STRUCTURAL RESPONSE OF A RAILWAY WAGON TRAVERSING A DIPPED RAIL JOINT

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INTRODUCTION

On typical jointed track, as opposed to continuously welded rail, the main source of high vertical dynamic loads coincides with the rail joints which occur at approximately every 18 metres. As the wheel passes over the joint the track deflects and, depending on the local sub-structure conditions, the rail assumes a varying degree of permanent set with the depth of the centre of the dip varying from zero to 25 mm in extreme cases. Measurements during track testing and calculations using mono-cycle models running over idealised dipped-joints (1) have led to the design of wagon suspensions on the basis of their response to typical dipped-joints at various vehicle speeds. For a dip depth of 20 mm and a speed of 20 m/s, the peak force transmitted through a modern 2-axle suspension varies approximately $\pm 30\%$ about the mean (static) force in the fully laden condition and rather more than that in the unladen condition. As a result of this and of earlier experience of weld failures, the $\pm 0.3g$ factor has become almost standard on BR as a rule-of-thumb method for calculating dynamic stress range when designing wagon frames against fatigue. The factor is applied to the static fully-laden stresses as calculated manually or by finite element methods and the resulting maximum and minimum stresses are compared with the allowable values at 2×10^6 cycles, the so-called fatigue limit as defined in BS.153(2) for various classes of weld.

This is a conservative approach in several respects. Both ends of the vehicle are considered to strike the dip at the same instant and it is assumed to run fully laden and at its maximum speed continuously. Conversely, work done on suspension response using assumed structural modes and on flexible bogie frames using time-history response methods have shown that the 'g-factor' approach, which implies a rigid structure, can considerably underestimate the stresses if the frame is flexible and there are large concentrations of mass. In addition to this, more recent work on the fatigue of weldments shows that more damage may result from the large numbers of low amplitude stress cycles than from the relatively few high amplitude ones.

In order to gain more understanding of wagon structural dynamics, particularly in terms of stresses, it was decided to investigate the flexible response of a laden 2-axle container wagon (Fig. 1) traversing a symmetrical dipped-joint. The calculation was done using the Newmark-Beta method (3) incorporated in the British Rail NEWPAC finite element program. This note summaries the method and gives a brief selection of the results obtained.

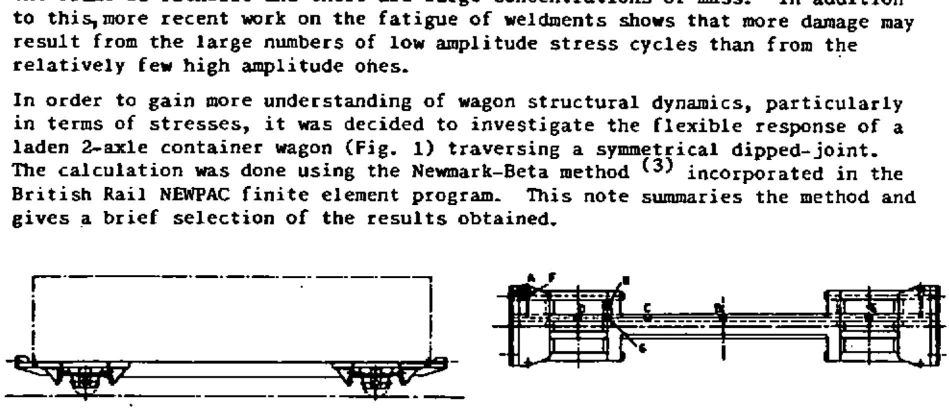

Fig. 1. Two-Axle Container Wagon

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REPRESENTATION OF WAGON AND DIPPED-JOINT

For the response calculation the wagon frame and 40 ft. ISO container were represented by a 3-D finite element beam model (Fig. 2) incorporating linear mass, spring and damper elements to represent the suspensions and flexible track. The idealised suspension and track system and assumed parabolic dip form are shown in Fig. 3.

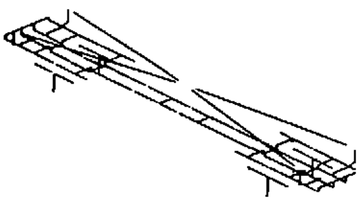


Fig. 2. Beam Idealisation

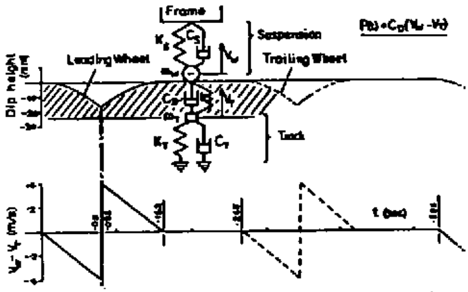


Fig. 3. Idealised Suspension & Dip

The track stiffness, damping and equivalent mass were assumed to be uniform throughout the dip and the whole system was assumed to be symmetrical about the centre-line of the track thus enabling only half of the wagon and one rail to be modelled. A 16 mm dip of 5.4 metres span was found from mono-cycle models to give approximately ±30% change in suspension force at 33.6 m/s and this was used in the response calculation. The bending moment excursions obtained were divided by the corresponding static values and the resulting factors applied to static stresses obtained from a detailed F.E. plate model (Fig. 4) to give dynamic stress increments.

DYNAMIC SOLUTION

In matrix terms the equation to be solved is $M\ddot{z} + C_F\dot{z} + Kz = P(t)$ where M, C_F, K and P(t) are the condensed mass, damping, stiffness and forcing matrices in terms of the master degrees of freedom in the model, which are chosen to adequately represent the main modes. For the half-model 342 total degrees of freedom were condensed to 28 masters. The damping matrix, $C_F = C + aM + bK$ where C is the matrix of damper elements representing the suspension and track and a and b are constants to provide a degree of structural damping. For this calculation 2% critical damping at 50 Hz was chosen, giving a = 6.28 and b = 0.000637.

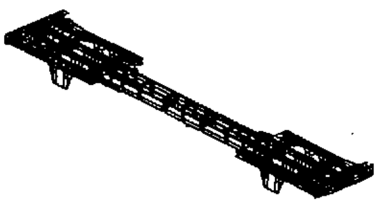


Fig. 4. Plate Idealisation

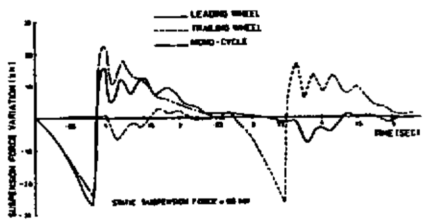
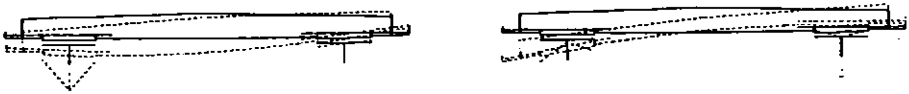


Fig. 5. Suspension Force History

The forcing function P(t) was applied to a very stiff damper of value $C_D = 10^{10}$ Ns/m inserted between the track mass and the wheel to give the required linear increase and decrease of vertical velocity through the parabolic dip. To avoid the sudden impulse caused by a step change in velocity at the bottom of the dip a short parabolic transition was inserted of total time 0.002s - equivalent to 0.067 m.

RESULTS

As the first wheel runs down the dip the spring extends and the damper resists the axle movement thus giving a reduction in suspension force (Fig. 5) of approximately 31% at the bottom of the dip when the spring has extended to about 10 mm and the frame above the leading axle has dropped about 5 mm as shown in Fig. 6a.



a) 0.09 sec. b) 0.146 sec.

Fig. 6. Deflected Shapes at Two Instants in Time.

When the wheel strikes the other half of the dip it is accelerated rapidly upwards, the initial rate of acceleration being dependent largely on the wheelset mass and track mass, stiffness and damping. The suspension force rapidly increases as the spring compresses resulting in an upward peak 17% greater than the static value. The bending wave thus produced travels along the frame and reaches the trailing suspension shortly afterwards. As the leading wheel comes out of the dip the frame bends the other way as shown in Fig. 6b. When the trailing suspension passes over the dip an almost identical force variation occurs. The results for a corresponding mono-cycle model, in which the container and flexible frame are replaced by a single equivalent mass, show a similar suspension force variation but with peaks of +27% and -28% above and below the static value.

The computed acceleration histories at two points in the frame are shown in Fig. 7 and are similar in form to those measured on running vehicles. The centre of the frame (B, Fig. 1) sees the bending pulses from each axle in turn followed by oscillations apparently corresponding to the first (9.4 Hz) and second (5.9 Hz) symmetrical bending modes. The accelerations at A, which is attached to the massive container, are low compared with those in other parts of the frame.

The bending moment at B (Fig. 8) clearly shows a contribution from the first bending mode and that at C, which is nearer to the suspension, shows also some anti-symmetrical bending (29 Hz). The two major cycles at C give peaks of +25%, -35% and +22%, -19% compared with the static value, giving a 25% greater dynamic range than that shown by the suspension force. The table below gives a summary of the dynamic factors at a number of points in the beam model and of the peak stress values obtained by factoring the corresponding static stresses from the detailed plate model. According to Reference 2 these stresses are not damaging even in low-classification weld regions.

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Location (Fig. 1)	Static Stress in Plate Model (MN/m^2)	Dynamic Factors (g) in Beam Model		Stress Peaks (MN/m^2) in Plate Model	
		Max.	Min.	Max.	Min.
B	-52	0.28	-0.25	-39	-67
C	-55	0.31	-0.26	-41	-72
D	-89	0.28	-0.25	-67	-114
E	-89	0.31	-0.26	-66	-117
F	+83	0.26	-0.25	105	62
G	-59	0.29	-0.24	-45	-76
H	-37	-0.29	0.22	-26	-45

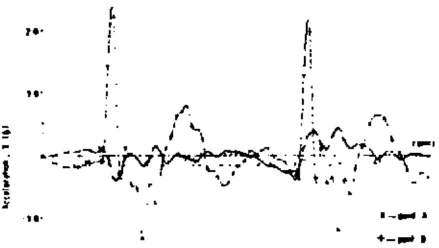

Fig. 7. Acceleration Histories

Fig. 8. Bending Moment Variations

CONCLUSIONS

The suspension forces and maximum dynamic stress ranges obtained from the flexible model response are close to those found using a simple mono-cycle. However the flexible model shows that each axle produces a similar stress cycle as it passes over the dip and that significant increments arise from frame bending modes. The close agreement with the mono-cycle on peak dynamic factors may therefore be partly coincidental.

Because the accelerations are not directly linked to stress values at the same position they are of little use in determining dynamic stresses.

Examination of the bending stresses at several points on the frame seems to indicate that a modal analysis would require at least five flexible modes to give similar results.

REFERENCES

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