British Rail Class 91 Locomotives

GEC ALSTHOM

Class 91 locomotives

The Class 91 locomotives have been designed to provide high speed services on the East Coast Main Line (Kings Cross to Edinburgh) and the West Coast Main Line (Euston to Glasgow). The equipment has been conservatively designed in order to provide high availability and high reliability together with long life.

1. The operational specification

British Rail are currently electrifying the East Coast Main Line from London to Edinburgh Fig 1 and the Class 91 was ordered as a new design of high speed locomotive for the route.

BR required the locomotive to undertake passenger duties for both day and sleeper trains together with fast parcels traffic over the whole of the BR electrified network. On the East Coast Main Line it will haul trains of up to 11 MkIV coaches and a driving trailer Fig 2. The driving van trailer enables push-pull operation of the trains which increases utilisation of the locomotive fleet by reducing the number of shunting movements at terminal stations. The Class 91 locomotive is also required to haul sleeper trains of up to 750 tonnes over the 1.5% gradients over Shap and Beattock summits on the Anglo-Scottish West Coast Main Line (London to Glasgow).

The specification requires the locomotive to be able to haul tilting coaches and the Class 91 has, therefore, been designed for cant deficiencies of up to 9° and provided with a 175 kVA 3-phase inverter to supply tilting power to the train.

Fig. 1 Map of BR electrified lines

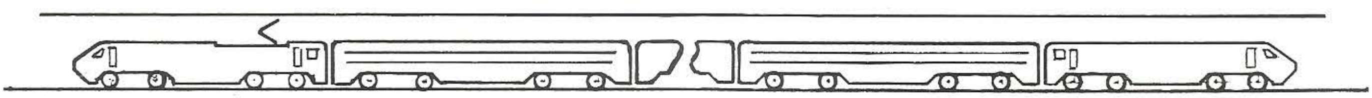


Fig. 2 Train formation: Class 91 locomotive, trailer coaches plus driving van trailer.

2. The technical specification

The specification requires the locomotive to run at 225 km/h with occasional overspeeding to 240 km/h without damage to systems. To further reduce journey times the locomotive also has to travel around curves at up to 9° cant deficiency.

The locomotive maintains full stability over this speed range with a combination of wheel/rail conicities between 0.05 to 0.4 and creep values between 0.5 and 1 × Kalkers calculated values. This is achieved without increasing the existing level of lateral forces and vertical forces; Prud'homme limits and P_1, P_2 forces respectively. (References 1 and 2).

The tractive effort — speed curve requires 20% adhesion at starting with emergency push-out duty at a level of 24% adhesion. Continuous power rating is 4.54 MW at 153 km/h together with a power of 3.75 MW at 225 km/h. Fig 3.

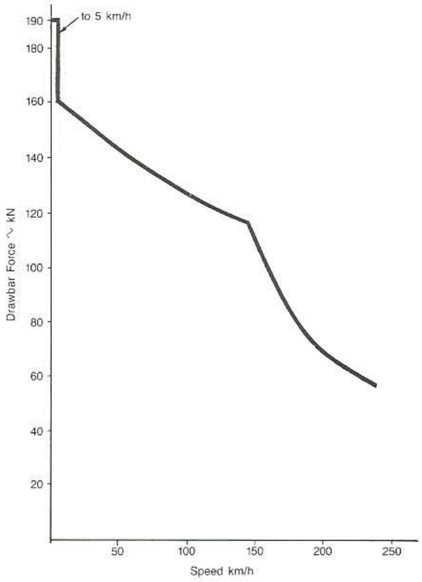


Fig. 3 Tractive effort/speed curve.

3. Transmission

The lateral track force targets have been met by reducing the mass of the bogie, achieved by suspending the traction motors from the vehicle underframe. The traction motors drive the bogie mounted gearboxes via short, low friction cardan shafts Fig 4. This arrangement gives the added advantage of a bogie with a low yaw inertia to assist stability. As a rough comparison the gearbox and drive effectively weigh 1200 kg compared with a traction motor of 3300 kg.

The transformer is also mounted on the underframe of the locomotive and this together with the underframe mounted traction motors, gives the body a low centre of gravity. A low centre of gravity helps to reduce body roll when the locomotive is curving at high speeds and helps to maintain good contact with the overhead catenary.

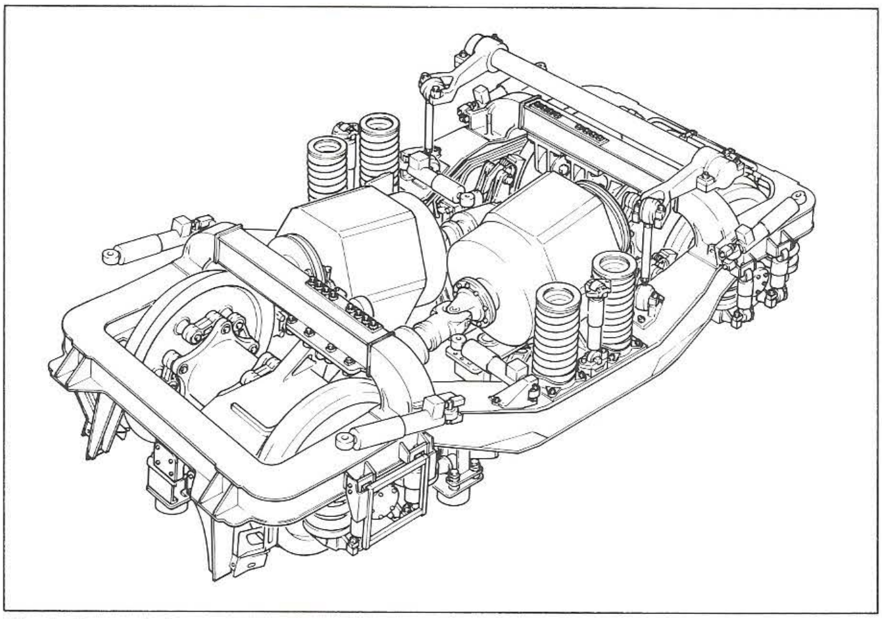
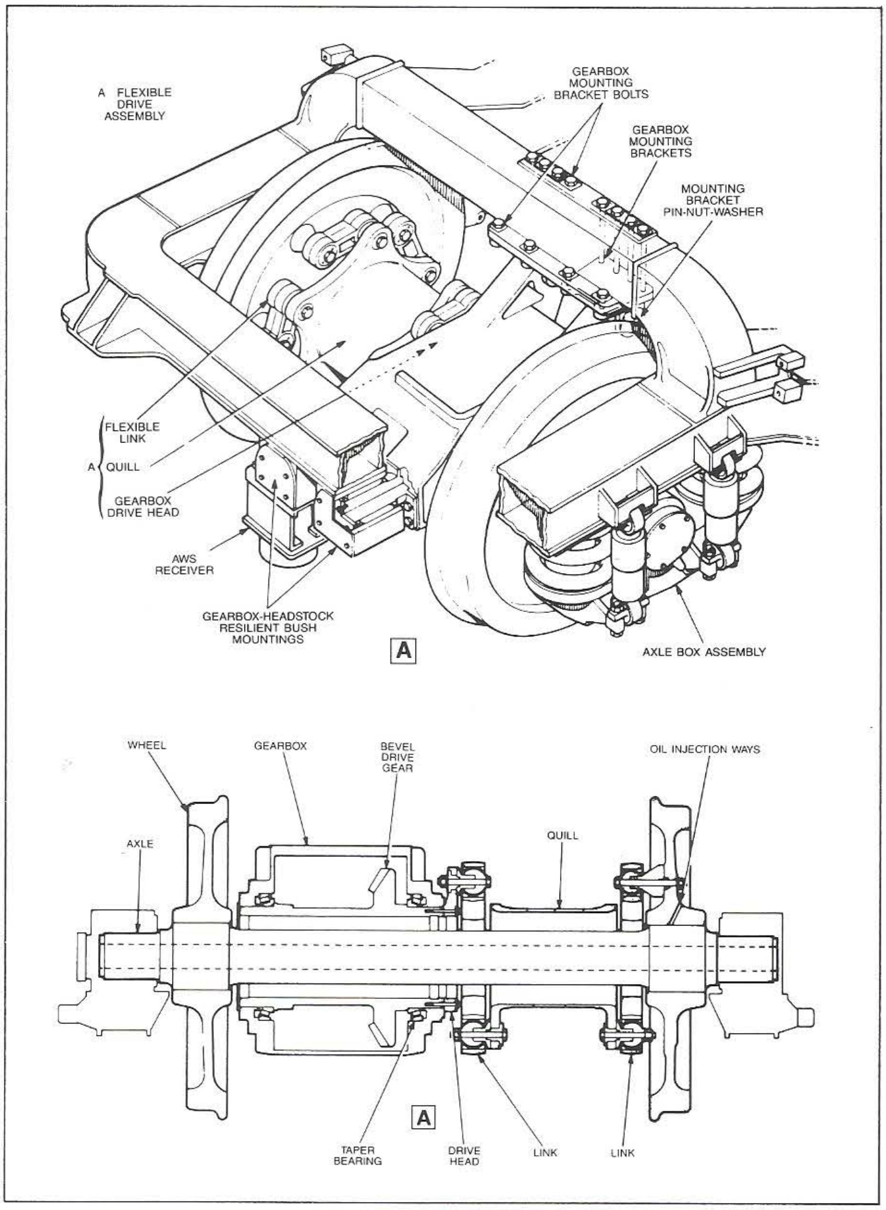


Fig. 4 Transmission system.



The P_1 and P_2 vertical track forces are kept to an acceptable level by designing the wheelset to have a low unsprung mass by using a flexible quill coupling drive system to the wheelset. Fig 5.

Fig. 5 Final drive arrangement showing flexible quill.

The final-drive coupling connects the gearbox to the wheelset and transmits the driving torque. As the gearbox is flexibly mounted on the bogie frame and the wheelset can move vertically, laterally and longitudinally with respect to the bogie frame, the flexible coupling has to be able to accept these movements. The coupling uses a quill tube driving through 5 links which are allowed to move by rubber bushes at their ends.

The final drive coupling was tested in the laboratory both statically and dynamically to demonstrate the integrity of the components and the capability of the rubber bushes. It was subjected to an artificial and exaggerated load situation of full rotational speed whilst offset both radially and axially at its maximum displacement.

The final drive gearbox Fig 6 is a single spiral-bevel wheel/pinion system with a hollow output shaft through which the axle passes. The power to be transmitted within the space limited by loading gauge and speed of the bearings made the gearbox a high risk item. It was, therefore, thought prudent to go forward with two designs of gearbox; Voith of Heidenheim, West Germany and David Brown of Huddersfield being the chosen suppliers.

The resultant design of the two boxes is very similar but with differences in gear-cutting processes (Voith use Klingelnberg and David Brown the Gleason method) input pinion bearing arrangements and lubrication system. At starting, the lubrication of the gears and bearings is by weir and splash until about 20 km/h when the oil pump provides forced lubrication.

The gearbox designs have been subject to various design reviews and audits which resulted in minor modifications to the designs prior to the manufacture. The standard test programmes carried out by each manufacturer also revealed the need for further adjustments to lubrication systems and bearing pre-loads.

A rig was set up at GEC Mechanical Laboratories at Stafford to test a gearbox at full speed/torque inputs together with the offset loadings due to cardan shaft and quill drive displacements. This demonstrated the performance of the gearbox bearings and their lubrication system. In addition further analysis was made of the gearbox vibration noise patterns for future comparison with service measurements.

Further test work was carried out on a transmission system with a simulated traction motor inertia. Parameters were evaluated which were used in a mathematical model to calculate the torsional resonant frequency of the system. This model has also been used to predict the behaviour of the transmission system to a step torque at the wheelset resulting from a change on wheel/rail adhesion levels in a stick/slip mode.

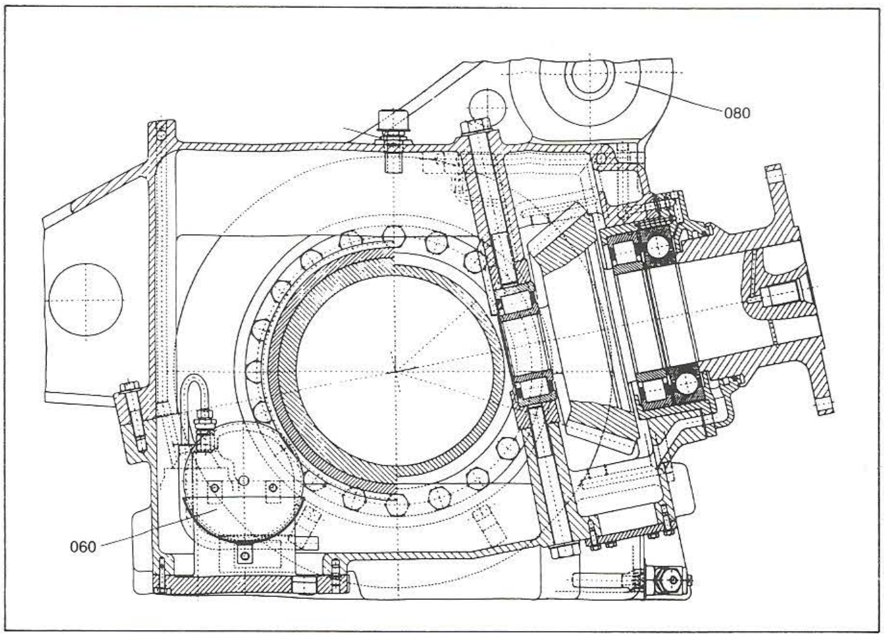
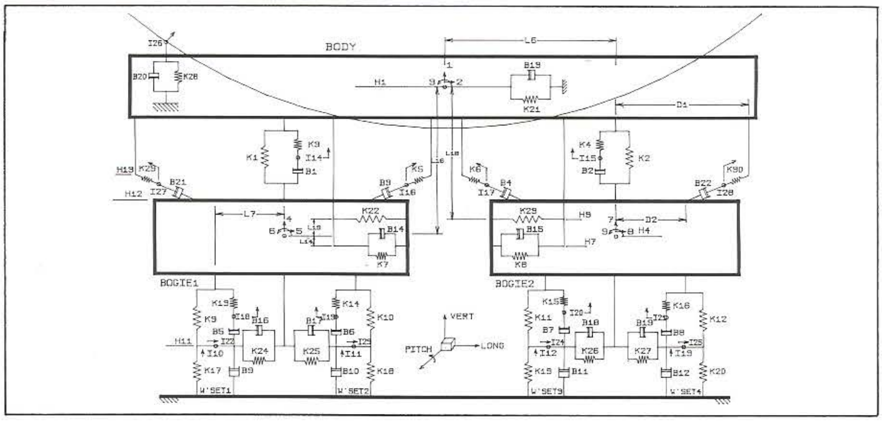


Fig. 6 Final drive gearbox.

4. Bogie

The parameters for the suspension components were evaluated using a dynamic model of the suspension system developed by the GEC Engineering Research Centre's Mechanical Laboratories at Stafford, based on work carried out by British Rail Research at Derby. The model was used to ensure that the chosen parameters met the specified ride and stability characteristics. Figs 7 and 8.

The bogie frame is a ladder design, fabricated from welded structural steel plate with integral steel castings, to give a life of 35 years under service conditions. The steel castings are primarily used at complex joints where the stresses would be unacceptably high for welds in a fatigue environment.



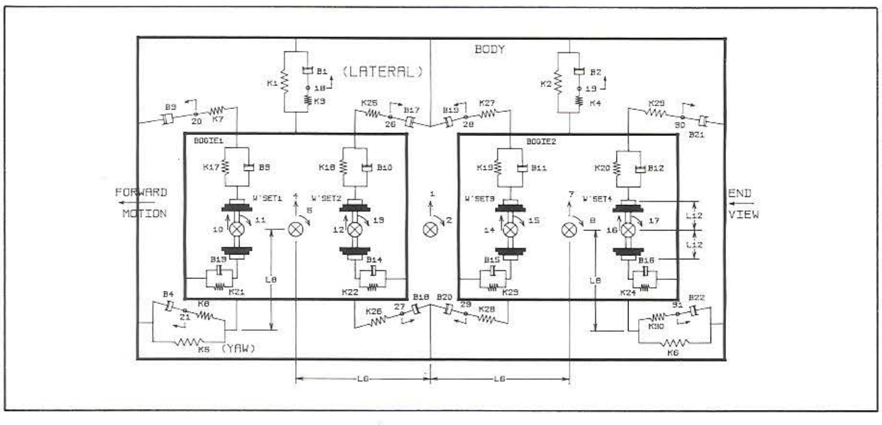


Fig. 7 Lateral model of vehicle suspension.

Fig. 8 Vertical model of vehicle suspension.

The primary suspension system Fig 9 comprises of coil springs to give the vertical stiffness, rolling rubber ring units to provide the necessary lateral and longitudinal restraint with damping by hydraulic dampers. The secondary suspension uses flexicoil springs and vertical dampers, the lateral restraint being by two dampers and rubber bump stops. Rotational stability is ensured by four inclined yaw dampers arranged longitudinally. Traction restraint is by a linkage connecting the bogie frame to the body structure pedestals which allows the relative rotational movement between body and bogie.
A further consideration in the design of the vehcle suspension system was the requirement to achieve a pantograph sway of 120 mm when curving to 200 km/h, 9° cant deficiency and a lateral dynamic acceleration on the body. This led to the requirement of a secondary anti-roll bar on each bogie.

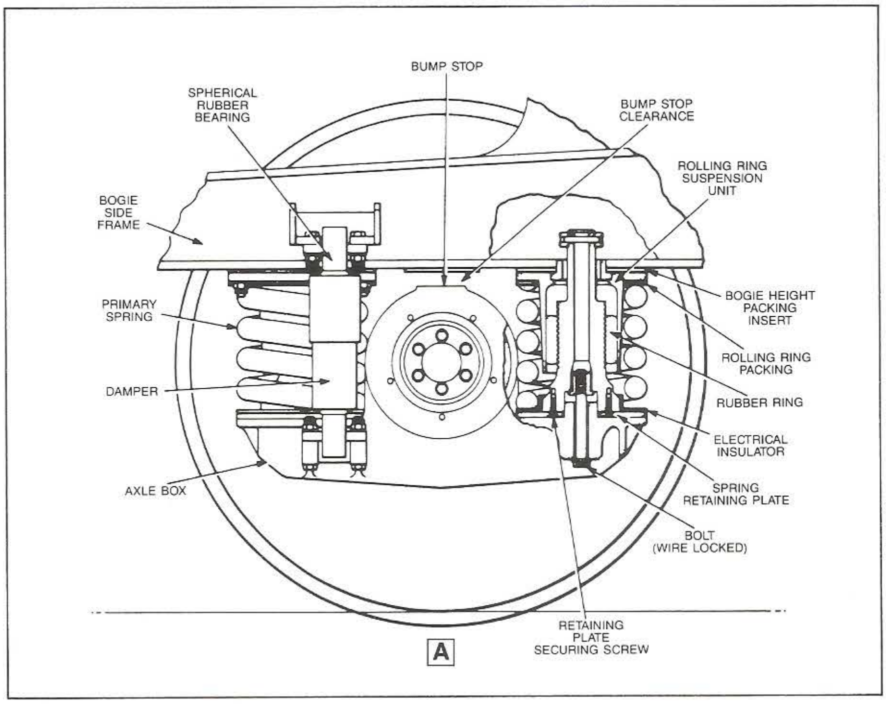


Fig. 9 Bogie primary suspension.

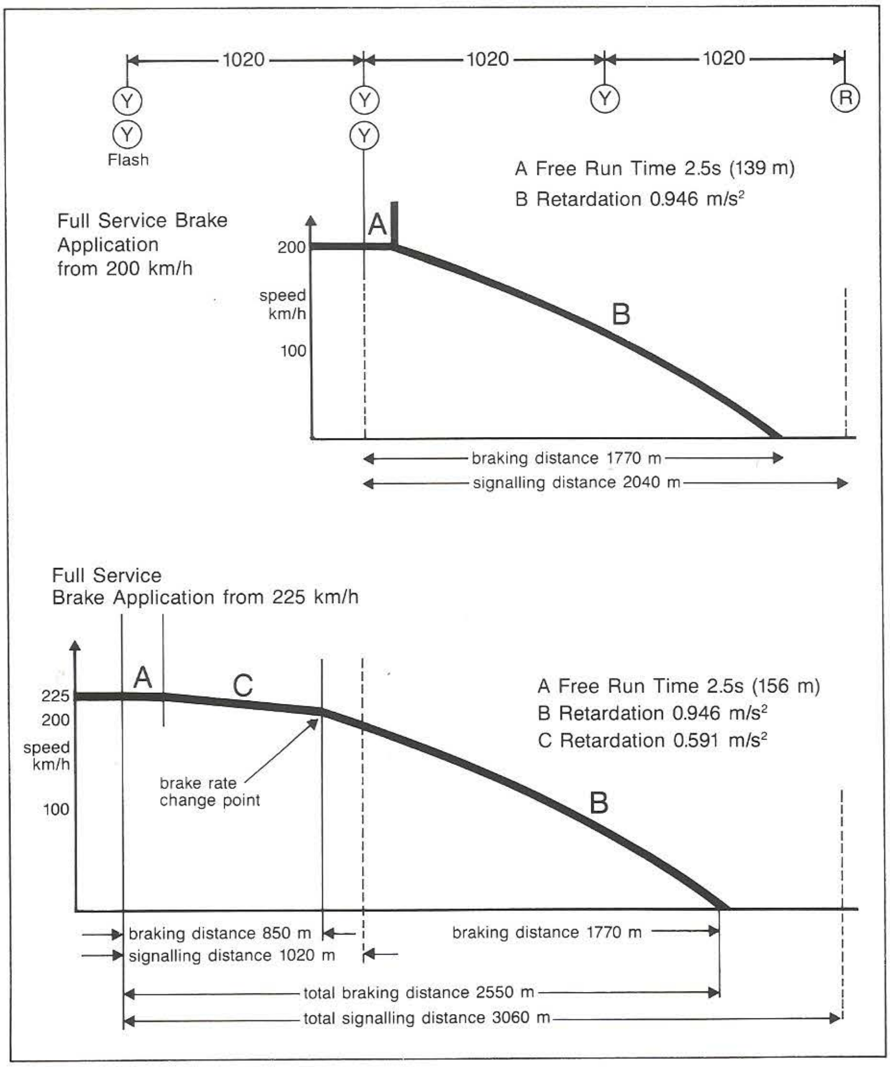
5. Brake systems

The main form of braking on the locomotive is dynamic which reduces mechanical maintenance when compared to friction braking. Rheostatic braking is used down to about 50 km/h when the friction brakes begin to blend in. Friction brakes are also used for the emergency duty if the rheostatic brake is not available.

To further reduce the bogie and unsprung masses, the friction brake system includes a disc brake which is mounted on the armature shaft at the rear of the traction motor.

The locomotive braking system has to provide two-stage braking, equivalent to 0.941 m/s^2 for 200 km/h, to stop with a margin within the existing signalling distances. An extra signalling aspect is utilised for speeds above 200 km/h, with a retardation from 240 to 200 km/h of 0.541 m/s^2. Fig 10.

Friction braking is achieved by a combination of disc brakes and wheel tread brakes acting together. The disc hub is connected to an extension of the armature shaft at the back of the traction motor at the opposite end to the cardan shaft drive. The actuators and calipers for the disc are mounted on the motor carcase. At 225 km/h the disc runs at 2140 rev/min and is therefore of a low-energy type to minimise aerodynamic loss. The pattern of the ventilating vanes has also been designed to reduce the level of emitted noise and to prevent a pure tone.



The wheel is subject to thermal stressing due to the energy input at the wheel tread from the friction brakes. A finite-elements analysis of the wheel was carried out to ensure that the stress levels arising from the combination of thermal and mechanical loads were acceptable. This led to the decision to use single-sided brake actuators to reduce the risk of martensite forming on the wheel tread, leading to thermal cracking.

Fig. 10 Braking performance.

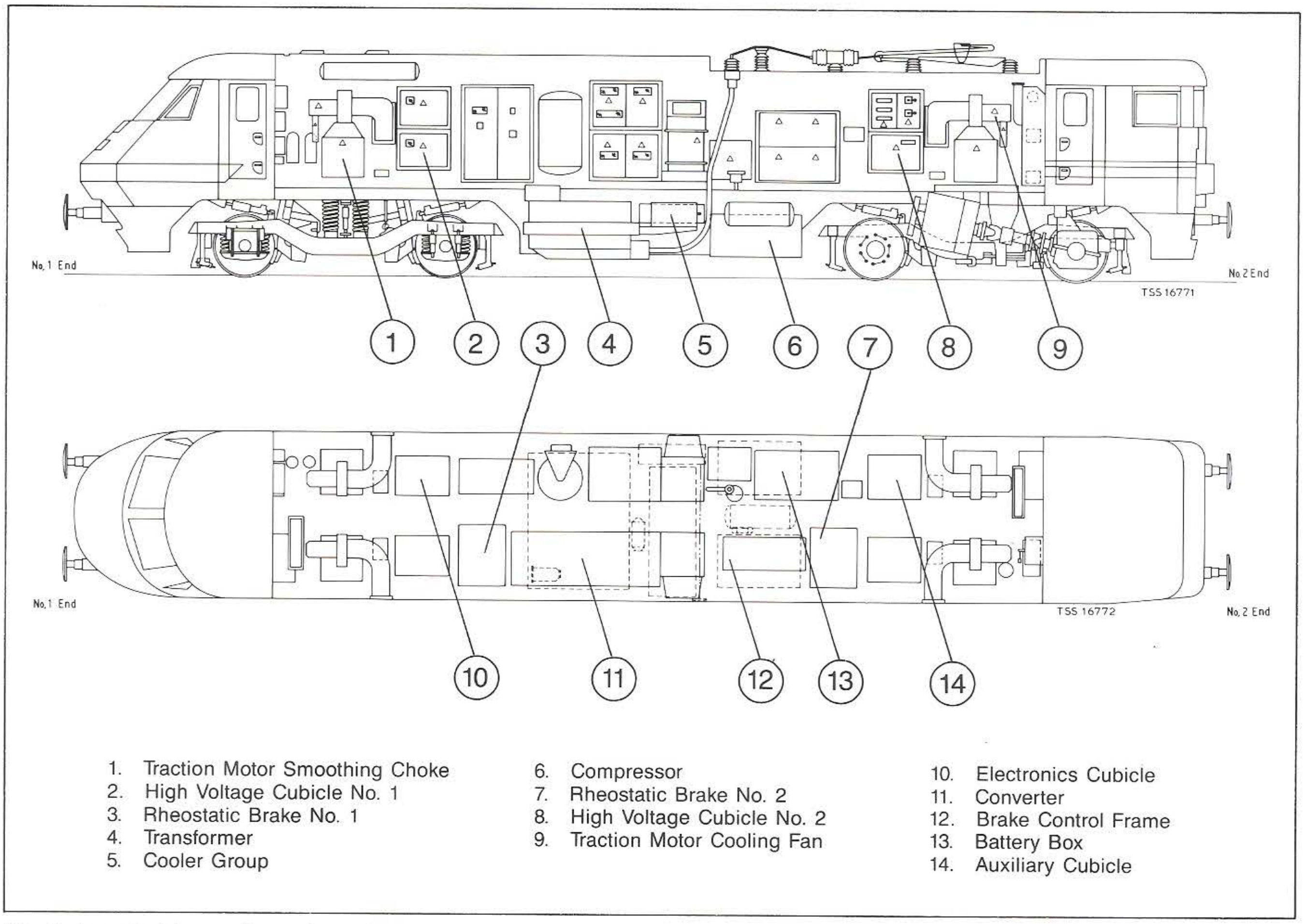


Fig. 11 Layout of locomotive equipment.

6. Power rating

The development of the 25kV locomotives on British Railways can be seen in the following table, all the locomotives being 80 tonne Bo-Bos designed for hauling both passenger trains and fast freight trains.

Class	Year	Speed	Continuous rating
81	1957	160 km/h	2.5 MW
86	1963	160 km/h	2.6 MW
87	1971	175 km/h	3.7 MW
91	1984	240 km/h	4.5 MW

7. The power equipment

The layout of the locomotive is shown in Fig 11 and the power circuit in Fig 12. The main transformer is mounted below the floor on four resilient mounts, and is integrated with its radiator and oil pump. Mounted above the transformer radiator is the heat exchanger for the oil cooled thyristor converter. Oil cooling Fig 13 allows the maximum output from large modern semiconductor devices and, in the Class 91, two asymmetrical series-connected bridges provide the supply to both motors in one bogie. The following table shows a comparison with synchronous and asynchronous locomotives and with earlier dc designs. It also illustrates the small number of devices used in a modern converter locomotive.

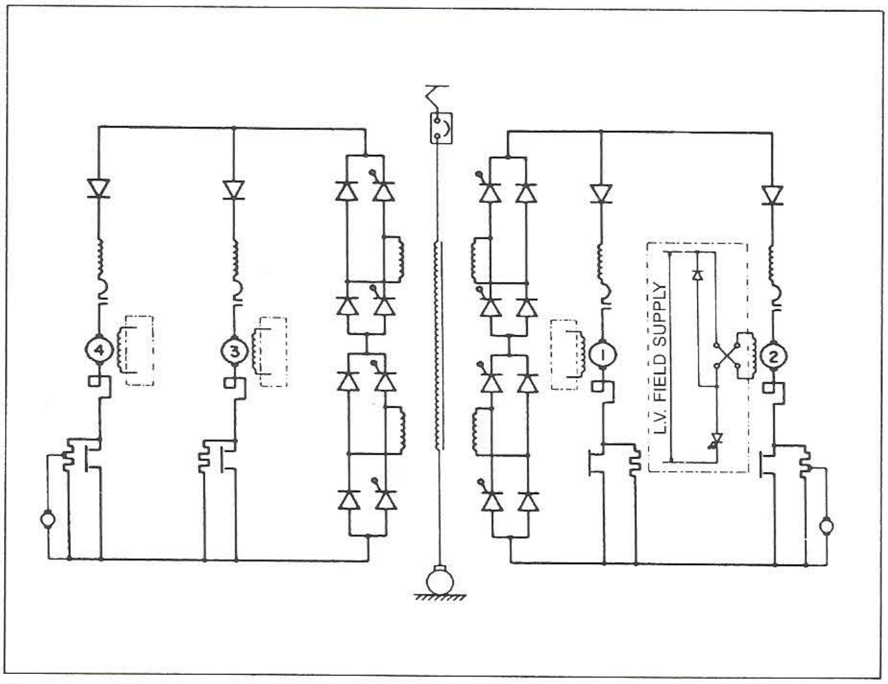


Fig. 12 Locomotive power circuit.

Locomotive	Drive	Year	No. of devices	Reference
SNCF BB15000	dc	1971	136	Ref 3
BR APT-T	dc	1974	40	Ref 4
Class 91	dc	1987	28	—
DB E120	async	1983	c450	Ref 5
SNCF BB10003	async	1983	c350	Ref 6
SNCF BB10004	sync	1982	132	Ref 7
SNCF BB26000	sync	1987	96	Ref 8

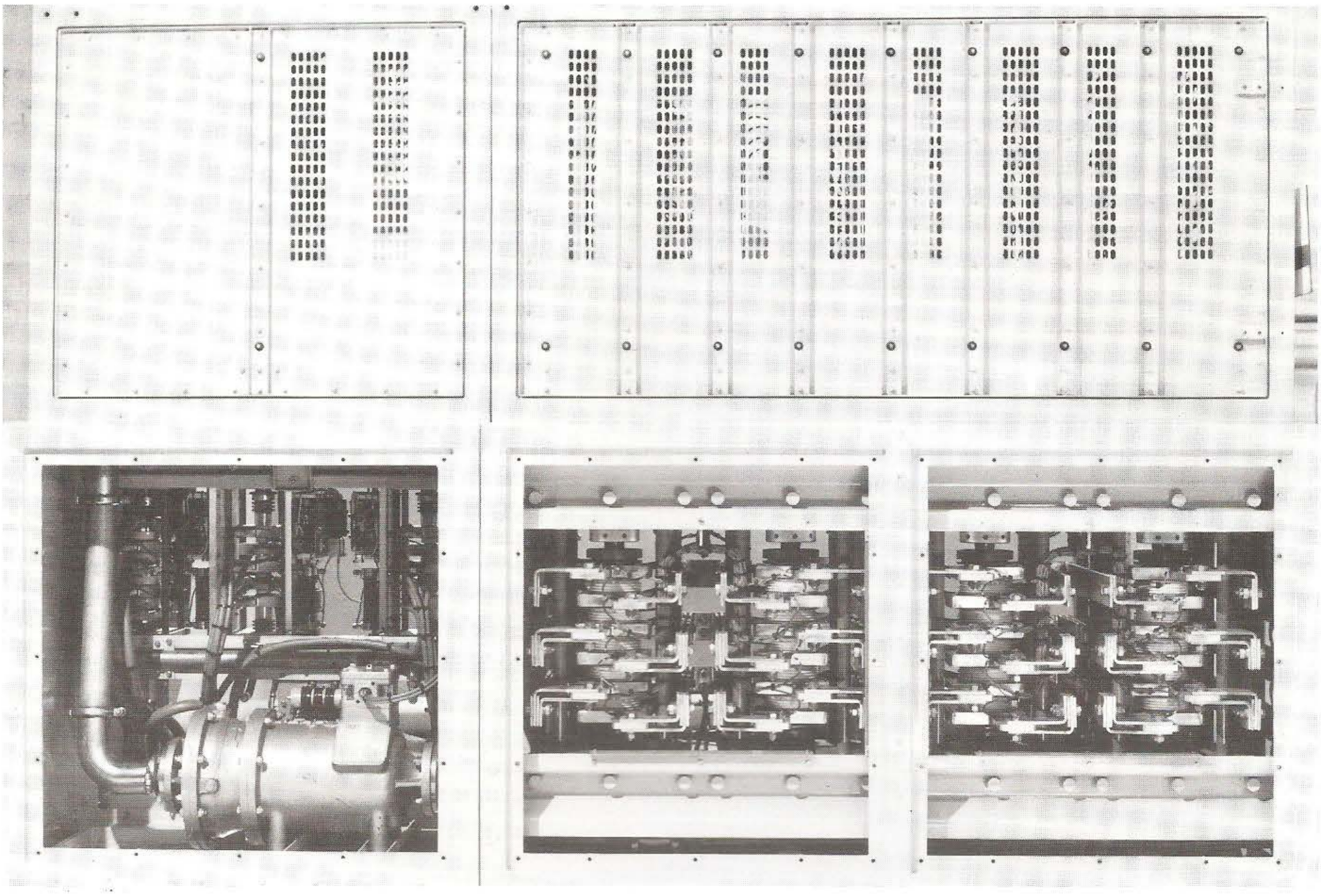
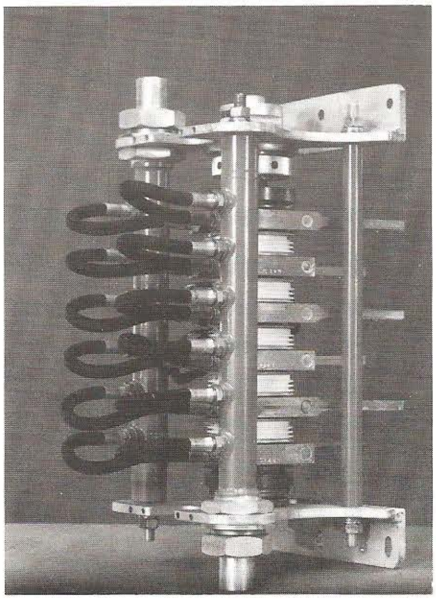


Fig. 13 Oil cooled thyristor converter assembly.



8. Control systems

The Class 91 locomotive, like all recent locomotives and EMUs manufactured by GEC, uses direct digital control. The microprocessor is an Intel 8086 16-bit device and provides control over the armature and field currents and the operation of the various electromechanical components. In addition it controls wheelspin and provides fault monitoring and self-diagnostic facilities.

A separate time division multiplex (TDM) system is used to receive control signals from the DVT, when the locomotive is being used for propelling, and to transmit indications to the driver.

The locomotive incorporates a sophisticated control system in which the speeds of the axles are compared with each other and with a signal from a doppler radar, supplied by GEC Sensors Ltd. Any mismatch in speeds or excessive acceleration causes a reduction in the power applied to that axle.

9. Harmonic currents and power factor

The power circuit design has been optimised to reduce harmonic currents and displacement factor levels. The reactance of each transformer secondary winding has been specified at a nominal 15% to provide harmonic reduction without significantly reducing the displacement factor.

In addition the firing angle of the two converter bridges is arranged to be different. This leads to an appreciable reduction in psophometric current (ie the sum of the harmonic currents weighted to correspond to the sensitivity of base-band telephone circuits) without infringing device safety factor or penalising the transformer design.

The microprocessor control system will select which of the two transformer secondary windings to associate with first bridge operation on each bogie, depending on the direction of travel. This has the effect of minimizing the psophometric current and making it independent of direction of travel.

The combination of these design techniques achieves a low level of harmonics without resorting to the use of harmonic filtration which would incur cost, weight and reliability penalties.

10. Project timescale

GEC TPL received the order for the Class 91 Locomotives in February 1986. The first locomotive was handed over to BR just 2 years later and within a week was running on test on the Main Line (Ref 9).

During 1988 the locomotives underwent a rigorous testing programme. The locomotive was first tested statically to show that the safety against derailment criteria were met and in addition static body-sway tests were carried out to provide information on pantograph sway.

The ride stability and track force tests have shown that the former are acceptable and confirmed that the bogie/transmission arrangement has given lateral track forces that are well within specification.

The wheelslip tests were carried out on track that had been treated to give artificially low adhesion. In commissioning the system the use of the radar was introduced to give a true measure of speed indication when all wheels are slipping. At this stage the software controlled drive system gave great flexibility to allow the optimum selection of parameters.

The contract allowed a period of 20 months from the delivery of the first locomotive to the entry into flat passenger service. The Class 91 first hauled a passenger train into Leeds in September 1988 and progress of development work is expected to allow the locomotives to be used in regular passenger service several months before the contractual date of October 1989.

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