Grundzuge der Fuhrung der Scheienenfahrnzeuge" published in 1954. Valuable and pioneering work has also been done by Carter in this field. Research on this subject is however still continuing and the latest theory of "dynamic instability" is under examination at the British Railway Research Centre at Derby. This theory, if sustained by exeprimental results, would bring a radical change in the very concept of lateral oscillations of railway vehicles. We would, however, confine ourselves in this paper to the current theory on bogie design.

4.2. Calculation of lateral frequencies

4.2.1. The geometry of bolster suspension of a bogie is shown in the following figure :—

4.2.2. The natural frequency of vertical link freely suspended from a bracket is analogous to that of a pendulum and is given by :—

$$f=\frac{1}{2\pi}\sqrt{\frac{g}{l}}=\frac{1}{6{\cdot}28}\sqrt{\frac{386}{l}}$$

$$=3{\cdot}13\ \sqrt{{}^{1}/l_{1}}\ \text{cycles/sec.} \qquad \ldots.(1)$$

4.2.3. This relation also applies to the vertical swing link of a bogie, and it will be noted that the only parameter affecting natural frequency is the length. The picture, however, changes when it comes to inclined l'nks ; but the natural

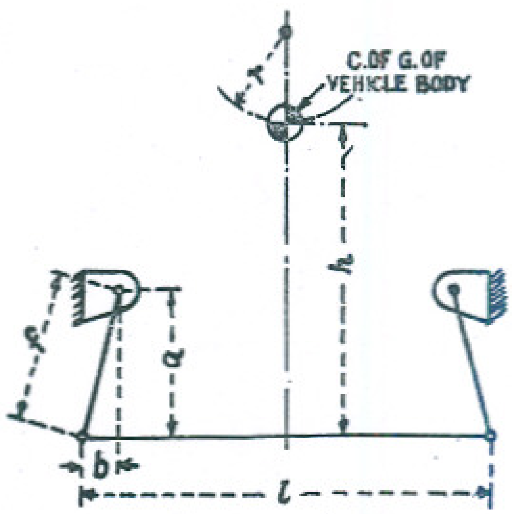

FIG: 2

frequency of such a system could also be determined from the above relationship by substituting r for l, where r is determined by :—

$$r = \frac{a(al-2bh)^2}{l^2f^2-2bl(f^2+a^2)-4ab^2h} \text{ (in)} \qquad \ldots.(2)$$

4.2.4. It will be noted that the value of r depends entirely on the geometry of swing link arrangement, as well as upon the height 'h' of the centre of gravity above the lower link-ends. This means that with a load-carrying vehicle, the natural frequency will be governed by the position of the centre of gravity which will vary with the loads.

4.2.5. After having calculated the value of r, the natural frequency of lateral oscillations or a coach, having one particular suspension arrangement, can be worked out for different loadings.

4.2.6. Long vertical swing link is now the most common choice of designers of bogies because of the following imperial advantages. Its use results in low natural frequencies of body oscillations in horizontal plane and provides elastic lateral coupling between the bogie and the body. Above all, the most important advantage is that speeds at which resonance is likely to occur on account of oscillations set up by the sinusoidal motion of the wheels on track, coinciding with the natural frequency of swaying of the car body, will be restricted to low ranges only.

4.3. Nosing frequency

Apart from the above-mentioned oscillations, an even more important one is the natural frequency of the car body on swing links, about the vertical axis, which is known as the nosing frequency. It is therefore necessary to determine its magnitudes. Formula for this natural frequency is

$$f_n = \frac{a}{2\pi} \sqrt{\frac{C_l}{I_z}} \text{ cycles per sec} \qquad \ldots.(3)$$

where a (ft.) is half the distance between the bogie centres, C_l (lbs per ft.) is the centring force, I_Z =(lb. ft./sec.) the polar moment of inertia about Z axis. While calculating the amount of inertia, due consideration is given to the load to be carried by the coach in service and the moment of inertia is modified accordingly. Nosing frequency can be calculated by substituting the values in the above quoted equation.

4.4. **Combined Oscillations (Swaying)**

4.4.1. In addition to the lateral and rotational (about Z axis) oscillations of the vehicle body considered so far, particular attention has to be paid to the combined oscillations. The most important combined oscillation is the one resulting from the action of lateral oscillations of swing link and the body rolling on bolster springs. These oscillations are made up of lateral displacement through a distance y, and roll through an angle β about a point often located at the level of the top of bolster springs. The lateral displacement y and the roll β can be combined and represented by a swaying movement with the centre of rotation usually located just above or just below the rail level. Lower positions of rotational centre signify low lateral frequency due to long swing links.

4.4.2. Since the system has two degrees of freedom *i.e.* lateral displacement and roll, there will be two natural frequencies ; a lower one with the body swaying about a point near rail level, and a higher one with the body swaying about a point above the centre of gravity of the body. The theoretical aspects of these modes of body oscillations have been investigated by Borgeaud, who has derived the following quadratic equation for obtaining the natural frequency of swaying of vertical links and of inclined links based on its effective length and other similar centring devices.

$$i^2 w_0^4 - h_i^2 (w_2^2 + w_4^2) w_0^2 + h_i^2 w_2^2 w_4^2 = 0 \qquad \ldots.(4)$$

Solving this equation we get

$$w_0 = \frac{h_i^2 (w_2^2 + w_4^2) \pm \sqrt{[h_i^2 (w_2^2 + w_4^2)]^2 - 4i^2 h_i^2 w_2^2 w_4^2}}{2i^2} \qquad \ldots.(5)$$

This expression provides two values of sway frequencies of the body w_{oh} and

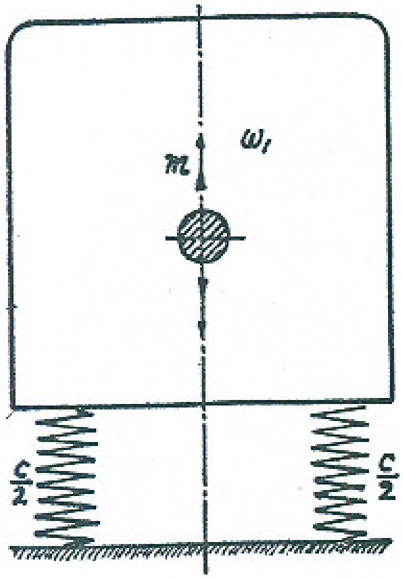

FIG: 3

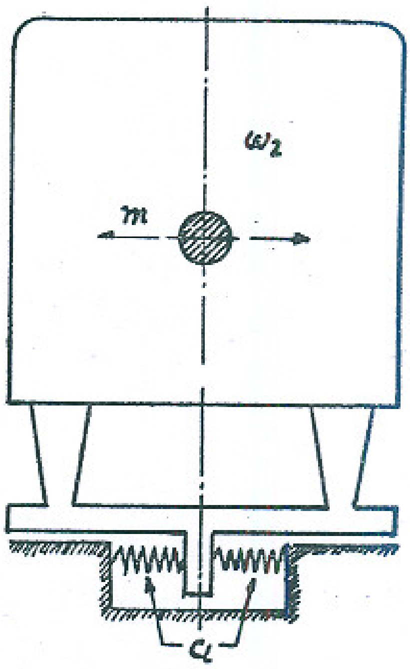

FIG: 4

w_{ol} the higher about a point above the floor level, and the lower about a point near the rail level. The magnitude of w_1 w_2 w_3 w_4 *i.e.* the magnitude of natural frequency of bouncing, lateral and roll oscillations can be readily worked out as under

(*i*) $w_1^2 = \frac{C}{m}$

(*ii*) $w_2^2 = \frac{C_1}{m}$

(*iii*) For determining w_3 it is necessary to know the distance h_s. between the point of rotation and centre of gravity of the body which is about 5.8 ft for passenger coaches. Therefore $w_3^2 = g/h_s$.

(*iv*) For determining w_4 it is necessary that we know the value of i and the value of h_i^2. i is the radius of gyration of the body about the longitudinal axis x through the centre of gravity of the coach. Its approximate value for coaches has been calculated to be i=3.75 ft. The value of w_4^2 is then given by

$$w_4^2 = \frac{i^2 w_1^2 - h_s^2 w_3^2}{h_i^2} \quad \ldots.(9)$$

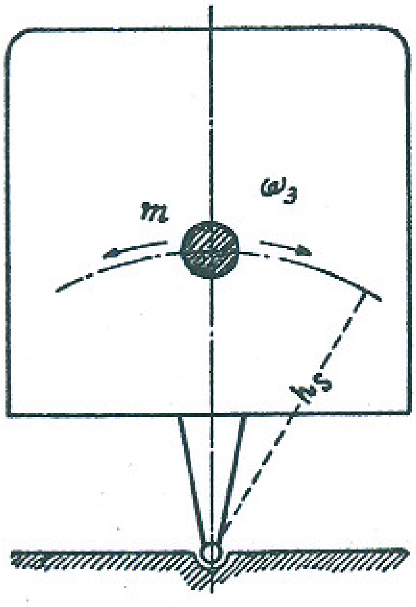

FIG: 5

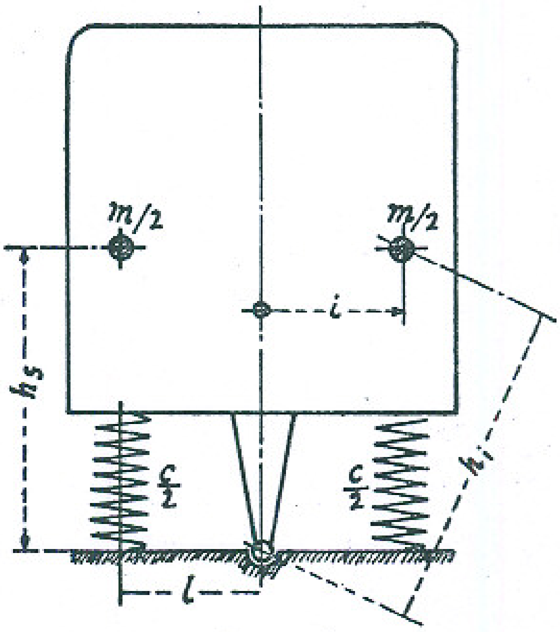

FIG: 6

4.4.3. Once we have values of these frequencies, the swaying frequencies are calculated by substituting the values in the expression given above.

4.4 4. In the foregoing paragraphs, expressions have been derived to determine the natural frequency of oscillations in the lateral plane of the suspension gear, *viz*., that of lateral, nosing and swaying. Now we have to ensure that resonance between the above discussed oscillations and the frequency of the exciting force, is as for removed from the operational speeds as possible. Of all the external causes exciting oscillations in the lateral plane, only the frequency of vibrations set up by the sinusoidal motion of wheels secured to a common axle, owing to the conicity of wheel tyre, can be ascertained theoretically.

4.5. Sinusoidal motion of wheels secured to common axle

4.5.1. The frequency of theoretical sinusoidal motion of the wheels secured to a common axle is given by the expression

$$f_s = \frac{V}{2\pi} \sqrt{\frac{1/n}{Rs}} \text{ cycles per sec} \qquad \ldots.(10)$$

in which v (ft./sec.) is the vehicle speed, R (ft.) is the radius of wheel, n is the tyre conicity and $2s$ is the distance between the rolling circles of tyres. For broad gauge (5 ft. 6 inches) $2s$ is taken as 68 inches.

4.5.2. The frequency for a bogie with two axles exactly parallel to each other and at right angles to the track is calculated by the expression :—

$$f_{s2} = f_s \sqrt{1 + \left(\frac{l}{2s}\right)^2} \qquad \ldots.(11)$$

where l is the wheel base.

It will be apparent from the above equation that frequency of the sinusoidal motion will be directly proportional to the speed, and should have a constant wave length. Such, however, is not the case due to the fact that there are many factors influencing this motion, such as friction between the wheel and rail, the action of body mass, interaction of axle and bogie and the lateral stiffness and clearance of primary suspension components. Hence, in actual practice, it has been ascertained that with increase in speed, frequency does not proportionately rise, and the wave length does increase.

4.5.3. The excitation frequency thus calculated is compared with the natural frequency of lateral oscillations for choosing the design parameters of bogie suspension in order to restrict resonance to lower speeds as already mentioned. This consideration in design is important so as to eliminate any possibility of resonance due to swaying being set up at operational speeds.

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4.5.4. Before closing the discussion on lateral oscillations, a brief reference is necessary to the magnitude of these oscillations generally encountered in service, and their influence on the deflection characteristics of primary and secondary springs. It has been established from tests that a passenger can comfortably stand a steadily applied lateral acceleration up to about .07 g. Coaches, however, occasionally experience an acceleration up to ig. or even higher peak values in service, and hence, the designer has to cater for an unbalanced force of ig applied at the centre of gravity of the vehicle body so as to limit the angular displacement at cant-rail height to about .02 radians.

4.5.5. To ensure good riding, the bolster spring static deflection should be higher than that of the primary spring. Therefore, the bolster springs are always softer than the axle springs. The roll stiffness of the primary and secondary springs of a bogie is given by :—

$$C_a = \frac{c_1 l_1^2}{4}$$

$$C_b = \frac{c_2 l_2^2}{4} \text{ respectively.}$$

Their combined effect is given by:—

$$C_t = \frac{1}{4} \times \frac{c_1 l_1^2 . c_2 l_2^2}{c_1 l_1^2 + c_2 l_2^2} \; t \text{ ft/radian} \quad ..(12)$$

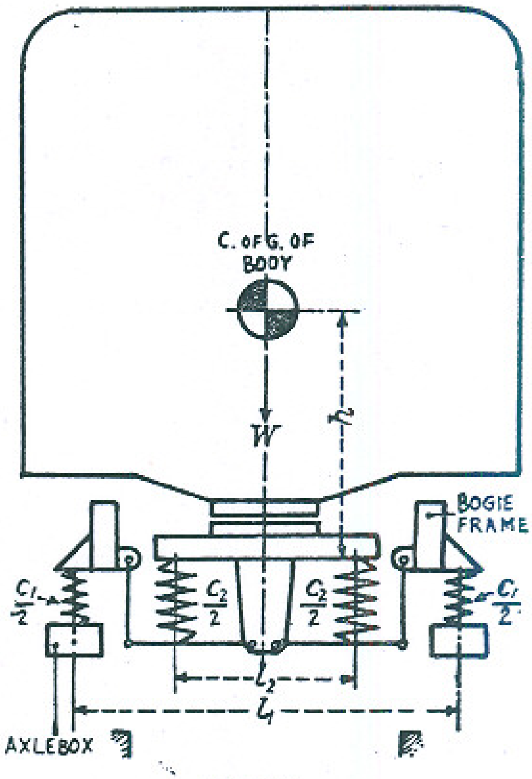

FIG: 7

where C_1 and C_2 are the specific deflections of primary and secondary springs in t/ft. of one bogie respectively, and L_1 and L_2 the lateral distance between the centre lines of the spring assembly or the spring base. The angle of roll is given by :—

$$\beta = \frac{Ph}{2c - wh} \quad ....(13)$$

where p (ton) is the total unbalanced force applied at the centre of gravity of the body, h(ft.) the distance from the axis of rotation to the centre of gravity of vertical body, $C = C_b$ or C_t, and W (ton) the body weight. It can be seen from the foregoing that the most important factor in the roll stiffness of a suspension arrangement is the magnitude of l_2 *i.e.* the lateral distance between the bolster spring centres. Designers are often tempted to keep this distance low and consequently stiffen the suspension arrangement to counteract undue roll at the expense of riding quality in the vertical plane.

5. Dampers

5.1. Before discussing oscillations in the vertical plane of a railway vehicle, a brief reference to Dampers is necessary. The term 'shock absorbers', commonly used for dampers in a vehicle suspension system, is a misnomer. Shocks are invariably absorbed by the suspension springs, while, as the hydraulic or friction dampers, dampen oscillations and reduce amplitude peaks. Damper system used in the suspension gear of railway vehicles have two different charactetistics :—

(*a*) Friction damping.

(*b*) Viscous damping.

Friction damping is inherent in all systems of suspension and is the result of frictional forces acting in opposition to the motion. Damping force K being constant in this case, peaks of decaying oscillations can be connected by a straight line :—

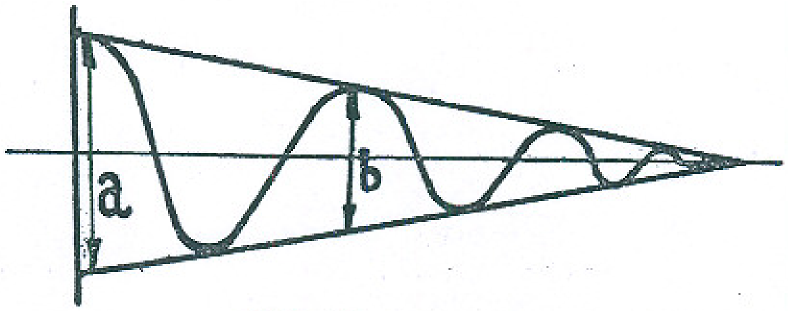

FIG: 8

The amplitude gets reduced by $4K_1$ at the end of each cycle. Thus, $b=a-4K_1$ in which $K_1=K/C$ where C (lbs. per in.) is the spring stiffness.

5.2. In the case of true theoretical viscous damping, the damping force opposing the oscillation is proportional to the velocity of the latter. The resulting force is represented by a proportionality factor p referred as the coefficient of viscous damping, or damping resistance. Its dimension being force (lb.) divided by velocity (ft./sec.), p is in lbs. sec./ft. Degree of damping is expressed in terms of the relative damping factor *i.e.* $D=\frac{p}{P_{cr}}=\frac{p}{2\sqrt{mc}}=\frac{p}{2mw}$ where P_c is the critical damping, *i.e.* with this damping factor the motion will cease to be a vibration. When disturbed from equilibrium the system would creep back to its position. In the above expression M (lb. sec.^{2}/ft.) is the mass of the oscillating system C (lb. per ft.) is the spring stiffness and W (radians per sec.) is the natural angular frequency. D is therefore non-dimensional in magnitude.

The critical damping is

$$D_{cr}=2\sqrt{mc}=2mw \qquad \ldots.(14)$$

and is taken as unity.

5.3. **Ratio of amplitude**

The ratio of amplitude X_m of one oscillation to the amplitude of X_m+1 of the following oscillation is given by

$$\frac{X_m}{X_m+1}=e\delta \qquad \ldots.(15)$$

where $e=2.7183$ is the base of natural logarithm, while

$$\delta=\frac{2\pi D}{\sqrt{1-D^2}} \qquad \ldots.(16)$$

is the logarithmic decrement of the oscillation decay. If δ is known the damping factor is given by :

$$D=\frac{\delta}{\sqrt{4\pi^2+\delta^2}} \qquad \ldots.(17)$$

The effect of D on oscillation pattern is shown in the following figure.

5.4. Generally, a damping factor of 0.2 to 0.25 is provided for achieving good riding quality in the vertical plane ; for the body nosing it can be increased to 0.3 to 0.4. Higher damping factors result in jerky action and induce high frequency oscillations of the body structure.

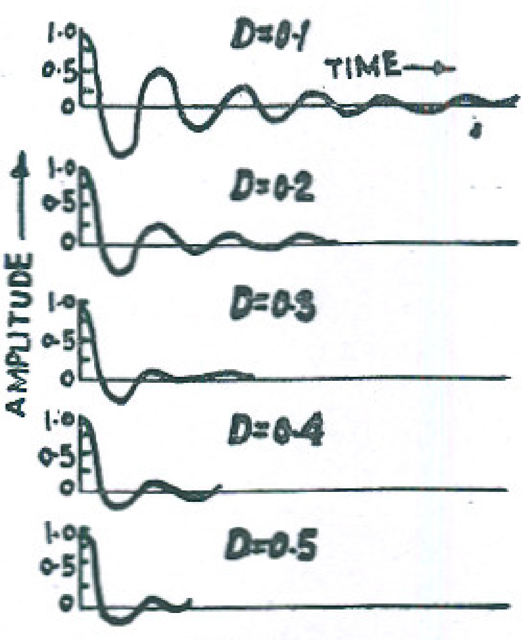

FIG: 9

6. **Vertical Oscillations**

6.1. Consideration of vertical oscillations of a railway vehicle is of a considerably simpler nature than its oscillations in the lateral plane. This is because of the fact that these oscillations originate from track irregularities, the prominent of which are the rail joints and its frequency can be easily calculated. The resonance speed of a vehicle is given by the formula :—

$$V = \frac{2fl}{1 \cdot 467} \qquad \ldots .(18)$$

where f is the natural vertical frequency of the bogie and L is the rail length.

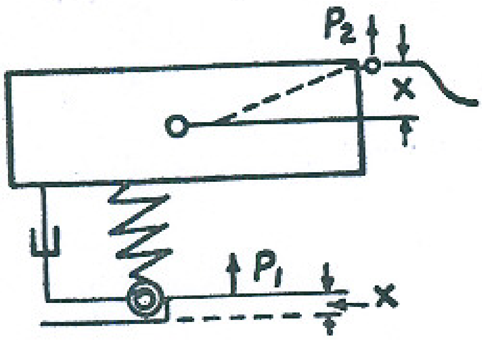

FIG: 10

6.2. A system consisting of a spring supported mass and a hdyraulic damper as shown in the following figure is known as a system with one degree of freedom:—

6.3. The relative transmissibility of such a system can be approximately evaluated with the aid of the expression :—

$$\text{R. T} \sqrt{\frac{1+4\,D_t^2\,r^2}{(1-r^2)^2+D_l^2\,r^2}} \qquad \ldots .(19)$$

where r is the frequency ratio.

The acceleration, in terms of absolute transmissibility, is obtained by multiplying the above expression by the square of the frequency ratio *i.e.*

$$\text{A. T.} = r^2 \sqrt{\frac{1+4\,D_l^2\,r^2}{(1-r^2)^2+4\,D_t^2\,r^2}} \qquad \ldots .(20)$$

6.4. From these equations it can be seen that in a system in which the frequency ratio exceeds 1.41 the dampers act like an amplifier. As a railway vehicle has to be operated on a wide range of frequency ratios, a compromise must be achieved by adequately choosing the damping factor so that when the ratio is less than 1.41 the good effect of damping is made use of, and in case it exceeds 1.41 the accelerations are not amplified to an uncomfortable range. This characteristic, as previously pointed out, is obtained when the damping factor is 0.2 to 0.25 for vertical oscillations.

6.5. The bogie, however, is still a complicated system. It consists of a primary spring system with total stiffness C_1, supporting the bogie mass m_1 and incorporating a damper with a damping resistance of p_1, which carries in turn bolster springs with a total stiffness of c_2 and a damping resistance p_1, and supporting the vehicle body with total mass m_2. This system has two degree freedom and the pattern of oscillations are more complex.

6.6. The system shown in the following figure will therefore have two natural frequencies of bouncing *i.e.* w_1 and w_2 as well as two natural frequencies of pitching *i.e.* w_3 and w_4.

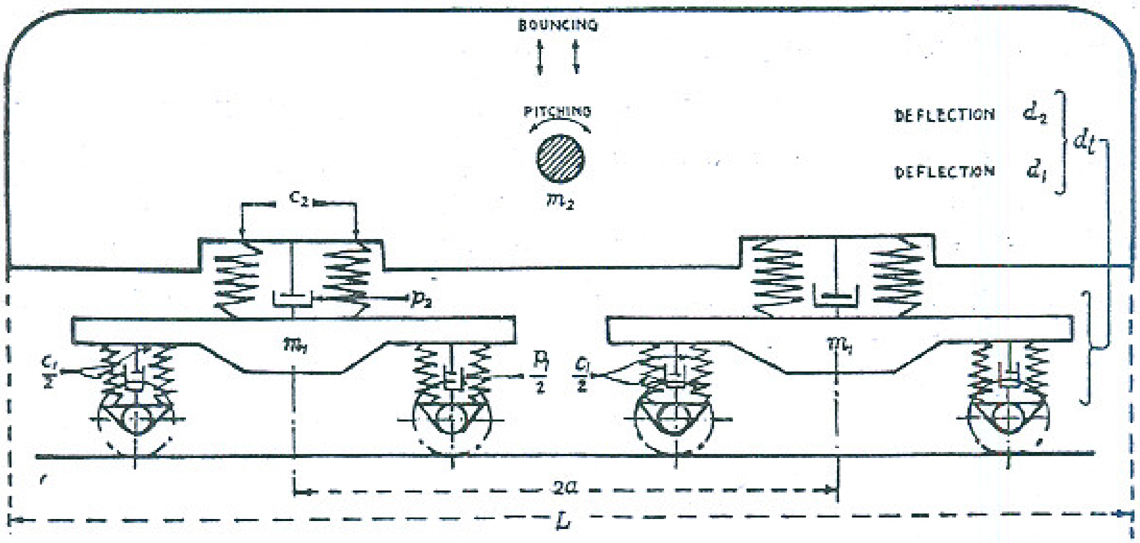

FIG: 11

The relative transmissibility of this system is given by the expression

$$\sqrt{\frac{\left(1+4D_2^2\lambda_2^2\right)\left(1+\frac{4D_1^2\lambda_2^2}{\mu p}\right)}{\left[\left(\lambda_{01}^2-1\right)\left(\lambda_{02}^2-1\right)-4D_1D_2\lambda_1^2\lambda_2^2\right]+4\lambda_2^2\left\{D_1^2\left(1-\lambda_2^2\right)\lambda_1+D_2\left[1-\frac{1}{\mu}\left(1+\frac{1}{\beta}\right)\lambda_2^2\right]\right\}^2}}$$

where $D_1=\frac{p_1}{\sqrt{2m_1c^1}}=\frac{p_1}{2m_1w_1}$

and

$$D_2=\frac{p_2}{\sqrt{2m_2c_1}}=\frac{p_2}{2m_2w_2}$$

p_1 and p_2 is the damping resistance in terms of force (lbs.) per unit of displacement velocity (ft. per sec.) at the primary and secondary suspension, respectively.

$$w_1=\sqrt{\frac{c_1}{m_1}} \text{ and } w_2 \sqrt{\frac{c_2}{m_2}}$$

$$\lambda_1=\frac{w}{w_1} \text{ and } \lambda_2=\frac{w}{w_2}$$

where w is the excitation frequency.

$$\lambda_{o_1}=\frac{w}{w_3} \text{ and } \lambda_{o_2}=\frac{w}{w_4}$$

$$\mu=\frac{c_1}{c_2} \text{ and } l=\frac{m_2}{m_1}.$$

6.7. It is apparent that the equation above is complex and working out by routine calculation would be time-consuming. One is, therefore, forced to make use of digital computers which simplify such calculations. Once the equations are set up, values with different damping factors, spring stiffness of primary springs as well as that of secondary springs at various frequency rations are calculated in 10 to 15 minutes only. The best combination is naturally selected for the suspension arrangement of the vehicle.

6.8. In the absence of a digital computer, this expression is slightly modified to give approximate value of the relative transmissibility to give approximate value of the relative transmissibility. The derivation and relation between various variables are shown below with reference to the above figure :—

m_1 & m_2 (lbs, sec.^{2}/ft.) are the mass of the bogie and vehicle body respectively.

Total m=W lbs./g ft. sec.^{2}

$$C_1 = \frac{\left(\frac{m_1+m_2}{2}\right) g \times 12}{d}$$

where d_1 & d_2 are the deflection of primary and secondary spring in inches.

$$1 = mr^2$$

where r^2 the radius of gyration for coaches is taken as .33L.

The excitation frequency $w=2\pi f$ where f=cycle per sec.

Bouncing frequency is given by

$$1 = mr^2$$

$$w_{12} = \sqrt{\left(\frac{c_1+c_2}{2m_1}+\frac{c_2}{m_2}\right) \pm \sqrt{\left(\frac{c_1+c_2}{2m_1}+\frac{c_2}{m_2}\right)^2 - \frac{2c_1c_2}{m_1m_2}}} \text{ rad./sec.}$$

Once we know $w_{1,\,2}$ the damping force can be calculated by the equation $D_2=2p_2/2m_2\, w_1\, 2\pi$

The pitching frequency is ascertained by

$$w_{3,\,4} = \sqrt{\left(\frac{c_1+c_2}{2m_1}+\frac{a\,c_2}{1_z}\right) \pm \sqrt{\left(\frac{c_1+c_2}{2m_1}+\frac{a_2c_2}{1_z}\right)^2 - \frac{2a^2c_1c_2}{m_1 1_z}}} \text{ rad./sec.}$$

Damping for pitching is then calculated by

$D_3=2p_2/(2\ 1_z\ w_3\ 2\pi)$ approximately equal to 0.2.

$$\text{Relative transmissibility R.T.} \approx \sqrt{\frac{1+4D+\gamma^2}{(1-\gamma^2)^2+4D_t^2\gamma^2}}$$

where

$$Dt = \frac{pt}{[2(n_1 m_1 + m_2)w_x]}$$

pt is total damping resistance

$$= pt \approx p_1\eta_1^2 + p_2\eta_2^2 \quad w_x = w_1 \text{ or } w_3$$

η_1 is relative primary spring deflection $= \frac{d_1}{dt}$

η_2 is relative secondary spring deflection $= \frac{d_2}{dt}$

$\eta_1 + \eta_2$ will be equal to 1.

$r = \frac{w}{w_1}$ or $\frac{w}{w_3}$ where w is excitation frequency.

6.9. The use of the above modified equation permits an appreciation of the effect of d_1 and d_2, and of c_1 and c_2, as well as that of m_1 and m_2 on the values of relative transmissibility, *viz.* w_1 or w_2. These enable the designer to make a choice of these variables, as well as the magnitude of D_1 and D_2 to ensure acceptable relative transmissibility values over the entire speed range of the vehicle, expressed in terms of the frequency ratio value r.

7. Evaluation of Running Quality of Coaches

7.1. The analysis of the lateral and vertical car body oscillation dealt with above and the effect of various parameters like spring stiffness, damping factor etc. do not enable one to correlate the human factor underlying riding comfort with the overall property of the suspension system. The problem was to spell out in mathematical terms the effect of the resultant oscillations on human sensations. This has been a subject of extensive study by the Rolling Stock Test Department of the Reisch Bahn at Berlin—Grunewald and Dr. Eng Sperling in which an attempt has been made to evaluate the effect of car body oscillations on human beings and to establish a mathematical expression in terms of amplitude and frequency of lateral and vertical vibrations and to arrive at an index value to specify the running quality of a coach. The concept underlying the evaluation of running quality of a vehicle is briefly dealt with below.

7.2. The running quality of a coach with respect to induced vibrations is determined by the following parameters :—

(*i*) Acceleration $b = \frac{d^2y}{dt^2}$ ....(22)

(*ii*) Variation of acceleration in unit time, termed as 'impulsion' or

'jerk' $r=\frac{d^3y}{dt^3}$ ....(23)

and (*iii*) Work done per unit of mass (amplitude of oscillation $\times$ acceleration)

$$y \cdot \frac{d^2y}{dt^2} \qquad \text{....(24)}$$

7.3. For sinusoidal vibrations, (where a is the amplitude and f is the frequency),

(*iv*) maximum acceleration $b_{max}=a.(2\pi f)^2$ ....(25)

(*v*) maximum impulsion $r_{max}=a.(2\pi f)^3$ ....(26)

(*vi*) work done in one vibration cycle$=a^2.(2\pi f)^2$ ....(27)

7.4. The product of the maximum impulsion (or jerk) and the work done in one cycle is taken as the *basis* for the evaluation of running quality. This is given by the expression$=a^3.(2\pi f)^5$

$$=C\times a^3 f^3 \qquad \text{....(28)}$$

7.5. Human sensitivity to vibrations has, however, been found not to be directly proportional to the above expression, but to vary according to an exponential law, *viz* :—

$$E=C\sqrt[10]{a^3 f^5}$$

7.6. The running quality evaluation mark ("Wertziffer W_z") is given by

$$W_z=2{\cdot}7\sqrt[10]{a^3 f^5} \qquad \text{....(29)}$$

wherein a =amplitude of oscillation in cms, and

f =frequency in *c.p.s.*

7.7. Recently, a corrective factor $F(f)$ has been introduced in the above expression, in order to take human reactions more closely into account. The corrected formula for the riding quality is

$$W_z=2{\cdot}7\sqrt[10]{a^3 f^5\, F(f)} \qquad \text{....(30)}$$

$$W_z=0{\cdot}896\sqrt[10]{\frac{b^3}{f}F(f)} \qquad \text{....(31)}$$

7.8. Fig 12 shows the values of the corrective factor $F(f)$ for various values of frequency f. The need for the corrective factor arose because a close study of human reactions to vibrations brought out that :—

(*i*) for any particular frequency, lateral oscillations result in a sensation of discomfort which is twice greater than that produced by the vertical ones, and (*ii*) the frequency range, 4—8 *c.p.s.*, produces the maximum discomfort, and frequencies above 50 *c.p.s.* do not contribute to the sensation of discomfort as they get filtered by the human body.

FREQUENCY CORRECTION FACTOR

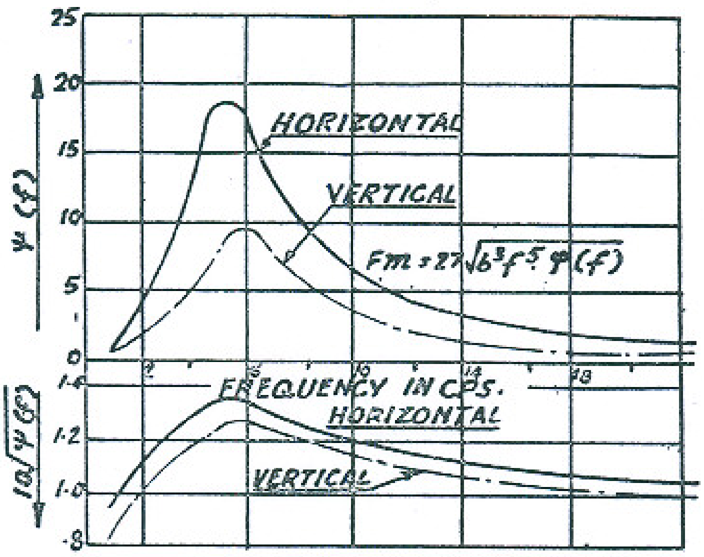

FIG: 12

7.9. W_z values for the vertical and lateral vibrations are calculated separately and their combined effect is assessed as follows :—

$$W_z = \sqrt[10]{W_{z(\text{lateral})}^{10} + W_{z(\text{vertical})}^{10}} \quad \ldots.(32)$$

7.10. Appreciation of Running Quality Evaluation Mark

7.10.1. To form a judgement about the running quality of a vehicle with respect of the evaluation mark W_z, the following scale of comparative merit has been standardised by the German Federal Railways :—

Evaluation Mark W_z.	*Running Quality Interpretation*
1.0	Very good.
1.5	Nearly very good.
2.0	Good.
2.5	Nearly good.
3.0	Satisfactory.
3.5	Still satisfactory.
4.0	Fit for operation. (Serviceable)
4.5	Unfit for operation.
5.0	Dangerous for operation.

7.10.2. The evaluation mark 3 to 3.25 has been laid down as the lower limit of running quality still acceptable in the case of passenger coaches, and the mark 4 to 4.25 in the case of goods stock.

8. Need for the trials

From the foregoing analysis it would be obvious that a bogie coming up to the requisite running quality cannot be developed from theoretical considerations only. On account of the number of variables involved, including extraneous factors like track conditions etc., the proper course is to work out preliminary designs of the alternatives and subject them to trials in order to arrive at the best suspension system suiting a particular set of conditions. With a view to evolving a new suspension arrangement for our coaches which would give a ride index value of 2.5 to 3, a series of trials were undertaken on this railway in collaboration with the German Firm M/s Link Hofmann Busch.

9. Proto-type test bogies and their characteristics

9.1. Messrs. L. H. B., designed five different types of suspension arrangements for the light weight integral type semi-tubular coaches. The spring characteristics and distribution of total static deflection between the primary and secondary springs are shown in Fig. 13.

9.2. In order to test all the above five alternatives, the Firm also supplied three sets of proto-type bogies in which any of the above suspension arrangement could be incorporated by suitably changing the springs of the secondary suspension. The proto-type bogies were similar to the LHB standard bogies, except that they were of partly riveted construction and had shock absorbers fitted to dampen oscillations of primary and secondary springs.

9.3. Out of the three sets of bogies, one set had a harder primary suspension and was designated as A, and a second set had softer primary suspension and was classified as B. Combinations 1, 2, 3 and 4 were achieved by suitably changing the springs of the bolsters

9.4. Two sets of bogies had arrangements for fitting laminated springs at the bolster. Laminated springs were suspended on to the frame of the bogie by either short or long shackles. The third set had arrangements for use of coil springs in the secondary suspension and had a spring plank which was suspended on to the bogie frame by either short or long shackles.

9.5. The five combinations of suspension arrangements were tested as under :—

(*i*) A_1—harder helical coil axle springs and laminated bolster springs with a total static deflection of 0.42 cm per MP (0.163 in per ton) distributed in the ratio of 40 : 60 between the primary and secondary springs.

(*ii*) B_1—softer helical coil axle springs and laminated bolster springs of total static deflection of 0.50 cm MP (0.194 in/ton) with a distribution ratio of 50 : 50 between the primary and secondary springs.

(*iii*) A_2—same primary springs as A_1 but with helical coil springs at the bolster. The total static deflection of this arrangement was .56 cm per MP (0.217 in/ton) with a distribution ratio of 30 : 70 between the primary and secondary springs.

(*iv*) A_3—also same springs at the axles as A_1 but with softer helical coil bolster springs than A_2 with a total static deflection of 0.83 cm. per MP (0.322 in./ton) distributed in the ratio of 20 : 80 between the primary and secondary springs.

(*v*) B_4—same helical coil springs as in B_1 at the axles but with helical coil springs at the bolster, instead of laminated springs of B_1, having a total static deflection of 0.82 cm. per MP (0.318 in./ton) distributed in the ratio of 30 : 70 between the primary and secondary springs.

9.6. As a result of the requirement to restrict variation in buffer height to 3 inches between tare condition and when loaded to twice the pay load, a limitation is imposed on the choice of specific deflection of springs. This requirement limits the specific deflection of bogies for lower class to 0.55 cm per MP (.217 in/ton). From the data given above it would be seen that as the A_3 and B_4 bogies have higher specific deflections they cannot be used for lower class coaches.

9.7. With regard to the damping arrangements, four types of dampers with different characteristics were provided for use in combination with the five alternative suspension gear arrangements. For the axle springs, the two types of dampers which were used had a damping capacity of (+1200/—1100 lbs.) and +440/—2100 lbs.), while for damping the bolster springs the two dampers had (+1110/—1110 lbs) and (+2100/—2100 lbs) damping capacity. The lower capacity damper at the axles was always fitted in conjunction with the lower capacity damper at the bolster, and the higher one at the axle with higher capacity damper at the bolsters.

10. Test Carriages

10.1. Coach Nos. SWGT 4006, WGNT 4302 WGNT 4303 and WGNT 4307 were used for testing the proto-type bogies. These are all light weight steel carriages built by M/s L.H.B. The tare weight of these carriages is 25 tons and they are 10 ft. 8 in. wide with a carrying capacity of 88 passengers. The lifting and lowering of these carriages were done at the Moghalpura Shops, as also the changing of various types of prototype bogies. After the first few trials most of the combination of spring gears were eliminated and in the end only one coach was required for carrying out further trials. Coach No. WGNT 4307 was used mostly for this purpose.

10.2. For trying out bogies having spring gear arrangement to A_3 and B_4 combination, meant for upper class coaches, the tare weight of upper class coaches was made up by an additional dead load which consisted of wagon springs loaded uniformly and the weight of coach No. WGNT 430.7 was thereby increased from 24 to 30.5 tons.

11. Selection of test track

11.1. For carrying out riding tests, it is essential that test runs be made under such conditions as will ensure results free of bias from any source. Therefore, exceptionally good or bad track is to be avoided for such trials. In order to be closer to the representative conditions, it is desirable that a coach should be run on more than one section of track, since track characteristics do vary to some extent from one section to another.

11.2. The aim of our trials was first to determine the effect of change of suspension parameters on the running quality, and then to select the best possible combination which would have good riding characteristics on the P.W.R. track. Hence, trial bogies were first run on the same test track in order to determine the ride index of each individual bogie under identical conditions. The best one amongst them was later tested on different sections of the main line to ascertain its behaviour in actual service conditions. For carrying out preliminary trials, a test track straight and level, with average maintenance standard and 2 km. ($1\frac{1}{4}$ miles) in length was chosen. Trials were made at 40, 50 and 60 m.p.h.

12. Test Track

The actual tests were carried out between Kissan and Renala Khurd and measurements recorded during the run between mile posts 484/24 and 486/6 in the up direction. The track in this section is straight and has a gradient of .03%. The track consists of rails of 100 lbs. per yard A.R.E.A. section 42 ft. in length, and steel trough sleepers spaced at 2 ft. with 12" ballast cushion.

13. Duration of trial

These trials lasted for an overall period of 3 months and 9 days, because they had to be done intermittently, as the trial coaches had to be brought to shops for changing the bogies, etc.

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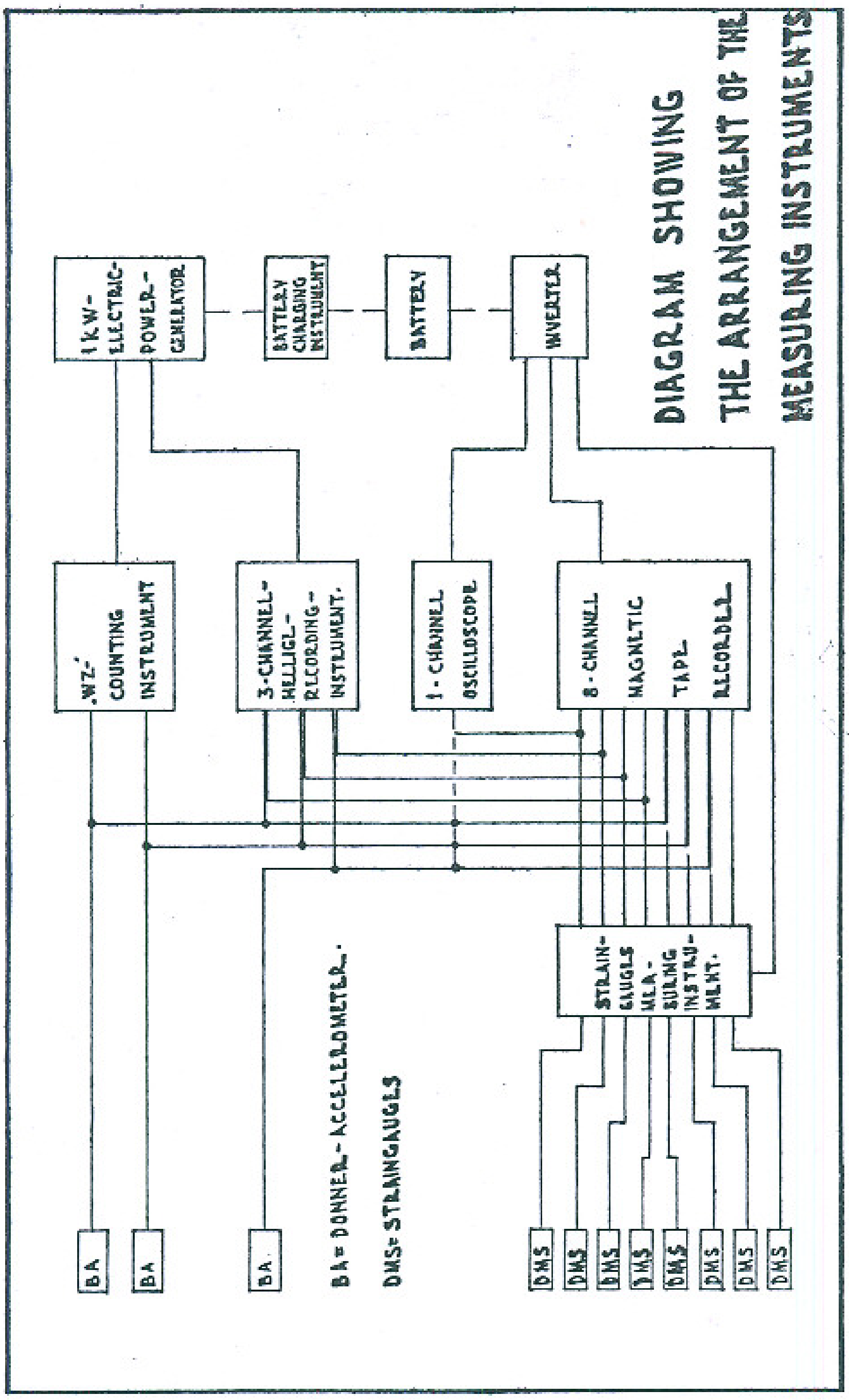

FIG: 14

14. Test Procedure

In order to determine the inherent property of a particular bogie suspension it is necessary that it should be loosely coupled as the last vehicle in the test train, so that free movement of the vehicle in all the planes is possible, without any restraint from neighbouring coach. Therefore, the trial train consisted of a 2000 H. P. Co Co locomotive followed by two coaches which were either to be tested on that day or had already completed their trial run. After these coaches WTLR 5328 was attached which carried all the recording instruments. Last of all, the coach under trial was loosely coupled and cables from the accelerometers and strain gauges were connected to the instruments in the WTLR. As mentioned earlier, the trial readings were only recorded during the run from Kissan to Renala Khurd in order to have the test coach always as the last vehicle without shunting and disturbing the recording arrangement. The test coach used to be in tare condition. The accelerometers were kept on the floor of the test coach at the rear end bogie, which transmitted shoks in the vertical and lateral planes directly to the analog computer and to the tape recorder for evaluation and record.

15. Instruments used on the trials

During the trials the following instruments were used to measure the Wz values, and also to find strains developed at various points of the bogies at different speeds :—

For measuring Wz values :—

(*i*) Inductance type Accelerometers.
(*ii*) Amplifiers Bridge Type (Single Channel).
(*iii*) Analog Computer.

For measuring strain :—

(*i*) Resistance Strain on Gauge.
(*ii*) Amplifier Bridges type W (Six Channels).
(*iii*) Magnetic Tape Recorder.
(*iv*) Three Channel Recorder.

Diagram showing the arrangements of the measuring instruments is given in Fig. 14.

A brief description of the working principles and operation of the instruments used for measuring the Wz value is given below :—

15.1. Accelerometer

15.1.1. The accelerometer consists of two symmetrical coils with a soft iron core in them. The iron rod which forms the core of the two coils is

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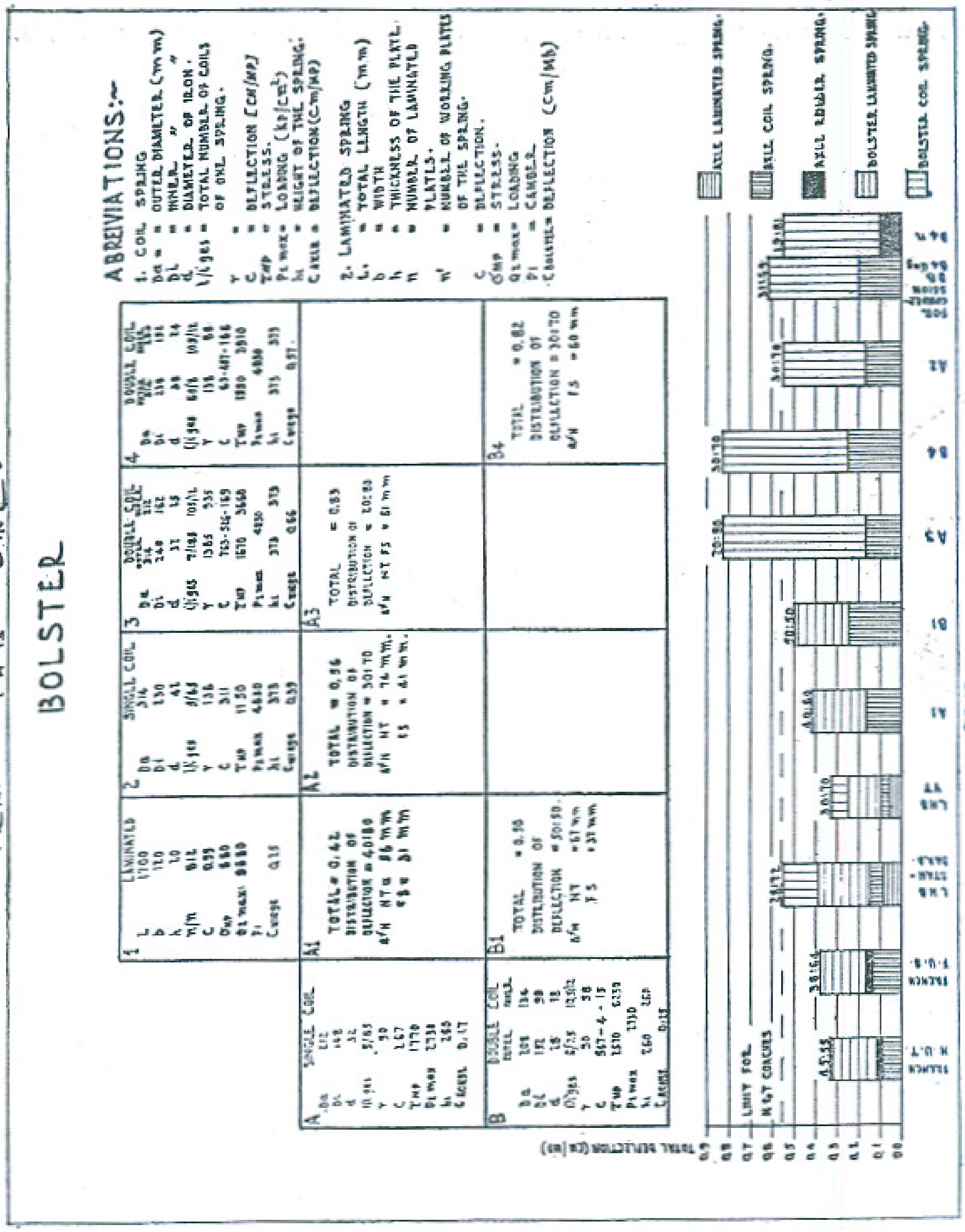
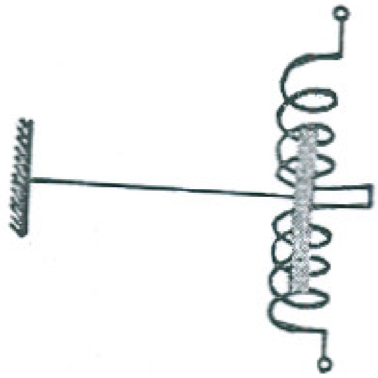
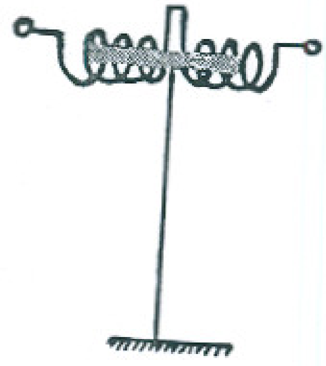


FIG: 13

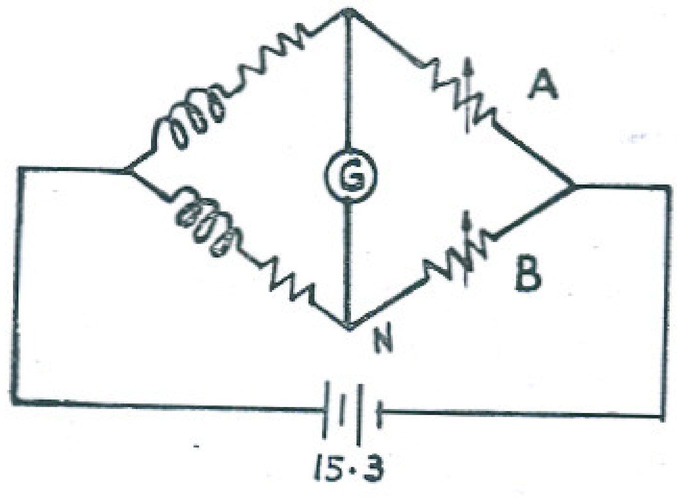
connected to the free end of a strip so that with any suitable vibration the plunger will move along the axes of the coils. Two such systems are mounted on one frame in such a manner that in one case the plunger is vertical and in the other case the plunger is horizontal as shown in the following diagrams.



15·1



15·2



15·3

15.1.2. It is clear from the Figure 15 that the plunger of the first figure is very sensitive to vibrations in the vertical direction, whereas the plunger of the second figure is equally sensitive to vibrations in the horizontal direction. The two systems can, therefore, be very satisfactorily used in detecting vertical and horizontal vibrations. The accelerometer is connected to the two arms of the bridge as shown above. Under normal conditions *i.e.* without any disturbance to the accelerometer, the wheatstone bridge is electrically balanced by adjusting the amplitude and relative phase of the currents in the two arms (A & B) of the bridge with the help of control knobs provided. Obviously, the bridge will be balanced when the voltages at points M & N are equal in magnitude and in exact phase opposition to each other.

15.1.3. When the plunger of any of the two systems moves due to vibrations, it will result in changing the inductance of the coils, thus throwing the bridge circuit out of balance. The amount of deflection at the Galvanometer will be proportionate to the amount of vibrations. If the bridge is properly calibrated, the exact quantity of the vibrations can be obtained from the reading of the Galvanometer.

15.2. **Calibration of the Bridge for Accelerometer**

15.2.1. The Accelerometer is connected to the Bridge circuit as shown in the diagram and the bridge is balanced by suitable adjustment of the amplitude control and phase control knobs provided in the two arms of the bridge.

The accelerometer is then turned by 90° and the deflection of the meter of the bridge is noted and is adjusted to any amount on the meter by the help of the controls provided for this purpose. Thus if the reading of the meter is 100 unit then 100 units$=1_g$ (Acceleration) (since to overturn a body, minimum acceleration needed is—1_g, *i.e.*, when the applied force can overcome 1_g effect of the gravitational force of the earth) and for any other reading of the meter corresponding value of acceleration can be calculated. When the instrument is set and adjusted like this, it is said to be calibrated. Since the computor can work only up to $0.2g$ these bridges are also calibrated for this value. This is accomplished in the following manner.

15.2.2. The output of the meter can be adjusted by a selector switch to 50, 100, 200, 500, 1000, 2000, 5000, 10000, 20000, 50000, 100000 ranges. Thus after placing selector switch at 1000 units range for 1_g value of the accelerometer by suitably adjusting the output control knob of the bridge amplifier. Having done so we say 1000 units on the meter$=1_g$. The selector switch is now turned to 200 units range and, therefore, full scale deflection of 200 units of the meter will now correspond to $\frac{1\times 200}{1000}=0.2g$. The bridge is, therefore, now calibrated for $0.2g$ value and thereafter the bridge knobs are not to be disturbed as otherwise the calibration of the bridge will be destroyed.

15.3. **Analog Computer**

15.3.1. An Analog Computer computes the value of any quantity of a Dynamical System in terms of electrical quantities of an Electrical System which is analogous to the Mechanical System originally under study. For doing so it employs a D.C. amplifier which by virtue of its construction has the following fundamental properties :—

(*i*) *High Gain.*—By virtue of its amplifying action its output is manifold its input and as such it can be used as a multiplier circuit in a Computer.

(*ii*) *Phase Reversal.*—By virtue of its property of reversing the phase of the output from that of its input, it can be used to create negative quantities if needed in a computing process.

15.3.2. By making suitable electrical circuits, the D.C. Amplifier can also be made to perform other most valuable mathematical operations, such as :—

(*a*) summing up of two or more variables,

(*b*) to differentiate a variable w.r.t. time,

(*c*) to integrate a variable w.r.t. time.

15.3.3. These circuits can be arranged suitably to give the solution of any mathematical equation which represents the behaviour of a Dynamical System in terms of Electrical quantities. This process of mutual inter-connection of fundamental circuits is called the programming of the system and forms the heart of the computer.

15.3.4. For computing the W_z value the following programmed circuit has been used :—

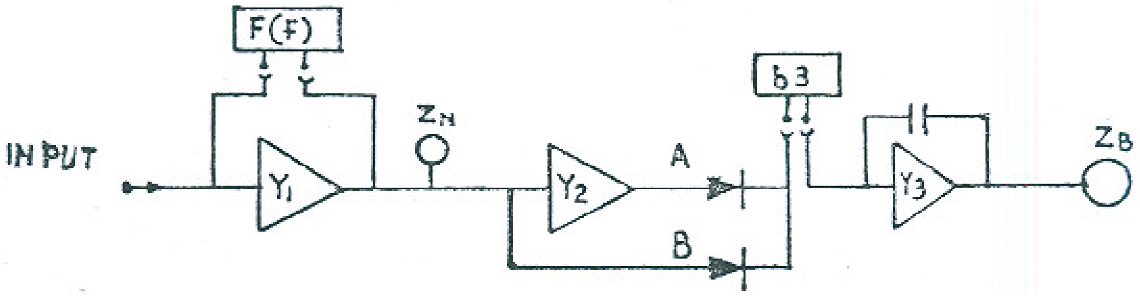


FIG: 15

15.3.5. This circuit gives the numerical value of

$\int \frac{b^3}{f}$ F(F) whereas the value of T in the formula :—

$$W_z^{10}=K\frac{1}{T}\int\frac{b^3}{f}.\ \ F(f)$$

is giving by a separate timing circuit and K is known by virtue of calibration to be 4.63 and 2.11 for horizontal and vertical channels respectively.

15.3.6. The output of the bridge is fed into this network in which functions b^3 & F (f) are generated in the boxes shown. These boxes contain suitably calibrated non-linear function generating circuitry the outputs of which are fed into the network as shown and every pulse of the incoming signal is measured by Z_n whereas amplifier V_2, Diodes A and B perform the full wave rectification of the signal fed to them by the previous stage. It is clear from the diagram that Diode 'B' will conduct +ive half cycle of the signal and will block—ive portion of the cycle. Amplifier V_2, however, changes the —ive portion into +ive cycle by virtue of its sign changing characteristics. The —ive portion of the cycle (originally —ive) is, therefore, also conducted by Diode 'A'. Thus all the incoming signals are rectified to +ive pulses by this stage and the result is fed to the final integrating stage along with the component b^3 coming from separate circuit. The output of the final stage thus gives the integral required which is measured on the meter Z_b.

15.3.7. The computer contains two such channels. They are used for measuring W_z values in the horizontal and vertical directions respectively.

15.3.8. Position F_1 of the switch gives the W_z values for goods wagons whereas F_2 & F_3 positions of the switch give the W_z values of passenger stock in horizontal and vertical directions respectively.

15.4. Calibration of the Computer

15.4.1. Since the Computer is built around D.C. amplifiers, which have their inherent problem of causing drift in the output due to their high gain qualities, their output is adjusted to zero value by the control knobs provided for this purpose. The integrating amplifiers Nos. 3 and 6 are then switched on and along with the programming units their outputs are also made zero by control screws f_1 and f_2. This would ensure that with zero input to the computer there will be zero output.

15.4.2. A known input of any magnitude from the bridge is then fed to the horizontal and vertical calculator and the amount indicated by them is noted down for the tests carried. It was seen that with an input of 0.2g the reading of the computer was 3 in the vertical direction and 3.9 in the horizontal direction. These are instrument constants and constant k of the formula is given by $k=(5/x)^3$ and for above value of x, k has been found out to be 4.63 and 2.11 for vertical and horizontal directions.

16. Discussion of Trial Results

16.1. In order to try out all the alternative suspension arrangements mentioned, we had to conduct 142 trial runs on the test track. The trials were so progressed that suspension arrangements which gave a W_z value higher than 3 at 60 m.p.h. were eliminated from further tests as the objective was to select two suitable suspension arrangements with ride index less than 3, one for the upper class coaches and the other for the lower class.

16.2. Results of these trials are tabulated in appendices to this paper. A study of the test results would reveal the fact that they are in close agreement to the theoretically anticipated results, and that the behaviour of various parameters influencing the riding quality was as expected from theoretical considerations.

16.3. A (1) Proto-Type Bogie

The results of A(1) suspension arrangement have been compiled at Appendix 1. This suspension arrangement has been tested in three different conditions, *i.e.* without any shock absorber, with 500 KP axle shock absorbers and then with 950 KP shock absorbers at the axles. Since the static deflection distribution was in the ratio of 40 : 60 between primary and secondary springs and the specific deflection of this suspension arrangement was 0.42 cm per MP (.163″ per ton), the ride index value at 60 m.p.h. in both the horizontal and vertical planes did not show any remarkable improvement on the previous tested bogies. Naturally this suspension arrangement was not tried out any further.

16.4 B (1) Proto-Type Bogie

16.4.A-1. The results of B (1) suspension arrangement with a total specific deflection 0.5 cm. per MP (0.194″ per ton) distributed in the ratio of 50 : 50 between primary and secondary, are shown at Appendix 2. A study of these results would indicate that the riding quality of this arrangement reacted sharply with the change in operating conditions. With short shackles and without any shock absorbers the total Wz value came to 3.15 at 60 m.p.h. which was not very encouraging. The use of 500 KP shock absorbers at the bolster and axles resulted in appreciable reduction of total Wz value which became 2.9. The improvement in the Wz value in the vertical plane is very prominent due to the provision of softer springs in this suspension. Contrary to expectation, incorporation of bolster links instead of the conventional guides worsened the running and the Wz value rose to 3.25 at 60 m.p.h.

16.4.2. Improvement shown by the provision of shock absorbers at the bolster and axle, led to further trials of this suspension arrangement with long shackles and at higher speeds. With long shackles and without any shock absorber the bogie showed the anticipated improvement in the lateral plane, and the total Wz value came down to 2.9 at 60 m.p.h., but it rose to 3.1 at 70 m.p.h. and thus exceeded the limiting value of 3. Provision of 500 KP axle shock absorbers also could not improve the Wz value and therefore further trial of this suspension arrangement was abandoned.

16.5. From the foregoing discussion it would be seen that bogie having laminated springs at the bolster were left out from further considerations due to their higher Wz values at 60 m.p.h. and above. The investigation therefore narrowed down to all-coil spring suspension arrangement.

16.6. A (2) Proto-Type Bogie

16.6.1. Results of the trial of A2 suspension arrangement having specific deflection of .56 cm. per MP (0.217″ per ton) distributed in the ratio of 30 : 70 between primary and secondary springs have been tabulated at Appendix 3. The better static deflection distribution, softer springs, long shackles, greater bolster spring base and provision of shock absorbers at the bolster and axles resulted in low total W_z values which came to 2.7 and 2.8 at 60 and 70 m.p.h. respectively. This arrangement was therefore tested with other variables in order to study the amount of deterioration, if any, in riding quality. Its behaviour under the tare load of upper class coaches was also investigated and it was found that the total W_z value remained as 2.70 and 2.8 at 60 & 70 m.p.h. respectively. With 950 KP shock absorbers the result showed deterioration indicating that the 950 KP shock absorber provided more than the optimum damping factor which raised the value from 2.80 to 2.85 for tare condition of upper class coaches. This test also established that the damping factor provided by 500 KP shock absorber was closest to the optimum damping factor under the conditions prevalent on the P.W.R.

16.6.2. Having established these facts, we tried the A2 suspension arrangement under adverse conditions as with worn out types, without shock absorbers etc. A very interesting phenomenon can be noticed from the trial results at 70 m.p.h. for the following three conditions *i.e.* (*i*) with worn-out tyres, with bolster and axle shock absorbers, (*ii*) with worn-out tyres and with bolster shock absorbers only, and (*iii*) with worn-out tyres and without any shock absorbers. For the first two cases, the Wz value remained the same *viz.* 3.0. With all the shock absorbers removed, the ride index value was recorded as 2.9 only *i.e.* there was an improvement on the removal of shock absorbers. The increase in Wz value with shock absorbers can be explained if it is recalled that dampers behave like amplifiers if the frequency ratio *i.e.* the ratio between the exciting frequency and the natural frequency, is greater than 1.41, and for conditions prevailing in this particular test run the frequency ratio at 70 m.p.h. appears to have exceeded 1.41. As the Wz value with this type of suspension arrangement came up to the required minimum it was selected for further trials on other sections of the track.

16.7. A (3) Proto-Type Bogie

16.7.1. Results of trials of suspension arrangement having the softest possible springs with a specific deflection of .83 cm. per MP (.332″ per ton) distributed in the ratio of 20 : 80 between the primary and secondary springs are tabulated at Appendix 4. A_3 bogies were first tested with short shackles and with 500 KP shock absorbers at the axles and the bolster, which gave a Wz value in the lateral plane of 2.75 at 60 m.p.h. while the total Wz value came to 2.6.

16.7.2. The next trial of A_3 was made with long shackles and there was a marked improvement in the Wz value in the lateral plane. At the speed of 70 m.p.h., the Wz value in the lateral plane came down to 2.55 and the total Wz value to 2.50 only. An increase of 0.05 was noticed in the total Wz value at 70 m.p.h. when the shock absorbers were removed either from the axles or from the bolster.

16.7.3. Trials carried out with worn-out tyres, first with shock absorbers provided at the axles and the bolster, and second with shock absorbers at the bolster only, at 70 m.p.h., resulted in the deterioration of the Wz value to 2.7 and 2.75 respectively, which still left sufficient margin with respect to the higher maximum acceptable value of three.

16.7.4. Results of trial runs of this type of suspension with eccentric wheel tyres (a condition which has remote possibility of being encountered in service) has also been recorded with both the axle and bolster shock absorbers fitted, and with bolster shock absorbers only. The total Wz value in this condition and at 70 m.p.h. came to 2.85 and 3 respectively. This indicated that A_3 bogie can be safely recommended to be used for upper class coaches and would ensure a ride index value lower than three even in adverse circumstances. A_3 bogie was also selected for further trials on the other sections.

16.8. **B Proto-Type Bogie**

The last type of suspension arrangement tried out was B_4, means for use in upper class coaches, having a specific deflection of .82 cm. per MP (0.318 inches per ton) distributed in the ratio of 30 : 70 between the primary and secondary suspensions and the results obtained are at Appendix 5. Since the design parameters are similar to A_3 suspension, except for the static deflection distribution, the results were more or less the same as that of A_3. The total ride index value of this type of suspension arrangement, with long shackles and with shock absorbers at the bolster and axles, came to 2.5 at 70 m.p.h. With the axle shock absorber removed it deteriorated to 2.55, while the value remained as 2.50 when the shock absorbers were provided at the axles only.

16.9. Because of the fact that the bolster springs exercise greater influence on the roll oscillations (see Equation 13) it is desirable that they should be as soft as possible. Care should however be taken at the same time not to make the axle springs unduly stiff, as stiffer axle springs would subject the bogie frame to high stresses. It has been experimentally determined that the optimum distribution ratio of specific deflections should be 20% to axles and 80% to bolster, and hence, A_3 bogie is preferable to B_4. Strain gauging of A_3 bogie, whose axle springs even though stiffer (sp : deflection .17 cm. per MP as against .25 cm. per MP of B_4), confirmed that the dynamic stresses induced in the bogie frame remained within permissible limits.

16.10. Trial runs of A_3 suspension arrangement on other sections of track were conducted between Lahore and Lalamusa at various mileages and speeds. The results therefrom have been compiled at Appendix 6. These trials were conducted with worn types and with shock absorbers provided at the bolster only. It was found that in none of the trial runs the Wz value exceeded the limit of three.

16.11. Results of extensive trial runs of A_2 bogie have been tabulated in Appendix 7. These trials were conducted between miles 101 and 31 on the Kotri-Karachi section with shock absorbers provided at the axles and bolster, and also without any shock absorber. The results obtained on this section indicated that Wz value exceeded the limit of 3. The track was in a poorly maintained condition and in order to establish that this suspension arrangement is none the more sensitive to bad track than any other suspension currently in use on P.W.R., an old L.H.B. bogie was tested between the same section of the track. Same amount of deterioration, as previously recorded for A_2 bogie, was noticed which confirmed that better riding quality could not be achieved by changing the parameters of bogie suspension arrangements if the track conditions remained the same as found between miles 101 to 31.

16.12. A_2 bogie was also run with the coach firmly coupled, and loosely coupled; the results of the trials are at Appendix 8. These trials were conducted between mile 30 and 102 on certain portion of this section, Wz value exceeded the limit of three. As the overall average of the Wz values for trial runs made under good as well as adverse track conditions did not exceed the limiting value of 3, A_2 bogie was considered as suitable for use with lower class coaches.

16.13. During the trials, a defect in performance of the bogie came to light. While running on the test track as well as in other sections, the bolster of the A_2 and A_3 bogie used to strike against the bogie sole bar. This defect could not be eliminated even when the clearance between the bolster and the sole bar was increased from $1\frac{5}{8}''$ to the maximum permissible limit of $2''$. The defect can, however, be overcome either by providing rubber spring stops at the sole bar, or absorbers which would restrict the lateral displacement of the bolster.

17. Conclusions and Recommendations

17.1. Detailed analysis of the test results given above brings out the fact that the best choice, considering the track conditions prevailing on the P. W. Railway is an all coil spring bogie with suspension arrangement characteristics corresponding to proto-type A_2 for the lower class and proto-type A_3 for the upper class coaches. The choice of these two suspension arrangements has further advantages in that both the types have axle springs of the same characteristics and their only difference is the specific deflections of the bolster springs. This simplifies maintenance and achieves standardisation to the maximum extent possible. It is, therefore, recommended that A_2 and A_3 types of suspension arrangement be adopted for the lower and upper class coaches respectively.

17.2. The test results establish the fact that condition of wheel tyres has a great influence on the riding quality of a coach. In order to maintain the wheel tyres in good shape in service for longer periods, the use of heat treated tyres, recently introduced on the Japanese National Railways should be examined for adoption on the P. W. Railway. Another practice which would help maintain the wheel tyres in good order for longer periods would be the use of composite brake shoes. Among many other advantages claimed by its manufacturers, its use results in imparting a polished surface to the wheel tread which has a good effect on ride index value of a coach. In order to study the economic aspects of adopting composite brake shoes in lieu of the cast iron brake blocks it would be necessary to try out these shoes in actual service.

17.3. The use of rubber fittings and shock absorbers might give rise to some maintenance problems. The extremes of climate and dust might affect the life of rubber fittings and reduce the efficiency of the shock absorbers and might even render them ineffective. Hence, in early stages of the introduction of A_2 and A_3 bogies, a close watch on the performance of the rubber fittings and shock absorbers would be necessary. It would also be necessary to provide adequate facilities for the proper maintenance shock absorbers.

APPENDIX 1

LHB64-A_1

Serial No.	Trial No.	Bogie, Type and Construction	Number of the Carbody	Weight of the Carbody (ts)	Speed (Mph)	Wz Horizontal	Wz Vertical	Wz Total
8	6	LHB64-A_1	WGT	25	40	2,55	2,75	2,65
		Short shackles, without			50	2,80	3,55	3,35
		SA	4006		60	3,25	3,30	3,25
9	8	LHB64-A_1	WGT	25	40	2,65	2,75	2,70
					50	2,85	2,85	2,85
		Short shackles, with 5000*kp*-ASA	4006		60	3,20	3,10	3,15
10	7	LHB64-A_1	WGT	25	40	2,70	2,80	2,75
					50	2,80	2,85	2,85
		Short shackles, with 950*kp*-SA	4006		60	3,15	3,10	3,15

SA = Shock absorbers. ASA = Axle shock absorbers.

BSA = Bolster shock absorbers.

APPENDIX 2

LHB64-B_1

Serial No.	Trial No.	Bogie, Type and Construction	Number of the Carbody	Weight of the Carbody (ts)	Speed (Mph)	Wz Horizontal	Wz Vertical	Wz Total
11	10	LHB64-B_1	WGNT	25	40	2,45	3,30	3,10
					50	3,00	3,10	3,05
		Short shackles, without SA	4302		60	3,30	2,85	3,15
12	14	LHB64-B_1	WGNT	25	40	2,40	2.90	2,75
					50	3,05	3,00	3,05
		With links, without SA	4302		60	3,45	2,80	3,25
13	13	LHB64-B_1	WGNT	25	40	2,45	2,65	2,55
					50	2,75	2,70	2,75
		With 500*kp* ASA and 500*kp* BSA	4302		60	2,95	2,90	2,90
14	11	LHB64-B_1	WGNT	25	40	2,10	3,30	3,05
					50	2,55	3,15	3,00
		Without SA	4302		60	2,85	2,95	2,90
					70	3,25	2,85	3,10
15	12	LHB64-B_1	WGNT	25	40	2,20	2,65	2,50
					50	2,75	2,75	2,75
		With 500*kp* ASA	4302		60	2,90	2,90	2,95
					70	3,05	3,05	3,05

SA = Shock absorbers. ASA = Axle shock absorbers.
BSA = Bolster shock absorbers.

APPENDIX 3

LHB64-A2

Serial No.	Trial No.	Bogie, Type and Construction	Number of the Carbody	Weight of the Carbody (ts)	Speed (Mph)	Wz Horizontal	Wz Vertical	Wz Total
16	25	LHB64-A_2	WGNT	31	40	2,40	2,30	2,35
					50	2,80	2,35	2,65
		With ASA and 950 *kp* BSA	4307		60	3,00	2,40	2,85
17	24	LHB64-A_2	WGNT	31	40	2.55	2 25	2,40
					50	2,85	2,30	2,70
		With ASA and BSA	4307		65	2,95	2,45	2,70
					70	2,90	2,55	2,80
18	19	LHB64-A_2	WGNT	25	40	2,40	2,35	2,35
					50	2,75	2,35	2,60
		With ASA and BSA	4307		60	2,85	2,35	2,70
					70	2,95	2,55	2 80
19	41	LHB64-A_2	WGNT	25	40	2,60	2,45	2,55
		With ASA and BSA			50	2,85	2.50	2.57
		with worn tyres	4307		60	3,00	2,60	2,85
					70	3,15	2,70	3,00
20	42	LHB64-A_2	WGNT	25	40	2,65	2,45	2,55
					50	2,90	2,50	2,75
		With BSA only	4307		60	3,05	2,65	2,90
		with worn tyres			70	3,10	2,80	3,00
21	43	LHB64-A_2	WGNT	25	40	2,60	2,50	2,55
		Without any SA			50	2,85	2,55	2,75
		With worn tyres	4307		60	2,90	2,70	2,80
					70	3,00	2,85	2,90

SA = Shock absorbers. ASA = Axle shock absorbers.
BSA = Bolster shock absorbers.

APPENDIX 4

LHB64-A3

Serial No.	Trial No.	Bogie, Type and Construction	Number of the Carbody	Weight of the Carbody (ts)	Speed (Mph)	Wz Horizontal	Wz Vertical	Wz Total
22	15	$LHB64\text{-}A_3$ Short shackles, with ASA and BSA	WGNT 4307	31	40 50 60	2,45 2,75 2,75	1,95 2,00 2,15	2,30 2,55 2,60
23	16	$LHB64\text{-}A_3$ With ASA and BSA	WGNT 4307	31	40 50 60 70	2,30 2,70 2,65 2,55	1,95 2,10 2,20 2,45	2,20 2,55 2,50 2,50
24	31	$LHB64\text{-}A_3$ With BSA only	WGNT 4307	31	40 50 60 70	2,30 2,55 2,55 2,55	2,05 2,15 2,35 2,55	2,20 2,40 2,45 2,55
25	17	$LHB64\text{-}A_3$ With ASA only	WGNT 4307	31	40 50 60 70	2,65 2,50 2,55 2,55	2,00 2,20 2,35 2,55	2,50 2,40 2,45 2,55
26	40	$LHB64\text{-}A_3$ With worn tyres, with ASA and BSA	WGNT 4307	31	40 50 60 70	2,40 2,60 2,55 2,70	2,20 2,30 2,35 2,75	2,30 2,50 2,50 2,70
27	39	$LHB64\text{-}A_3$ With worn tyres, With BSA only	WGNT 4307	31	40 50 60 70	2,45 2,65 2,65 2,85	2,30 2,35 2,50 2,65	2,35 2,50 2,55 2,75
28	34	$LHB64\text{-}A_3$ With extrs. eccentric tyres, with ASA and BSA	WGNT 4307	31	40 50 60 70	2,50 2,75 2,75 2,8	2,60 2,95 2,7 2,85	2,55 2,85 2,75 2,85
29	35	$LHB64\text{-}A_3$ With extrs. eccentric tyres, with BSA only	WGNT 4307	31	40 50 60 70	2,50 2,70 2,70 2,95	2,60 2,95 2,75 3,05	2,55 2,85 2,75 3,00

SA=Shock absorbers. ASA=Axle shock absorbers.
BSA=Bolster shock absorbers.

APPENDIX 5

LHB 64-B4

Serial No.	Trial No.	Bogie, Type and Construction	Number of the Carbody	Weight of the Carbody (ts)	Speed (Mph)	Wz Horizontal	Wz Vertical	Wz Total
30	27	$LHB64\text{-}B_4$	WGNT	31	40	2,30	2,10	2,15
		With ASA and BSA			50	2,60	2,30	2,50
			4307		60	2,55	2;35	2,45
					70	2,55	2,40	2,50
31	29	$LHB64\text{-}B_4$	WGNT	31	40	2,25	2,10	2,15
		With BSA only			50	2,65	2,25	2,50
			4307		60	2,60	2,40	2,55
					70	2,60	2,55	2,55
32	28	$LHB64\text{-}B_4$	WGNT	31	40	2,35	2,15	2,25
		With ASA only			50	2,65	2,20	2,50
			4307		60	2,60	2,35	2,25
					70	2,55	2,45	2,50

SA=Shock absorbers. ASA = Axle shock absorbers.

BSA=Bolster shock absorbers.

APPENDIX 6

Wz of coach No. 4307 with bogie type LHB64-A_3 with worn tyres and bolster shock absorbers only between Lahore and Lala Musa (26.01 1965)

Trial No.	Mile No.	Length of measuring section approx. (miles)	Speed (Mph)	Wz			
				Horizontal	Vertical	Longitudinal	Total
3,61	770	0,7	55	27	24	—	26
3,62	775	0,8	58	2,75	26	—	27
3,63	786	0,9	62	26	2,55	—	26
3,71	816	0,7	51	—	2,45	28	.
3,72	825	0,7	52	—	24	2,65	.
3,73	831	0,7	50	—	24	2,85	.
3,74	833 to 835	2,1	60 to 40	—	2,55	28	.
3,81	834 to 832	2,4	60	2,65	2,45	—	2,55
3,82	806	0,7	55	25	23	—	2,45
3,83	804	0,8	60	2,65	2,55	—	26
3,84	773	0,7	55	29	2,45	—	2,75

APPENDIX 7

Wz figures measured between Miles No. 101 *and* 31

Mile	Bogie $LHB64/A_2$								Old LHB B-"type bogie"			
	With axle and bolster-shock absorbeers				Without any shock absorbers							
	V Mph.	Horizontal	Wz Vertical	Total	V Mph.	Horizontal	Wz Vertical	Total	V Mph.	Horizontal	Wz Vertical	Total
100/15 to 99/15	—	—	—	—	60	3,10	2,75	3,00	58	3,80	3,35	3,65
92/4 ,, 91/4	—	—	—	—	55	3,05	2,80	2,95	53	3,40	3,45	3,40
86/18 ,, 85/18	—	—	—	—	58	3,15	2,75	3,00	50	3,30	3,25	3,25
84/24 ,, 83/23	50	3,05	2,65	2,90	59	3,10	2,80	3,00	58	3,50	3,30	3,40
80/24 ,, 79/24	57	3,35	2,75	3,15	59	3,20	2,80	3,05	60	3,95	3,40	3,75
78/24 ,, 77/24	55	3,25	2,65	3,10	60	3,20	2,75	3,10	59	3,95	3,25	3,75
71/12 ,, 70/22	55	3,30	2,80	3,10	58	3,30	2,85	3,15	59	3,75	3,55	3,65
68/20 ,, 67/20	—	—	—	—	61	3,40	3,05	3,30	57	3,85	3,60	3,75
66/1 ,, 65/1	—	—	—	—	52	3,25	2,80	3,10	—	—	—	—

[Contd.

APPENDIX 7—*Concld.*

Mile	Bogie LHB 64-A_2: With axle and bolster-shock absorbers				Bogie LHB 64-A_2: Without any shock absorbers				Old LHB B-"type bogie"			
	V Mph.	Horizontal	Wz Vertical	Total	V Mph.	Horizontal	Wz Vertical	Total	V Mph.	Horizontal	Wz Vertica	Total
65/15 ,, 64/15	—	—	—	—	—	—	—	—	55	3,60	3,35	3,50
63/21 ,, 62/21	—	—	—	—	—	—	—	—	60	4,00	3,45	3,80
60/14 ,, 59/14	—	—	—	—	59	3,45	2,90	3,30	60	4,10	3,40	3,90
52/21 ,, 51/21	—	—	—	—	—	—	—	—	56	3,75	3,55	3,65
45/24 ,, 44/24	55	3,45	2,85	3,30	—	—	—	—	—	—	—	—
43/22 ,, 42/22	—	—	—	—	53	3,25	2,90	3,10	53	3,35	3,60	3,50
39/24 ,, 38/24	59	3,40	2,90	3,25	55	3,35	3,10	3,25	53	3,70	3,60	3,65
33/21 ,, 32/21	—	—	—	—	—	—	—	—	59	4,00	3,20	3,75
32/21 ,, 31/21	60	3,50	2,75	3,30	—	—	—	—	—	—	—	—
684/24 to 686/6	60	3,00	2,60	2.85	60	2,90	2,70	2,80	60	3,50	3,15	3,35

APPENDIX 8

W_z measured between Miles 30 and 102

LHB64-A_2 bogie with bolster shock absorbers only (with worn tyres)

Mile			Speed (Mph)	Wz Horizontal	Wz Vertical	Wz Total
				COACH TIGHT COUPLED		
31/22	to	32/21	59	3,15	2,70	3,00
35/19		36/19	51	2,90	2,70	2,85
40/23		41/23	53	3,15	2,80	3,00
45/1		46/1	51	3,00	2,75	2,90
47/9		48/9	52	3,15	2,75	3,00
63/24		65/1	50	2,85	2,65	2,80
71/2		72/2	57	3,35	2,90	3,20
				COACH LOOSE COUPLED		
80/1		81/2	58	3,25	2,75	3,10
83/23		85/5	60	3,00	2,65	2,85
92/1		93/1	57	3,15	2,65	3,00
94/8		95/8	55	3,15	2,75	3,00
100/9		101/9	47	2,70	2,60	2,65

PROTO TYPE ALL COIL SPRING BOGIE.

PLATE-I

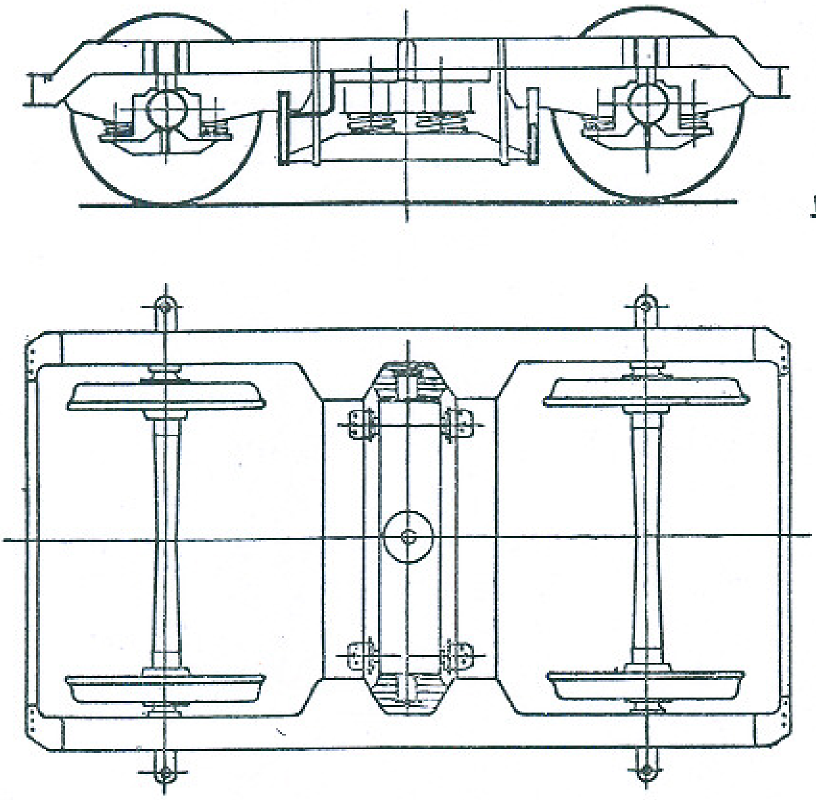


FIG: NO.1

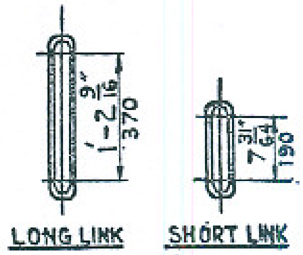


FIG: NO-2

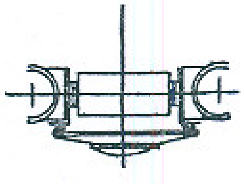


FIG: NO-3

LENGTH OF SOLE BAR OVER HEAD STOCK		14'-10 1/32"	4522 mm	
WIDTH OF BOGIE BETWEEN SOLE BAR		6'-7 3/4"	2025 mm	
WHEEL-BASE		9'-6"	2896 mm	
WHEEL DIAMETER		3'-7"	1092 mm	
AXLE BEARING		SELF ALIGNING DOUBLE ROW TWO BEARINGS PER BOX		
AXLE GUIDES		NARROW AXLE GUIDES WITH MANGANISE LINER AS SHOWN IN FIG. No 3.		

		Nº PER BOGIE	DRG: REFERANCE
AXLE SPRING	a... COIL	16	17522,02 B
	b... LAMINATED	—	—
BOLSTER SPRING	a... COIL	—	—
	b... LAMINATED	2	17522 B

		LONGITUDINAL		LATERAL	
DISTANCE BETWEEN SPRING	AXLE SPRING	1'-10 7/16"	570 mm	7'-4"	2235 mm
	BOLSTER	5'-5 15/16"	1700 mm	7'-4"	2235 mm

EFFECTIVE LENGTH OF BOLOSTER SWING HANGER .... AS SHOWN IN FIG No 2

PLATE–II

PROTO TYPE BOGIE WITH LAMINATED SPRINGS AT BOLSTER.

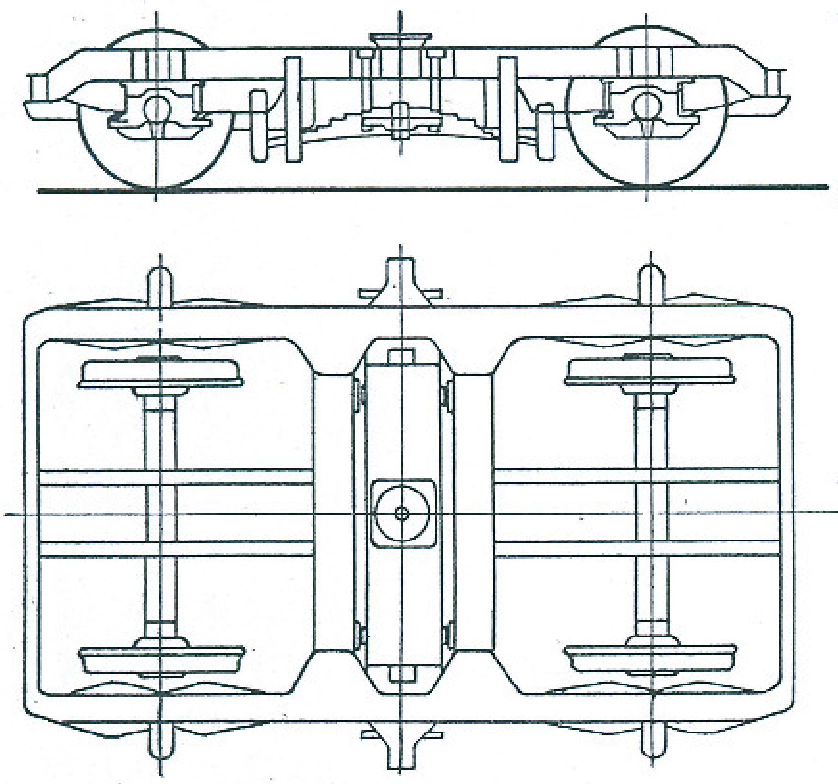


FIG: NO.1

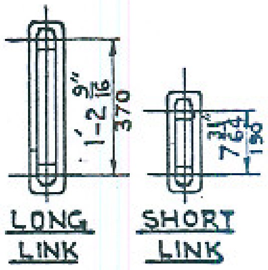


FIG: NO 2

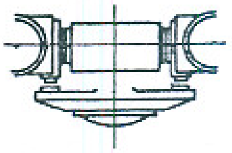


FIG: NO.3

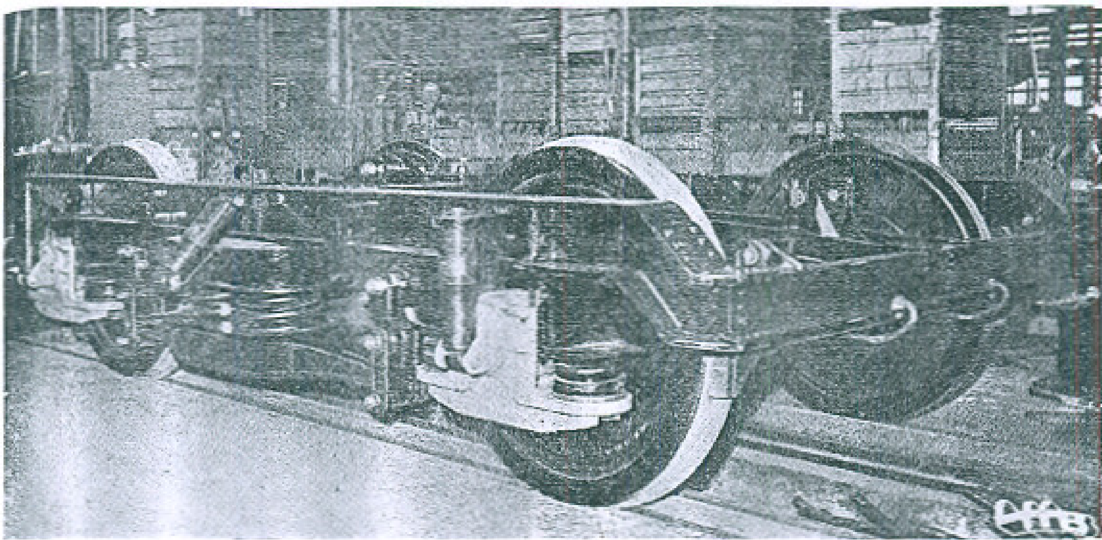
LENGTH OF SOLE BAR OVER HEAD STOCK	$14'-10\frac{1}{32}''$	4522 mm.
WIDTH OF BOGIE BETWEEN SOLE BAR	$6'-7\frac{3}{4}''$	2025 mm.
WHEEL BASE	$9'-6''$	2896 mm.
WHEEL DIAMETER	$3'-7''$	1092 mm.
AXLE BEARING	SELF ALIGNING DOUBLE ROW TWO BEARINGS PER BOX.	

AXLE GUIDES — NARROW AXLE GUIDES WITH MANGANISE LINER AS SHOWN IN FIG: Nº 3

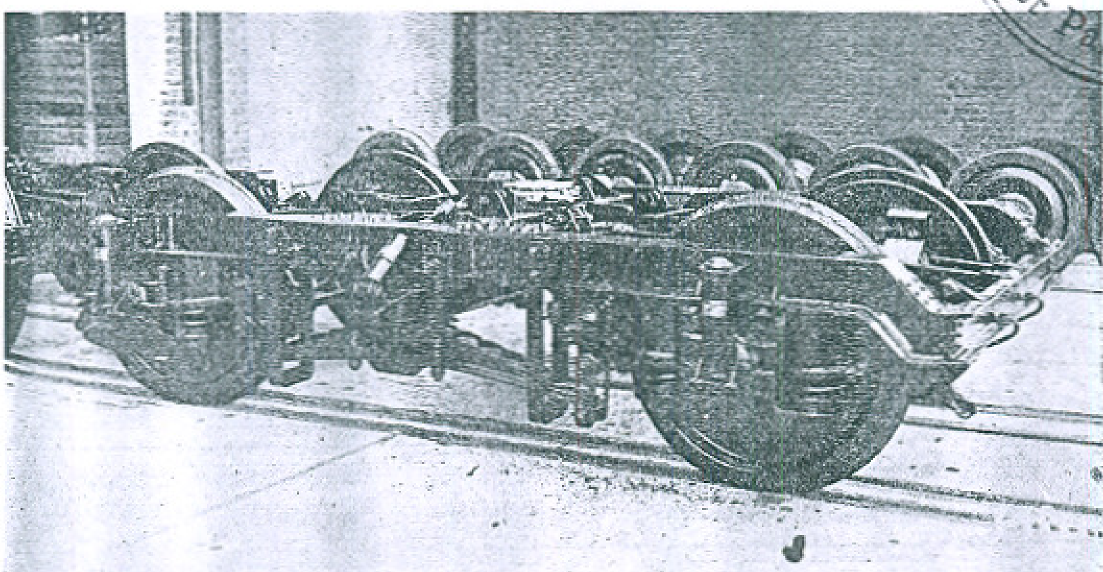
		Nº PER BOGIE	DRG: REFERANCE
AXLE SPRING	a — COIL	8	17522,02 A.
	b — LAMINATED	—	—
BOLSTER SPRING	a — COIL	—	17522,04.40 A
	b — LAMINATED		—

		LONGITUDINAL		LATERAL	
DISTANCE BETWEEN SPRINGS	AXLE SPRING	$1'-10\frac{7}{16}''$	570 mm.	$7'-4''$	2235 mm.
	BOLSTER "	$5'-6\frac{15}{16}''$	1700 mm.	$7'-4''$	2235 mm.

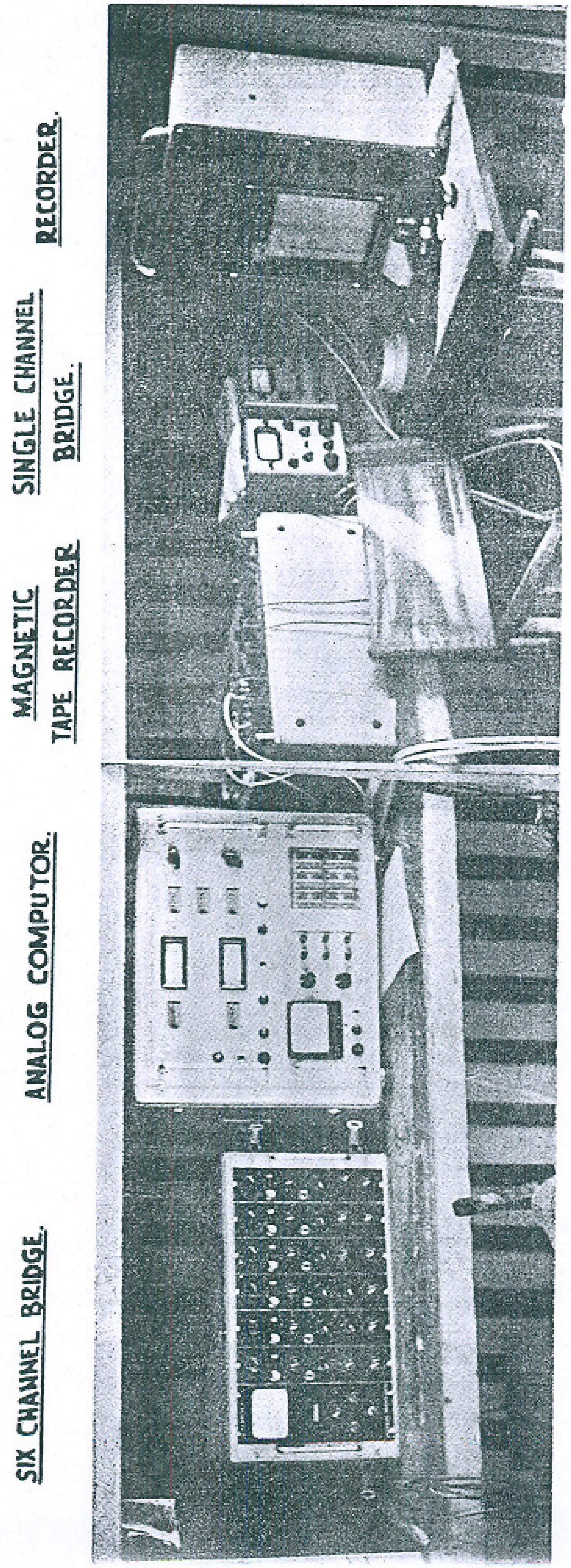
EFFECTIVE LENGTH OF BOLSTER SWING HANGER — AS SHOWN IN FIG: Nº 2



All Coil Spring Proto-Type Bogie.



Proto-Type Bogie with Laminated Springs at the Bolster.



Measuring Instruments used on the Trail.