Experimental Methods for the GRP Bogie Structure Integrity Assessment

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Abstract

A cost effective bogie is developed at a revolutionary design by the use of advanced fibre reinforced plastics composites operating under extreme load bearing conditions in a hostile environment.

This 22.5 tonnes per axle bogie comprises an upper and lower bogie GRP frame. These frames are moulded direct to the final shape and under a load the bogie frames deflect to provide the suspension part of the bogie. The composite GRP bogie is designed allows the wheels to follow more accurately the line of the track by self-steering effects of this bogie, with consequently a corresponding reduction of the track forces and improved the ride quality.

The concept of GRP bogie was evaluated at fifth-scale model by making, instrumenting and testing two bogies which have been in service more than 9 years fitted to a passenger wagon at Southampton’s Lakeside Railway to study the interactions between the track and the wagon. Every feature of the full scale bogie has been replicated including the axle boxes, central pivot point and the disc brakes. Two different suspension types were monitored during the runs at the Eastleigh railway track: a steel-bogie and a GRP-bogie equipped wagon.

A shaker test rig has been developed for testing the whole rail vehicle at various performance conditions as well as with various suspension types to get these interactions using electro-hydraulic testing system Schenck. Several tests were performed like drop test, sweep tests, track profile test and so on to assess the properties and structural integrity of the developed bogie.

Keywords: Glass fibre reinforced plastic GRP, bogie, damping, shaker rig

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1. GRP Bogie

1.1. Bogie description

This 25.5 tonnes per axle bogie comprises an upper and lower bogie GRP frame (Figure 1a). These frames are moulded direct to final shape and under load the bogie frames deflect to provide the suspension part of the bogie (Figure 1b).

The properties of the advanced glass fibre reinforced plastic (GRP) composites allow the functions of frame, suspension and damper elements to be integrated, thereby reducing the number of discrete components and joints. Unlike steel, the use of such materials allows the bogie frame to be tuned to provide the optimum stiffness properties in the principal orthogonal directions. The use of resilient elements in the axle box and at discrete parts of the bogie frame provides inherently high damping compared to steel, with the additional benefit of substantially reducing generation and transmission of noise.

The latter results in substantially improved the ride quality and a reduction in the track force, particularly under laden conditions when most track damages can occur. The composite bogie is designed to be more compliant in these directions, which allows the wheel to more accurately follow the line of the track, with consequently a corresponding reduction in the track forces and improved the ride quality.

![Fig. 1: GRP25 bogie (a) and (b) its double stiffness vertical deflection curve.](image)

1.2. Fifth-scale GRP bogie

The concept of GRP bogie was evaluated at the fifth-scale (Figure 2) by making, instrumenting and testing two bogies which have been in service for more than 9 years fitted to a passenger wagon at Southampton’s Lakeside Railway to study the interactions between the truck and the wagon. Every feature of the full scale bogie has been replicated including the axle box, central pivot point and the disc brakes. Two different suspension types were monitored during the runs at the Eastleigh railway track: a steel-bogied wagon and a GRP-bogied wagon.
1.3. Tests on full scale of one-half GRP bogie – side beam

The main investigations were performed with the full scale set bogie composite beams (bottom axle tie, compact lower beam and split upper beam, connected with axle boxes), just forming one side of composite bogie (see Figure 3). The vertical loading was performed through a steel plate with the help of one hydraulic cylinder 250kN with the stroke of 250 mm, controlled via system Labtronic, both produced by IST. The lower beam was provided with three pairs of strain gauges positioned at edges from the bottom side (two near axle boxes - T1, T2 and T5, T6), one in the middle – T3, T4). Two strain gauge rosettes R1, R2 were put to the lower beam side wall, one near the axle box, the other at the beam centre for checking of the sheer strain. The beam deflection was measured with a laser displacement sensor. The acting force and displacement of the hydraulic cylinder were measured as well.

1.4. Shaker rig test

The shaker rig has been developed for testing of the whole rail vehicle at various performance conditions as well as with various suspension types to get these interactions using electro-hydraulic testing system (Figure 4). Using this
The shaker rig has been assembled using clamping and support system Schenck 4000 with additionally manufactured parts just having the possibility of exciting each wagon wheel. Only one boogie was assembled on the wagon and loaded. The second boogie was disassembled. The wagon was supported under the second boogie with the rigid cube. Two hydropuls actuators Schenck PL 400 kN with two-step valves were situated under both boogie wheels at rear axle, two hydropuls actuators Schenck PL 630 kN with three-step valves were situated under both wheels at front axle. The wagon’s wheels were standing with its flanges on short steel beams, with no permission of its movement in longitudinal direction. The wheels were un-braked.

The actuators were controlled with Schenck control unit Labtronic 8800 with control programs individually prepared for each test.

1.4.1 Test instrumentation

The wagon was instrumented with four inductive displacement sensors HBM WA 300, measuring the relative displacement between wheel and wagon chassis at each wheel DSi, inductive accelerometers HBM B12, measuring the acceleration of each wheel VAWi, acceleration of front axle central point (wheels 1-2) VAA12, acceleration of rear axle central point (wheels 3-4) VAA34, acceleration of chassis at steel beam just over both wheels of front axle VACH1, VACH2 and in the middle between axles at this beam at both sidewalls BACH13, BACH24 (see Figure 5).

At the same time the displacement of each actuator as well as the force acting between the wheel and rail was measured with each actuator internal HBM displacement sensors DWi and loading cells LWi.

For the data acquisition, three the measuring amplifiers SPIDER8 were used, which were controlled using common lap top through software Catman 5.0, both from HBM, Germany.
1.4.2 Drop test

This test enables to find out the natural frequency and the critical damping of the suspension to be determined dynamically. The EU (Directive 96/53/EEC) drop test for road vehicle was adapted to rail, as there is no test method for measuring the critical damping dynamically for rail vehicles.

The drop tests have been performed with simultaneous drop of the front axle 1-2 actuators of 5 mm and 10 mm and with all four axles 1-4 actuators to excite the natural frequency. From the initial wagon position, the front axle actuators were slowly lifted out to the drop height and then the drop was realized with as rectangular as possible drop of them.

The damping ratio has been resolved by measuring the exponential decay of successive peaks \( p_i, p_{i+1} \) of vehicle chassis displacement or acceleration. The relative damping coefficient as logarithmic decrement \( \delta \) and the critical damping ratio are then calculated using the formula

\[
\delta = \frac{p_i}{p_{i+1}}
\]  

Damping ratio \( = \frac{1}{2\pi} \cdot \delta \)  

A typical drop test is shown in Figure 6 and the natural frequency and dynamic damping are shown in tables 2 and 3 as a function of drop height and load.
1.4.3 Sweep test

The frequency sweep test 0 – 30Hz has been carried out, which enables to identify all of the significant resonance peaks including the sprung and un-sprung mass peaks and body modes of the vehicle and their associated amplitude (Figure 7).

During the sweep test, first both actuators of front axle and then all actuators of both axles were linearly sinusoidal excited with constant amplitude of 0.5 and 1.0mm. The frequency of the harmonic actuator oscillation was varied from 0 to 30 Hz with the velocity 0.2Hz·s⁻¹.

So called bump test was realised, where the actuators were excited in phase. Comparison of theoretical suspension frequencies, calculated using the formula (3) and the peak measured frequencies for bump test can be realised.

\[
f_r = \frac{\pi \cdot g}{2} \cdot \sqrt{\frac{R}{F_m}}
\]

(3)

\[R\] spring rate [N.mm⁻¹]

\[F_m\] mean static load at one wheel [kN].

A typical time history of sweep test is given in Fig. 7.
1.4.4 Track profile test

The track simulation tests have been made by simulation of 1km sections of the UK freight acceptance route (Derby to Carnforth) recorded by the track recording vehicle digital data (samples of rail height at each 0.2476m for BQ2, 4, 6 and 0.2589 m for BQ9) to be able to compare the influence of railroad quality to the wagon behaviour. The 1 km sections BQ 2, 4, 6 and 9 were manually chosen from the whole record according the UK track quality bands classification, which vary from bands 1 to 12.

To have the possibility for comparison of these chosen track profiles and other profiles, the spectral density function has been calculated for each tested profile section.

Both left and right rail unevenness were simulated with actuators at each axle. For each quality band belongs a recommended velocity interval. The velocity simulation was realized by varying the time distance between each railroad sample in controlling actuator programme. The displacement signal of rear axle was realized by the time
delay of the front axle signal, the time delay was calculated as the ratio of the distance between both wagon axles to the tested velocity.

The tested track profiles are given in Figure 8a for right side. The power spectral densities spectra for all track sections and right rail are drawn in Figure 8b. For each track quality, more velocities were tested. The dynamic load coefficient DLC has been calculated from the sampled LW data, supposing, that the wear or damage of railroad is related to this coefficient as it is known for pavements, according the relation

\[
DLC = \frac{S_F}{F_m},
\]

(4)

where \( S_F \) is standard deviation of the dynamic load and \( F_m \) is mean dynamic load (static component, which is equal the load of one wheel).

The dynamic acceleration coefficient DAC is calculated from the sampled body accelerations, supposing, that the resistance to derailment is related to this coefficient

\[
DAC = \frac{S_a}{a_m},
\]

(5)

where \( S_a \) is standard deviation of the acceleration and \( a_m \) is mean acceleration.

2. Conclusions

The several test procedures were created and used for the both classical steel and GRP bogie investigation. The shaker rig method developed for bogie vehicles as well as axle wagons. This methodology allows to study dynamics characteristics of the bogies as well as their suspensions in laboratory conditions while a problem can be to measure these properties directly in operation while the homologation is missing.

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