TEST BENCH FOR TESTING OF THE TRACTION DRIVE OF RAILWAY VEHICLES

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This article gives an outline of the progress of work on the Study of UWB (University of West Bohemia) test bench for partly-suspended and fully-suspended variants of individual wheelset traction drives. First, a survey of existing designs of roller test benches was conducted. Its outcomes supported the idea of designing a completely new device. The main part of the article is devoted to the design of the test bench. Furthermore, the article briefly summarizes the progress and especially the partial results of the work on the task Computer simulation for optimization of the structural arrangement of the traction drive, which focuses on the development of a mathematical model of the aforementioned test bench.

KEYWORDS
test bench, traction drive, torque transmission, suspension, gearbox oil leakage, fully-suspended drive, computer simulation, optimization

1. INTRODUCTION

The manufacture, servicing and rebuilding of modern rolling stock involves in each case a range of tests which are conducted on the entire vehicle, its assemblies or even on separate components. In terms of testing of powered bogies, today's test stands can be classified as facilities for either static or dynamic testing. In this context, dynamic testing means that the rotational movement of wheelsets and the connected driving gear is applied by an external drive. By contrast, the primary purpose of a static test stand is to impose the full load on the bogie to enable determination of wheel forces and calculation of shim heights for each of the primary and secondary suspension springs.

2. STATE OF THE ART

The test stands which are used nowadays for testing entire powered bogies with full accessories are characterized by bridge configuration. It can be divided into a static and dynamic tests stand. In dynamic simulations of the bogie service, each wheel is driven by a separate electric motor via two rollers forming an endless rail. Consequently, these simulations do not qualify as load tests of the traction motor or the propulsion system as a whole. This understanding of the dynamic test is shared by all manufacturers of such stands. As a result, none of their products offers simulations of the real-life service of a traction drive and its components under full load, let alone their life testing. By the same token, no test stands offer simulations of running on an actual track where the effects of the track dynamics on both sprung and unsprung masses of the bogie need to be taken into account or where the risk of gearbox oil leakage can be monitored with regard to bogie tilting while the vehicle negotiates a curve.

2.1 Static test stands

The essential function of these stands is to subject the bogie to the full load which may arise in actual service. Depending on the bogie type, the load may be applied at one or two points. On Jacobs bogies, there may be up to four points of load application. The maximum load forces in these stands range from 2 × 250 kN to 2 × 450 kN but in the stands for three-axle bogies, they may go as high as 2 × 600 kN.

Static testing is understood as testing and measurement of those variables which are not necessarily associated with the bogie running. Such tasks comprise the following:

- Simulations of real-life load distribution on the bogie, and particularly on its frame;
- Load distribution between axles;
- Load distribution among individual wheels;
- Manual measurement of wheel diameter;
- Axle parallelism under load;
- Gauge;
- Primary suspension stiffness on individual wheels;
- Bogie frame top height above the rail top plane;
- Secondary suspension height above the rail top plane;
- Calculation of shim thickness for primary and secondary suspension;
- Tightness of secondary air suspension;
- Tightness of the air brake system.

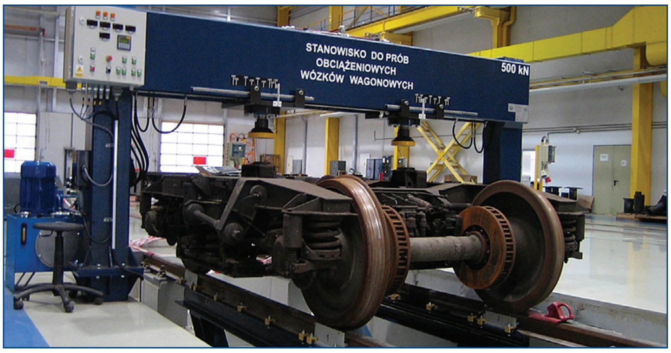

Figure 1. Static test stand from the company MTL Asco Rail [MTL Asco Rail 2012]

2.2 Dynamic test stands

As mentioned above, the stands for dynamic testing have separate drives for each bogie wheel. Torque transmission from the drive to the wheel is provided by two rollers forming an endless rail. When compared to static testing, this configuration offers the following additional capabilities:

- Automatic measurement of wheel diameter;
- Rotation test of the driving gear with or without vertical load;
- Application of lateral force;
- Guide wheel derailment test;
- Measurement of wheel tread profiles;
- Measurement of speed, temperature, vibration and noise;
- Measurement of bogie wheelbase and diagonal distance between axles;
- Measurement of bogie frame tilting angle;
- Measurement of wheel runout;

Without exception, today's advanced dynamic test stands can handle variable gauges and variable wheelbases. This holds for one of the best-

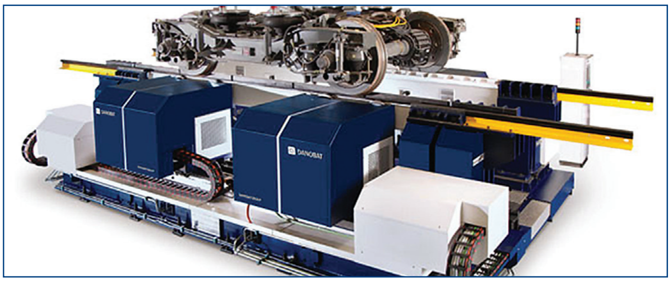

Figure 2. Danobat DTR-25 stand for dynamic testing [Railway Industry Equipment 2014]

equipped test stands, the Danobat DTR-25, which offers the advantage of testing bogies without disconnecting them from the vehicle. The entire plant is installed below the floor of the test hall. A vehicle can therefore stop on top of it and its bogies can be tested prior to any maintenance operation is started. Maintenance can thus be shortened and, where necessary, components which are still in adequate condition can be retained without servicing for a period longer than prescribed. Thanks to the manufacturer's collaboration with SKF, a bearing producer, this stand also offers advanced diagnostic technologies for simulations of running speeds above 100 km/h.

3. DEVELOPMENT OF TRACTION DRIVE TEST STAND DESIGN

The range of test stands available on the market today includes no equipment for testing traction drives and efficiency of power transmission from the traction motor to the wheelset. Equally lacking is equipment for modified life testing of traction drives by simulated track operation with all track effects encountered in real life. The purpose of the WP10 work package – which is carried out at the Regional Technological Institute (RTI) affiliated with the Faculty of Mechanical Engineering (FME) of the University of West Bohemia (UWB) – was to develop a design of a test bench for testing traction motor-gearbox-wheelset units while meeting these requirements. Before developing any test bench concepts, the operating conditions to be simulated on the bench and the required functions had to be identified. Their list is as follows:

- Simulations of the wheelset and the traction drive running on a real straight track;
- Simulations of the wheelset and the traction drive running on a real track with a horizontal curve and a cant up to 150 mm;
- Simulations of running under vertical load up to the maximum axle load;
- Simulations of a transverse load with a displacement of ± 10 mm;
- Loading the traction drive by the maximum torque anticipated in its real-life service;
- Excitation of the wheelset of the drive under test by forces simulating running on a real track;
- Simulations of vertical irregularities of the track;

Several changes to the test bench configuration were made during the design development. They are detailed in the final report RTI-VZ-2015-9. The final design is presented in the figures below.

It consists of a base frame isolated from its foundation, and of a main frame and a load frame, to which the traction drive under test is attached. The main frame can tilt sideways with respect to the base frame, thanks to two four-bar-linkage mechanisms and linear actuators. This arrangement is advantageous because it minimizes the deviation of the machine's centre of gravity from its central position, an occurrence

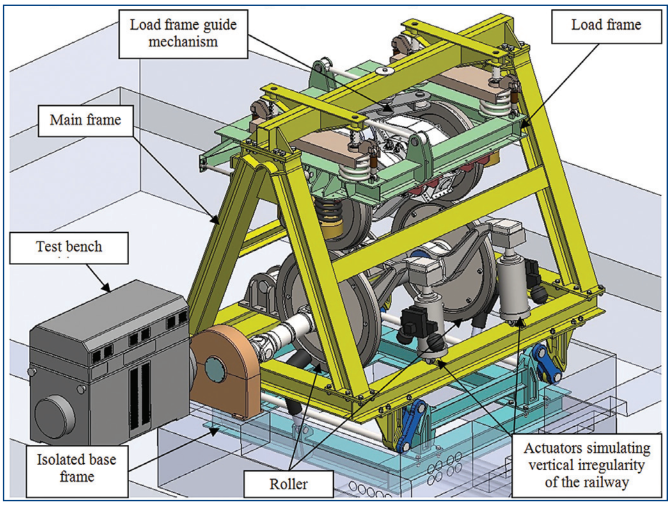

Figure 3. Test bench design configuration

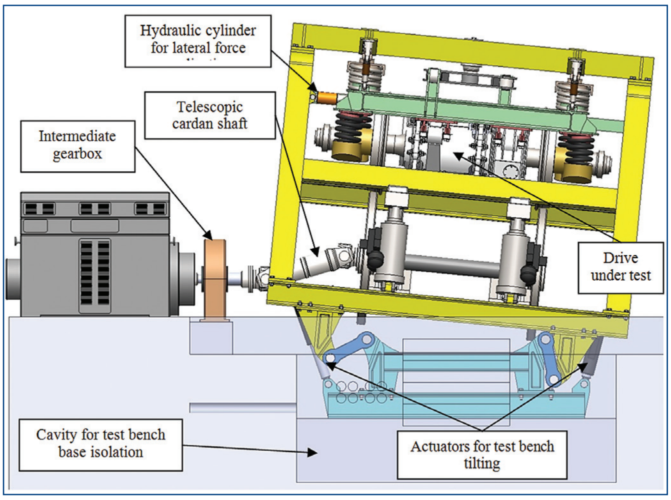

Figure 4. Front view of the bench in right-tilted end position

which is undesirable from the perspective of isolating the entire bench from the foundation. It contributes to safety as well, since the main frame returns from its tilted to its initial position spontaneously in the event of any failure. Attached to the main frame by two independently-excited rock arms is an axle with the rollers, the endless track. The load frame is designed as a modular assembly, to which the traction motor with the gearbox and the wheelset is attached by special adaptors. Its longitudinal position with respect to the main frame is controlled by an antiparallelogram linkage. Lateral forces can be applied to the wheelset guide mechanism by a hydraulic cylinder. By this means, forces induced by curve negotiation can be simulated. Vertical loads are imposed on the load frame by a system of springs representing the action of the secondary suspension. These springs are compressed by an assembly of ball screws by which the dead load of the carbody shell can be simulated.

The top cross beam of the main frame is removable to allow wheelsets to be lowered into the bench. Additional extension pieces can be placed between the main frame and the top cross beam to provide space for testing drives with wheelsets of diameters up to 1250 mm. At its proposed width, the main frame can accommodate drives for gauges of up to 1520 mm, as well as tram drives with traction motors on the outer sides of the wheelset.

Torque from the 500 kW test bench drive is transmitted to the rollers by an intermediate gearbox and a telescopic cardan shaft which enable sideways tilting of the frame. The wiring is proposed as two-circuit converter configuration which enables such connection between the driving and loading portions of the test bench, in which the power supply would only be required to compensate the power loss. The rollers consist of wheel rims bolted to steel discs press-fitted to the roller axle. Between these discs, the bearings of the rock arms are located. The wheel rims are therefore easy to substitute with special rims for wheelset excitation by surface irregularities or for alternative gauge testing [Trpak 2013], [Heller 2014].

3.1 Identification of structural details to be tested

The purpose of subjecting rail vehicle assemblies to conditions approaching those in the actual service is to ascertain the long-term impact of such loading on the functioning of the assembly. As the design of the test bench evolved, the specifications of its measurement and testing capabilities were defined and refined. Their list is as follows:

1) Performance of the rpm sensor of the traction motor under real service conditions and at real acceleration levels.
2) Performance of temperature sensors of the traction motor under real service conditions and at real acceleration levels.
3) Strain gauge measurement of stresses in traction motor mounting lugs and in other critical locations of the motor to be defined later.
4) Temperatures of bearing assemblies of the traction motor in a reference load schedule. The temperature data will be used for defining lubrication schedules for plastic grease.

5) Inspection of the condition of bearing assemblies (plastic grease condition) of the traction motor upon modified life test.
6) Pressure monitoring within the motor and the gear case in a reference operating schedule with respect to potential overpressure in the motor-gearbox system in case of the single motor-gearbox unit. The motor and the gear case will be provided with hose ports to permit this measurement.
7) Oil leak monitoring through the shared labyrinth seal of the motor-gearbox unit. Oil entry into the motor will be checked by means of an endoscope.
8) Testing the condition of gear couplings or other torque transmission mechanisms where there is flexible coupling between the motor and the gearbox. After completion of the modified life test, the coupling or other equivalent mechanisms will be sent to the manufacturer for gear condition assessment and overall inspection.
9) Temperatures of bearing assemblies of the gearbox in maximum torque and reference load schedules.
10) Strain gauge measurement of stresses in traction motor mounting lugs and in other critical locations of the gearbox to be defined later.
11) Assessment of gearbox oil condition upon the modified life test.
12) Assessment of gear condition in the gearbox upon the modified life test.
13) Assessment of the quill condition, if used.
14) Measurement of axle bearing temperature in maximum torque and reference load schedules.
15) Performance of the rpm sensor of the axle bearing under real service conditions and at real-world acceleration levels.
16) Tests of components for attachment of gear case to the bogie frame (hanger, silentblocks and others).

The key to accurate simulations of wheelset and traction drive operation is in acquiring excitation frequencies from real tracks for relevant vehicle types for whose bogies the drive is to be tested. The track dynamics must be taken into account with respect to tram vehicles, as well as sub-urban rail vehicles, vehicles for long-distance routes, and others [Dupal 2013].

4. COMPUTATIONAL SIMULATION OF DYNAMIC BEHAVIOUR OF UWB TEST BENCH

In the first stage of this project, the project team at the FME of Czech Technical University (CTU) was assigned an analytical sub-task for 2012 and 2013. This task involved developing a dynamic model of the UWB roller test bench and completing an initial mathematical simulation-based analysis of the forces imposed by the roller test bench on the foundation of the testing laboratory building of the Regional Technological Institute of the University of West Bohemia.

The initial approach to this sub-task and the results of simulation calculations for dynamic analysis of forces on the building foundation resulting from the fully-suspended drive variant have been presented in two research reports [Kolar 2012] and a conference paper [Kolar 2013]. The later approach to this sub-task and the results of simulation calculations for the dynamic analysis of forces on the building foundation resulting from the fully-suspended drive variant have been presented in two research reports [Kolar 2013] and [Fridrichovsky 2013].

The focus was on the effects of vertical axial stiffness of the isolation below the test bench foundation block, which consisted of a vibration-isolation rubber plate of 8×5×0.039 m size (shown in purple), and another rubber plate of 3×3×0.024 m size (shown in orange colour) under the test bench concrete base slab, as illustrated in Fig. 5.

The graph below shows a plot of dynamic stiffness of the vibration-isolation plate of 3000×3000×24 mm size (the orange plate) placed under the base slab in Fig. 5.

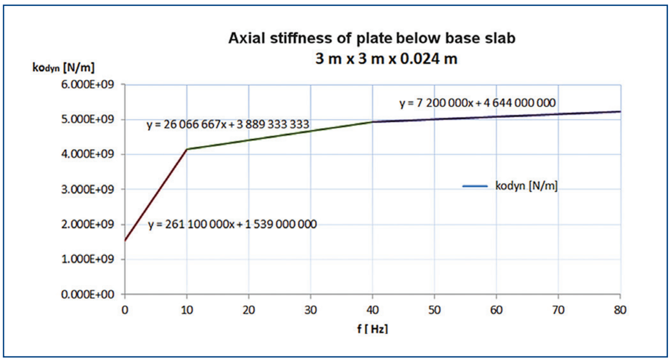

Figure 6. Axial stiffness of plate below base slab

The various profiles obtained by calculation with identical basic input values into the model are presented in Fig. 7. The profiles denoted by the subscript "**d**" in the name of the variable, e.g. $\mathbf{F}_{ZBd5mm}$, represent real-life scenarios where the axial stiffness of vibration-isolation plates reported by manufacturers varies with the excitation frequency. The profiles where there is no subscript "**d**" in the name of the variable, e.g. $\mathbf{F}_{ZB5mm}$, represent scenarios in which the axial stiffnesses ($\mathbf{k}_{\Sigma y}$, $\Sigma \mathbf{k}_y$) of vibration-isolation plates are constant and thus independent from the excitation frequency.

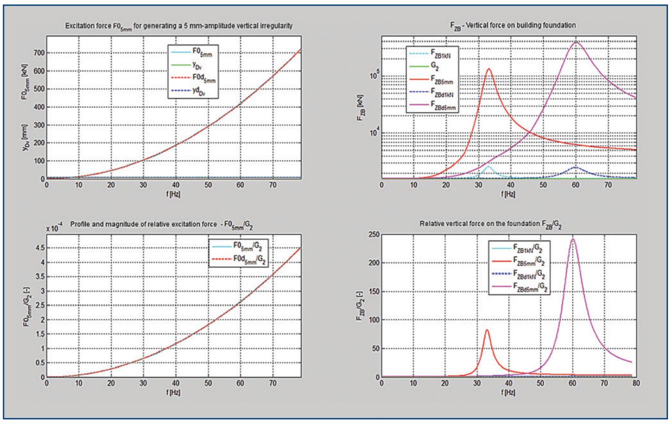

Figure 7. Plots of calculated characteristics based on fundamental parameters of axial stiffness of vibration-isolation plates

Fig. 7 suggests that with increasing dynamic axial stiffness of the vibration-isolation plates, the natural frequency of the base slab increases as well. When static stiffness values are considered, the natural frequency "f" is approximately 33 Hz. With increasing axial stiffness according to characteristics illustrated in Fig. 5, the natural frequency of the base slab rises to approximately 60 Hz, and the resulting maximum vertical force on the building foundation goes up from approximately $1.4 \cdot 10^5$ kN = 140 MN to approximately $3 \cdot 10^5$ kN = 300 MN. The basic weight load on the building foundation is approximately 1.6 MN.

Dynamic effects calculated with static axial stiffnesses of the vibration-isolation plates will bring the load to a level about 80 times higher than the static load. However, when the actual "dynamic" axial stiffnesses of

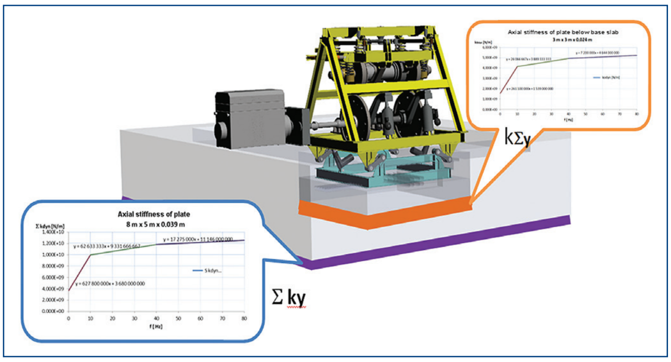

Figure 5. Schematic illustration of test bench foundation

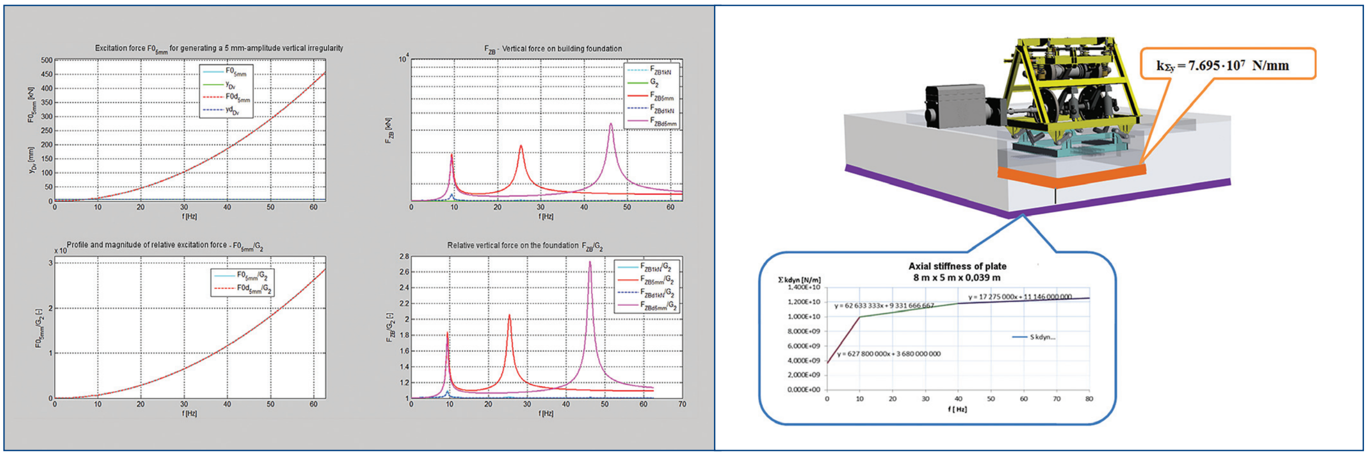

Figure 8. Values calculated with optimized axial stiffnesses of vibration-isolation plates

the vibration-isolation plates are used for the calculation, the load rises to 245 times the static level. **Forces of this magnitude arising at the excitation frequency of 60 Hz are unacceptable. Therefore, the stiffness of the base slab isolation must be reduced dramatically.**

Methods of reducing the stiffness of the isolation of the base slab were thus proposed and analysed. One of them involved the use of multiple-layer vibration-isolation rubber plate of the 3×3×0.24 m size (indicated by orange colour) and another employed air suspension. Effects of reducing the stiffness $\mathbf{k}_{\Sigma y}$ were thus explored. As calculations showed, the dynamic effects of the roller test bench on the building foundation can only be effectively suppressed if the vertical stiffness $\mathbf{k}_{\Sigma y}$ of the isolation of the base slab of the 3×3×0.6 m size is reduced to 5 % of its above-mentioned basic value. This means that the axial stiffness should be approximately $\mathbf{k}_{\Sigma y}$ = 7500 kN/mm and should remain constant, regardless of the excitation frequency, as illustrated in Fig. 8. To accomplish this, the base slab can be mounted either on air springs or, in a more cost-effective variant, on appropriately distributed coil springs. In both isolation variants for the 3×3×0.6 m concrete base slab, it would be advisable to add hydraulic dampers (the resulting damping constant should correspond to 10–15 % of the damping ratio). As a result, the forces induced by passage through resonance regions should drop because the preceding calculations were based on a damping ratio of approx. 2 % which was the estimate for the vibration-isolation plates. With this isolation for the concrete base slab of 3×3×0.6 m size, it would be possible to retain the previously suggested isolation of the foundation block of the 8×5 m footprint with 8×5×0.039 m plates.

At their meetings, the research team agreed that mathematical models of the UWB roller test bench would be developed gradually to simulate the effects of the powered wheelset running on a track with vertical irregularities and to monitor the impact on the torsional dynamics of:

a) Fully-suspended drive (single wheelset drive effected by an articulated quill around the axle);
b) Partly-suspended drive provided with an axle gearbox.

Development of the mathematical model of the test bench and the simulation of the fully-suspended drive will be the responsibility of the University of West Bohemia. The development of the mathematical model of the partly-suspended drive took place at the FME CTU in 2014 and 2015. It was part of the task entitled Computer simulation for optimization of the structural arrangement of the traction drive. The initial theoretical mathematical model of vertical dynamics of the UWB roller test bench was thus supplemented with equations of motion describing the torsional dynamics of the partly-suspended traction drive. The movement of the individual bodies which make up the roller test bench and the traction drive was analyzed and relevant equations of motion have been derived. Then the basic version of the computational program was developed and debugged in the Matlab environment. Detailed findings from these initial calculations have been published in the research report [Dub 2015]. In 2016, the computations and further efforts to improve the accuracy of the computational model of the partly-suspended drive will be continued.

5. CONCLUSION

Despite the abundance of designs of traction drives in today's rail vehicles, the following conclusions can be drawn regarding their design optimization [Kolar 2015]:

– **Low-floor tram vehicles** should operate in the following configurations:

a) Traction drives in which a longitudinally oriented traction motor attached to the outer side of the bogie frame drives fully or partly suspended single-wheel gearboxes for freely rotating wheels on portal axles, or fully or partly suspended axle gearboxes of tramway wheelsets with wheels of smaller diameters. From the weight perspective, these solutions with two traction motors per two-axle bogie are more favourable than four lower-power traction motors with suspended single-wheel gearboxes.

b) Where freely rotating wheels must be driven by an individual drive using synchronous motors with permanent magnets, the variant of a fully-suspended drive of freely rotating wheels is the appropriate choice. This configuration has a slight disadvantage of the large number of frequency converters-inverters, as each motor must be fed from its own converter.

– **Metro and sub-urban rail vehicles** will benefit from drive solutions which reflect the required traction power of the powered wheelset and the maximum operating speeds:

a) Metro vehicles operated at speeds V ≤ 80 km/h should use an economical and simple design involving a partly-suspended drive with a transverse asynchronous traction motor attached to the bogie frame. The output torque is then transmitted by an articulated shaft with two spherical couplings to the pinion of a partly-suspended compact axle-gearbox, i.e. a gearbox whose output shaft bearings are mounted to the large gear hub.

b) In electrical multiple units operated at speeds of V ≥ 120 km/h, the desirable variant is the dynamically more favourable fully-suspended transverse drive, in which the traction motor is integrated with a compact spur gearbox into a drive block suspended to the bogie frame. The driving torque from the hollow output shaft of the spur gearbox is transmitted to the wheelset via a quill around the axle.

– **In high-speed electrical units,** the appropriate traction drive configuration for the powered wheelset is represented by a simple partly-suspended design. A transverse synchronous traction motor is suspended to the bogie frame and drives the pinion of a partly-suspended compact three-wheel axle gearbox via an articulated shaft with two spherical gear couplings. The use of the three-wheel axle gearbox is favourable for dynamic performance, as it generates less force on the gearbox hanger than a two-wheel axle gearbox while transmitting the same driving power to the wheelset.

– **Electrical locomotives** will benefit from drive solutions which reflect the required traction power of the powered wheelset and the maximum operating speeds:

a) In shunting locomotives and freight locomotives, the suitable traction drive configuration consists of a simple design: an asynchronous nose-suspended motor with a pinion mounted in the axle gearbox case. The case provides one of the points of attachment to the axle. An overhung pinion in the spur gearing is not a suitable solution, as it leads to less favourable engagement conditions and typically necessitates longitudinal modification.

b) In locomotives operated at between 120 and 180 km/h, a simpler design with a partly-suspended drive is a favourable solution. A transverse asynchronous or synchronous traction motor is mounted on the bogie frame and torque transmission to the gearbox is provided by an articulate shaft. This articulate shaft comprises either two spherical gear couplings or disc joints, runs through a hollow pinion and drives either a two-wheel or a three-wheel compact axle gearbox, where the latter offers better dynamic performance at higher cost, while provided with a vertical or inclined hanger. With respect to reducing the dynamic effects on the torsional system of the drive, tangential suspension of the large gear wheel rim can be recommended.

c) The appropriate solution for high-speed locomotives, i.e. those operated at speeds between 190 and 230 km/h is a fully-suspended drive with a quill around the axle, as this configuration attenuates dynamic loads on the vehicle and the track. A theoretical analysis of mathematical models, which has been performed, showed that it is less sensitive to self-exciting torsional oscillations in the wheelset drive.

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