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Facility for performance testing of power transmission units

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Abstract

This paper describes the engineering design for a facility to test the performance of gearboxes and drive axles. Power transmission units are much more frequently tested on test rigs than in motor vehicles. The rig testing of such devices constitutes a very complex area because of the diversity of functions performed by individual components and their parts in the power transmission systems of motor vehicles and construction machinery. In the simulation tests carried out on simulation test rigs, the conditions of testing of individual units should match the expected conditions of operation for such units as much as possible. The test rigs used for this purpose are very complicated and expensive, but the results of rig tests are more reliable and accurate than they would be if other test methods were employed. The rig tests described here reflect the impact of anticipated service loads on the endurance of the unit under test.

Introduction

Vehicle testing may be classified in different ways, depending on the criteria adopted (Lanzendoerfer, 1977; Minchejmer, 1962; Orzełowski, 1995; Sitek & Syta, 2011). In terms of the object of testing, the following categories can be discerned:

1) testing of a complete vehicle;
2) testing of component units;
3) testing of individual parts.

As regards power transmission units, they are much more often tested on test rigs than in motor vehicles. Such tests can be divided into the following groups:

– performance tests, which assess the correctness of functioning of the unit;
– endurance tests, which estimate the service life of the unit and describe fatigue phenomena;
– ultimate strength tests, designed to identify the weakest parts in the unit, in order to determine the strength margin factor, and to assess the deformation of individual parts of the unit as a function of the load.

Due to the great diversity of functions performed by individual component units and their parts in a power transmission system, the testing of such devices constitutes a very complex activity. The laboratory tests of endurance for such units are carried out on specially prepared test rigs, using various test rig programming methods.

The rigs used for such tests are of two types:

– specialized test rigs for performance tests;
– simulation test rigs for the testing of dynamic characteristics, endurance testing, and optimizing testing of power transmission systems.

The performance tests of gearboxes, as carried out at our laboratory, include the checking of such characteristics as:

– maximum stabilized oil temperature;
– no-load mechanical drag;
– no-load drag power;
– static gear-change forces;
– gear engagement and synchronization times and synchronization forces;
– total angular backlash;
– noise.

The performance tests of drive axles, as carried out at our laboratory, includes the checking of such characteristics as:
– maximum stabilized oil temperature;
– no-load mechanical drag;
– no-load drag power;
– frictional drag in the differential gear;
– backlash / angular backlash in the final drive;
– noise.

In order to carry out of such tests, adequate research potential must be available to enable the preparation of an appropriate test rig. This paper addresses the design of a facility capable of testing the performance of power transmission units.

Performance testing of power transmission units

Maximum stabilized oil temperature

This is one of the tests that are most often carried out when examining the performance of mechanical gearboxes and drive axles. Gearbox mechanisms should not get too hot during operation. The maximum stabilized oil temperature is a term of art that entails the following conditions:
– no load is put on the output shaft;
– the input shaft speed, n_{ws}, is $0.75 \cdot n_{max}$, where n_{max} is the maximum speed of the vehicle propulsion motor;
– the gear in which the vehicle reaches its maximum speed is engaged;
– no extra cooling is used.

Under these conditions, and depending on the nominal input torque of the gearbox, the following conditions should obtain:
– for nominal torque of up to 250 Nm, the maximum oil temperature should not exceed 85°C;
– for nominal torque over 250 Nm, the maximum oil temperature should not exceed 100°C.

The gearbox oil temperature (Industrial Standard, 1976; PIMOT Technical Specifications, 1997) should be checked on a test rig that would make it possible to install the gearbox on it and to achieve the test parameters required. During the test, the oil temperature should be recorded continuously or at 30 min intervals. The measurement of the stabilized temperature may be considered complete when the oil temperature as indicated by the measuring instrument does not change by more than 2°C over a period of 60 min. The rotational speed of the gearbox input shaft should be kept within a tolerance range of ±50 min^{-1}.

Figure 1 shows an example of oil temperature vs. time curves measured for a 5-speed gearbox of a passenger car. In the case presented, the measurements were carried out in succession in each gear, with the gearbox being cooled to the ambient temperature after each measurement.

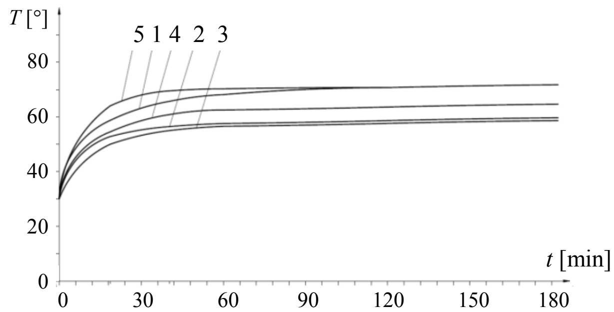

Figure 1. Oil temperature vs. time curves measured in individual gears for a 5-speed gearbox of a passenger car. Input shaft speed of $n_{ws} = 0.75 \cdot n_{max} = 3900$ min^{-1}; 1, 2, 3, 4, and 5 – numbers of individual gears

For drive axles (PIMOT Technical Specifications, 1996), the oil temperature (when stabilized) should not exceed 100°C. The temperature is measured at a rotational speed n_{we} of the drive axle input shaft equal to 0.75 of the n_{max} value corresponding to the maximum power of the vehicle engine, with no output load being applied to the drive axle, at an ambient temperature of 20°C ±5°C, and with no extra cooling. A higher oil temperature limit is allowed in the drive axles, where the oils used do not lose their lubricating properties at higher temperatures, and the axle remains leak-proof. The oil temperature in a drive axle should be checked on a test rig, in compliance with the test requirements, with the time history of the temperature recorded until it is stabilized. An example of the stabilized oil temperature vs. input shaft speed curve, plotted for measurements carried out at speeds ranging from 500 min^{-1} to 3 000 min^{-1} at 500 min^{-1} intervals, is presented in Figure 2.

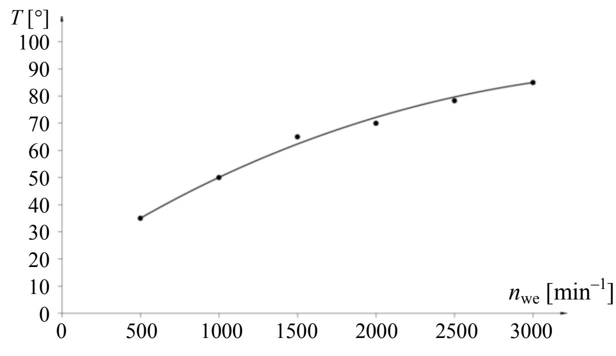

Figure 2. Stabilized oil temperature vs. input shaft speed curve, plotted for a drive axle designed for a bus. Constant input shaft speeds of n_{we} = 500, 1000, 1500, 2000, 2500, and 3000 min^{-1}

No-load mechanical drag and its power

The "manual" rotating of the gearbox shafts, with either the individual gears being engaged or in neutral, should be done with constant drag torque values appropriate for specific gears, without perceptible jams. The same requirement shall apply to the drive axle with its input shaft being rotated manually. The internal resistance (drag) torque is the sum of friction losses on teeth surfaces, in bearings, and on seals, and churning loss. This torque is usually measured on a test rig, where the gearbox is driven by an electric motor. The electric motor is so installed that its housing is free to turn and the housing reaction moment can be measured by means of a balance, a dynamometer, or a force measuring channel of adequate accuracy. During the measurements, no load is applied to the gearbox output shaft. The motor reaction moment is equal to the drag torque of the device under test (at a specific, constant, rotational speed). The measurements are carried out in different gears, at different predefined input shaft speeds.

The internal resistance to motion (drag) of a drive axle is measured in a similar way. The drive axle input shaft (final drive pinion) is driven by an electric motor, with no load applied to the axle shafts. The drag torque of the drive axle differential is measured almost identically; the only difference being that the left and right axle shafts of the drive axle are alternately immobilized. This alternate immobilization allows measurement of the drag torque necessary to stop one of the axle shafts while the other one is free to rotate as a function of the rotational speed of the final drive pinion. Instead of a balance system with the motor housing being free to turn, a torque meter installed between the motor and the device under test may be used to measure the drag torque. When the drag torque value at a specific speed of the input shaft of the device under test is known (from measurements), the power lost in overcoming the drag of the device can be calculated. The said power loss, determined for the input shaft speed equal to 0.75 of the speed corresponding to the maximum power capacity of the vehicle engine, should not exceed 2% of the maximum power of the vehicle engine. The power lost in overcoming the drag should be checked on a test rig at input shaft speeds ranging from 1,000 min^{-1} to the speed corresponding to the maximum vehicle engine speed, with the axle shafts rotating freely and the oil temperature remaining within the limits of 60−65°C.

Figures 3, 4, 5, 6 and 7 show example drag vs. input shaft speed curves plotted for individual gears of a 5-speed gearbox of a passenger car for input

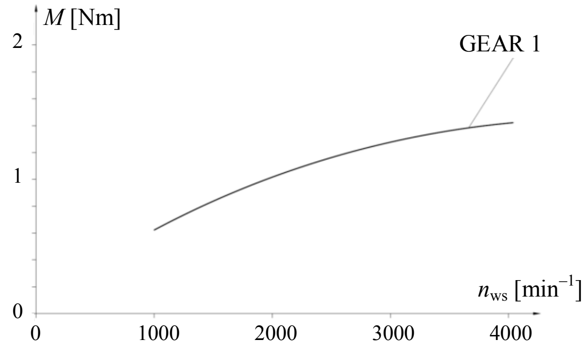

Figure 3. Drag torque vs. input shaft speed curve for a passenger car gearbox (GEAR 1)

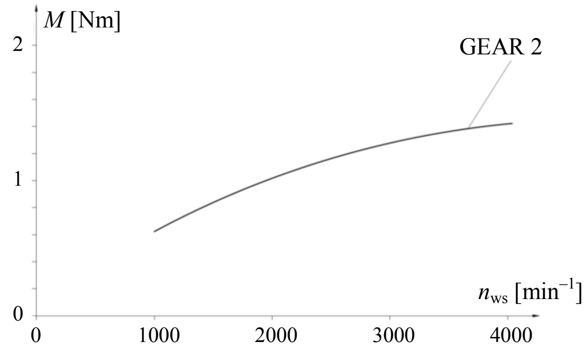

Figure 4. Drag torque vs. input shaft speed curve for a passenger car gearbox (GEAR 2)

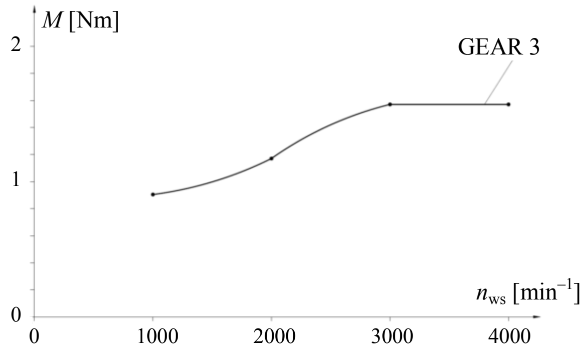

Figure 5. Drag torque vs. input shaft speed curve for a passenger car gearbox (GEAR 3)

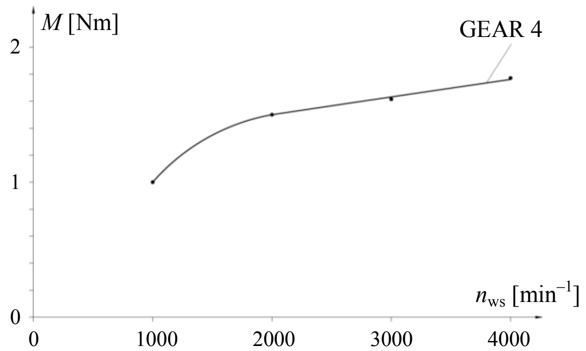

Figure 6. Drag torque vs. input shaft speed curve for a passenger car gearbox (GEAR 4)

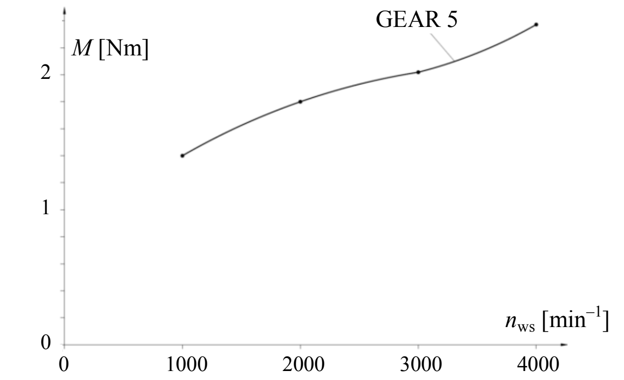

Figure 7. Drag torque vs. input shaft speed curve for a passenger car gearbox (GEAR 5)

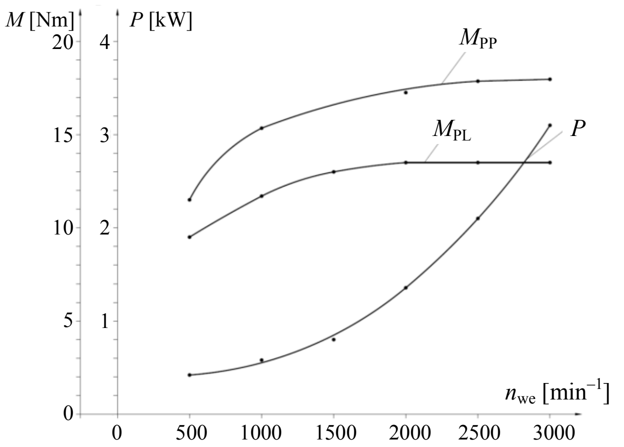

Figure 8. Curves representing the no-load drag power and drag torque of the differential of a bus drive axle: *P* − no-load drag power at both axle shafts rotating with no load; M_{PP} − differential drag torque at the right axle shaft immobilized; M_{PL} − differential drag torque at the left axle shaft immobilized; the results shown refer to the final drive input shaft; *P* − power; *M* − torque; n_{we} − speed of the final drive input shaft

shaft speeds ranging from 1,000 min^{-1} to $n_{ws} = 0.75 \cdot n_{max} = 3\ 900$ min^{-1}. The measurements were carried out at with oil temperature being within the range of 60−65°C. Figure 8 shows example curves representing the power of the no-load mechanical drag, and the drag torque of the differential of a bus drive axle as functions of the speed of the drive axle input shaft. The drag torque values were measured on the drive axle input shaft at rotational speeds ranging from 500 min^{-1} to 3,000 min^{-1}.

Functioning of gearbox control systems

The functioning of the gearbox control system is checked on a test rig, both when the gearbox is at not moving (to check the forces needed for static gear changing) and when it is driven (to check synchromesh functioning). In synchromesh gearboxes, the synchromesh should make it possible to change gears smoothly and quietly, with no audible grinding. The values of the forces needed for static gear changing in gearboxes remotely controlled should be measured on the external component of the internal gearbox control system, and they should not exceed the following upper limits:

– 300 N for passenger cars;
– 900 N for motor trucks and buses.

For gearboxes that are directly controlled, the limits on the forces measured on the gear-change lever knob are as specified below:

– 50 N for passenger cars;
– 150 N for motor trucks and buses.

The tests are carried out on a test rig making it possible to check the gearboxes for conformity with the above requirements. In most cases, the measurements are performed with strain gauge systems. The recording of the forces as functions of displacement or time offers a possibility of determining performance curves that characterize the functioning of gearbox control systems and indicate any jamming, seizure and other system operation problems. It is recommended that a specific gear be engaged several times before the measurement is made so that the gearbox elements that are to be connected with each other are brought into proper alignment.

An example curve representing the gear-change force, obtained as a result of a test carried out at our laboratory, is shown in Figure 9.

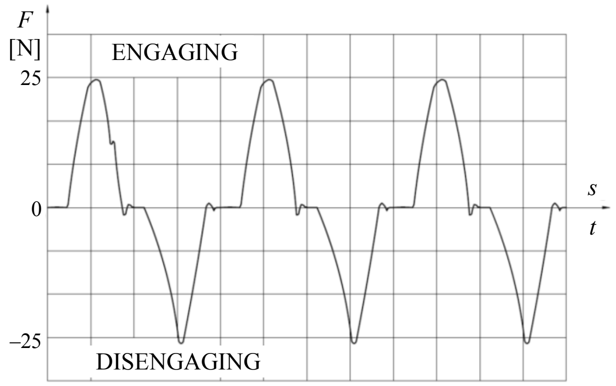

Figure 9. Example curve representing the gear-change force recorded when third gear was statically engaged in a passenger car, obtained in result of a test carried out at our laboratory

Synchromesh functioning

The functioning of the entire gearbox control system, inclusive of the synchromesh, is checked in a way similar to the checking of static gear changing, with the difference that the gearbox is then driven from the output shaft end at such a speed that the input shaft speed before the gear change conforms to the speed at which the gear is most often changed during normal vehicle operation. The rotational speed values at which the gears are changed are determined in road tests. If the actual operating data are unavailable, then the speeds may be calculated from vehicle manufacturer's instructions where the recommended gear-change speeds are usually specified. Otherwise, a rough assumption may be made that the input shaft speed should be 0.75 of the maximum engine speed for spark ignition engines, or 0.8 to 0.9 of the governed engine speed for diesel engines. A cylindrical inertial element with its moment of inertia being equal to that of the clutch disk is installed on the gearbox input shaft. Initially, preliminary trials are carried out with the gearbox being manually controlled to check whether the synchromesh units make it possible to change gears smoothly and quietly (with no "grinding"). Then, graphs representing the operation of the gearbox control system during the synchronization process are plotted with the use of measuring systems.

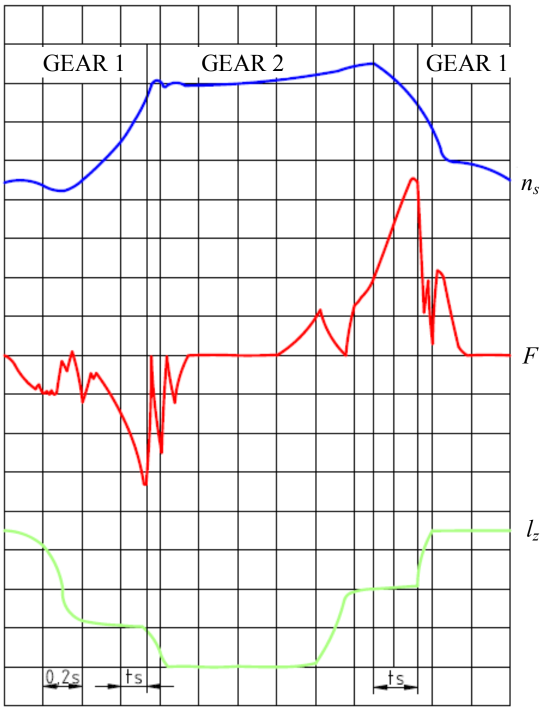

Figure 10. Example of curves representing the input shaft speed n_s, gear-change force F, and gear lever displacement, recorded during tests at our laboratory: t_s − synchronization time; the calibration scales for individual quantities have been omitted

To obtain comparable test results, which would enable reliable assessment of the synchromesh functioning, similar tests are carried out with the use of an automatic control system. In most cases, an electronically controlled electro-pneumatic valve controlling the feeding of compressed air to a gear-change lever actuator, is used for this purpose. Because air pressure can be adjusted, appropriate gear-change force can be obtained. At a preset rotational speed and with a repeatable gear-change force, the synchronization time is taken as a criterion for assessing synchromesh functioning. The synchronization time can be determined from the force and displacement curves recorded with the use of measuring systems, which include force and displacement sensors installed in the gearbox control system. An example of such curves recorded during tests actually carried out is presented in Figure 10.

Backlash in gear transmission units (gearboxes and drive axles)

In a pair of engaging gears, the width of a tooth space must slightly exceed the thickness of the mating tooth so that the latter can easily enter into the space. The backlash thus formed must be sufficient to protect the pair of gears from effects of imperfections in workmanship and thermal expansion, and to facilitate oil flow. Incorrect backlash may cause overheating of the gear transmission unit, increased noise level, and incorrect wear of tooth flanks.

As a measure of backlash, a parameter referred to as "angular backlash" is used. Based on the measured angular backlash value, the actual backlash between the flanks of the engaging gear teeth can be determined indirectly, if the radius from the axis of rotation of the gear wheel is known. Correct values of the backlash and radii may be specified by the manufacturer in the documentation of the transmission unit. For gearboxes, total backlash in individual gears are usually specified.

In gearboxes, the angular backlash values are most often measured with an optical protractor, fixed on the input (clutch) shaft, when the shaft is loaded with a pressing-down torque of a prescribed value. The measurements are carried out with the input shaft being loaded with the torque in both directions in succession. Sometimes the backlash has to be measured with the use of a dial gauge on the gearbox output shaft with the input shaft being immobilized, according to manufacturer's instructions given in the gearbox documentation.

For drive axles, the angular backlash values are determined in a manner similar to that used on gearboxes, with the measurements being carried out for different positions of gear wheels in the final drive and differential in relation to each other. During the measurements, both axle shafts are immobilized.

Depending on requirements, the backlash values may be measured on brand new test specimens as well as units that have experienced specific periods of endurance tests. For this reason, the backlash may be considered a tooth wear indicator.

Examples of backlash measurement results obtained from laboratory tests of passenger car gearboxes and drive axles intended for a delivery motor vehicle have been given in Tables 1 and 2.

Table 1. Backlash values [mm], determined at the testing of passenger car gearboxes

Gearbox specimen No.	Gear 1	Gear 2	Gear 3	Gear 4	Gear 5	Reverse
1 (4-speed)	2.68	3.04	3.85	2.41	–	1.41
2 (5-speed)	3.27	4.09	4.04	2.54	4.96	1.67
Requirements	2.6	3.2	3.2	2.6	3.3	2.0

Table 2. Backlash values [mm], determined at the testing of drive axles (final drives) intended for a delivery motor vehicle at our laboratory

Distance travelled, represented at the endurance rig tests	Drive axle specimen No. 1	Drive axle specimen No. 2	Require-ments
Before tests	0.18–0.27	0.11–0.22	0.15–0.20
100 000 km	0.21–0.31	0.13–0.26	
200 000 km	damaged before test completion	0.13–0.27	

Efficiency of gear transmission units (gearboxes and drive axles)

The efficiency of a gear transmission unit is defined as the ratio of useful power output to the power input of the transmission unit under consideration:

$$\eta = P_2 / P_1 = M_2\,\omega_2 / M_1\,\omega_1 \qquad (1)$$

where:

η – efficiency;
P_1 – power input to the transmission unit;
P_2 – useful power output from the transmission unit;
M_1 – torque on the transmission unit input shaft;
M_2 – torque on the transmission unit output shaft(s);
ω_1 – angular speed of the transmission unit input shaft;
ω_2 – angular speed of the transmission unit output shaft(s).

The useful power output can be measured directly or determined as the difference between the power input and power losses. In gear transmissions, power is lost due to the following:

- friction on the gears being in mesh;
- friction in bearings;
- oil churning.

The accuracy of determining the efficiency depends not only on the accuracy of measurements of the individual quantities comprising formula (1), but also on the form of the equation. For high-efficiency gear transmission units, better accuracy may be achieved by measuring the power losses in the unit under test rather than the input and output power values.

Various power loss measurement methods are known, which are employed depending on the accuracy level required and the laboratory equipment available. Examples of such power loss measurements include the following:

- estimating power losses on a four-square test rig (power-circulating gear testing machine);
- estimating power losses on a two-square test rig (open-loop testing machine with a torque dynamometer);
- estimating power losses by temperature measurement.

One of the methods of determining the efficiency of a gear transmission unit consists of measuring the torque and rotational speed on the transmission input and output. Using the test described here as an example, the transmission system tested was a combination of a gearbox and drive axle, and it was provided with a power take-off (PTO) shaft used to drive special implements.

The conditions and parameters of the example test were as follows:

- differential: locked (only one axle shaft was used as the power output during the test);
- internal brakes of the transmission system under test: adjusted according to manufacturer's instructions;
- angular speed of the transmission system input shaft: 2 300 min^{-1};
- load applied to the output of the transmission system (ground wheel hub): 2 000 Nm;
- number of gears in which the measurements were carried out for the transmission system under test: 4 (5);
- oil temperatures at which the measurements were carried out for the transmission system under test: 40°C ±5°C, 60°C ±5°C and 80°C ±5°C;
- main quantities measured: torque on the transmission input and output, rotational speed on the transmission input and output, and oil temperature in the transmission system.

Major components of the test rig:

- DC electric motor;
- test rig transmission units;
- eddy-current brake unit;
- transmission shafts.

Major measuring systems of the test rig:

- channels to measure the torque on the input and output of the transmission system under test;
- channels to measure the rotational speed on the input and output of the transmission system under test;
- channel to measure the oil temperature in the transmission system under test.

The general view of the test rig during the measurements of efficiency of the power transmission system under test is shown in Figure 11.

Figure 11. General view of the test rig to measure the efficiency of a power transmission system

Noise of operation of gear transmission units (gearboxes and drive axles) (an option)

The noise generated by gear transmission units during operation is measured during performance testing. The noise is determined in comparative terms by measuring the acoustic pressure in special test conditions adopted by convention. According to one of the methods usually adopted, the measurements are carried out either in very large rooms when all the other noise-emitting devices are switched off, or in anechoic chambers. At first, the direction of the highest emission level is identified by moving the microphone of a measuring instrument around the transmission unit under test. Then the microphone is placed in the direction of the highest emission level at a distance of 0.125 m from the unit under test, and measurements are carried out at different constant rotational speeds of the transmission unit. Before the measurements, the properties of the acoustic field are checked, to determine the difference between the average noise level at the measurement point and the average noise level at a point situated in the same direction but at a distance twice as long as that mentioned before. This difference shall not be less than 3.5 dB. There are no quantitative requirements regarding the noise generated by gear transmission units; therefore, it is assumed by experience that the highest noise level determined by this method should not exceed 87 dB(A) for passenger car gearboxes and 92 dB(A) for gearboxes of motor trucks and buses.

For the acoustic characteristics of gear transmission units to be more precisely examined, tests are conducted according to individually designed specifications.

In most cases, the tests consist of measuring the acoustic pressure or acoustic power, and carrying out a spectral analysis of the signals recorded. The spectra obtained make it possible to identify the major components of the sounds emitted; moreover, the reasons for the excessive noisiness of the gearbox under test can sometimes be deduced from the predominating frequencies. However, the complexity of gear transmissions can make inferences like this very difficult or impossible. For illustration purposes, the result of such a spectral analysis is shown in Figure 12. Near-field measurements, as it is in the case of gear transmission testing, are burdened with certain errors, and their results depend on the location of the measuring point. This requirement often results in a distorted picture of the spectral structure of the sound. Therefore, correlation and coherence methods and sound intensity measurements are used to identify the sources of noise in gear transmissions.

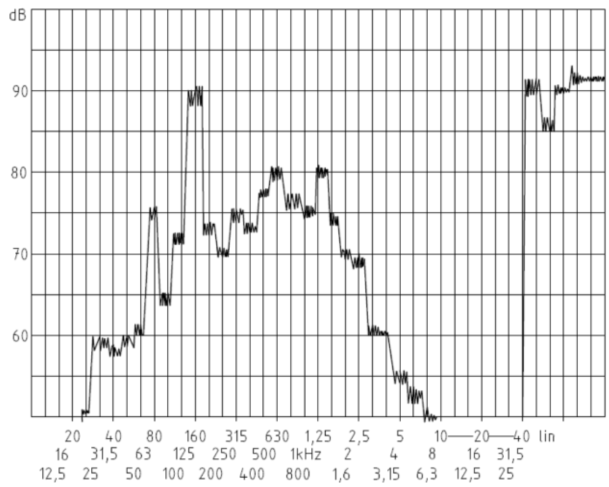

Figure 12. Graph showing results of third-octave spectral analysis of the noise emitted from a passenger car gearbox; the microphone was placed at a distance of 125 mm from the gearbox in the direction of maximum emission

Tests of the noise of operation of drive axles are identical to tests of the noise of gearboxes. Example results of testing the noise of operation of gearboxes and drive axles are shown in Tables 3 and 4.

Table 3. Noise [dB(A)] emitted from a passenger car gearbox, measured at our laboratory; the measurements were carried out at a gearbox oil temperature of 60–65°C

Gear	Noise [dB(A)] at different input shaft speeds [min^{-1}]					Requirements
	1000	2000	3000	4000	5000	
1	74.0	82.0	86.0	87.0	90.0	94 dB(A) max
2	74.0	83.0	85.0	90.0	92.0	
3	74.0	81.5	86.0	88.0	90.5	
4	77.5	83.0	86.0	90.0	93.5	
5	77.5	85.0	88.0	92.5	93.5	
Reverse	77.5	84.5	89.0	91.5	–	

Table 4. Noise [dB(A)] emitted from drive axles intended for a delivery motor vehicle, measured at our laboratory. The measurements were carried out at an input shaft speed of 2 500 min^{-1}

Drive axle specimen No.	Stabilized oil temperature [°C]	Conditions of operation	Noise [dB(A)]
1	98	Axle shafts free to rotate	92.5
		Left axle shaft immobilized	94
		Right axle shaft immobilized	95
2	95	Axle shafts free to rotate	84.5
		Left axle shaft immobilized	Axle shaft left free to rotate because of differential seizing
		Right axle shaft immobilized	Axle shaft left free to rotate because of differential seizing

Test rig designs

Example schematic drawings of test rigs designed for performance testing of gearboxes and drive axles are shown in Figures 13 and 14. Test rigs of this kind may be provided with measuring equipment that would make it possible to measure the oil temperature, no-load mechanical drag, no-load drag power, and noise.

Figure 15 shows an example schematic drawing of a test rig for the measuring of no-load mechanical drag and no-load drag power. An example schematic drawing of a test rig for the measuring of gear engagement and synchronization times and synchronization forces and for the checking of functioning of gearbox mechanisms is shown in Figure 16. A schematic diagram of a system to measure the static gear-change forces and to check the functioning of gearbox mechanisms has been presented in Figure 17. Example schematic drawings of test rigs for the measuring of backlash / angular backlash in a gearbox and in the final drive of a drive axle have been presented in Figures 18

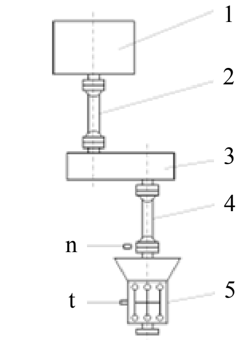

Figure 13. Schematic drawing of a test rig for determining the maximum stabilized oil temperature in gearboxes: 1 − electric motor; 2, 4 − transmission shafts; 3 − test rig gear transmission (optional); 5 − gearbox under test; n – gearbox input shaft speed measuring system; t − gearbox oil temperature measuring system

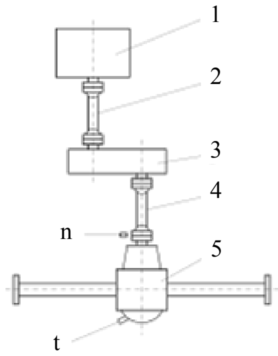

Figure 14. Schematic drawing of a test rig for determining the maximum stabilized oil temperature in drive axles: 1 − electric motor; 2, 4 − transmission shafts; 3 − test rig gear transmission (optional); 5 – drive axle under test; n – drive axle input shaft speed measuring system; t – drive axle oil temperature measuring system

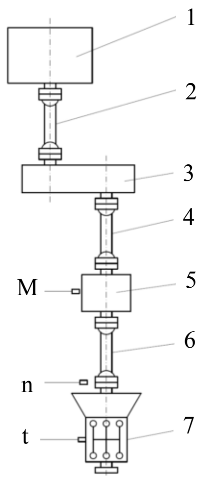

Figure 15. Schematic drawing of a test rig for the measuring of no-load mechanical drag and no-load drag power in gearboxes: 1 – electric motor; 2, 4, 6 − transmission shafts; 3 – test rig gear transmission (optional); 5 – torque meter; 7 – gearbox under test; M – gearbox input shaft torque measuring system; n – gearbox input shaft speed measuring system; t − gearbox oil temperature measuring system

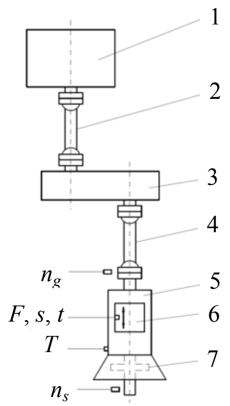

Figure 16. Schematic drawing of a test rig for the measuring of gear engagement and synchronization times and synchronization forces and for the checking of functioning of gearbox mechanisms: 1 − electric motor; 2, 4 − transmission shafts; 3 − test rig gear transmission (optional); 5 − gearbox under test; 6 – automatic gear-change system; 7 – clutch disc; n_g − gearbox output shaft speed; n_s − gearbox input shaft speed; F − gear-change (synchronization) force; s − gear-change lever displacement; t − time; T − gearbox oil temperature

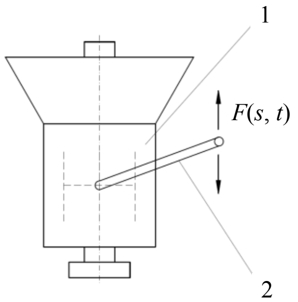

Figure 17. Schematic diagram (test principle) of a system to measure the static gear-change forces and to check the functioning of gearbox mechanisms: 1 − gearbox under test; 2 − gear-change lever; F − force on the gear-change lever knob at static gear changing; s − gear-change lever displacement; t − time

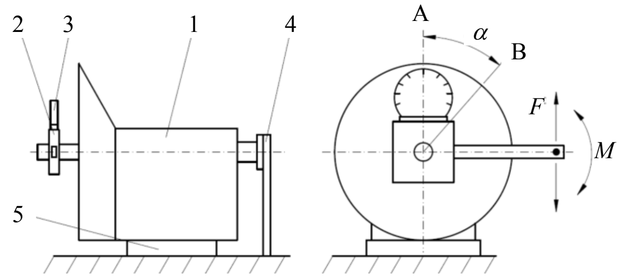

Figure 18. Schematic drawing (test principle) of a test rig for the measuring of backlash / angular backlash in a gearbox: 1 − gearbox under test; 2 − instrument base; 3 − protractor (e.g. optical); 4 − bracket to immobilize the output shaft of the gearbox under test; 5 − mounting fixture for the gearbox under test; F − force applied to the instrument base lever; M − gear-mesh loading torque; A, B − outermost positions of the gearbox shaft in mesh when loaded with a torque of ±M; α − angular backlash measured

and 19, respectively. Example schematic drawings of test rigs for determining the efficiency of gearboxes and drive axles have been shown in Figures 20 and 21, and a schematic diagram of a test rig for gear transmission noise examination is shown in Figure 22. As regards leak-tightness testing, no schematics of special test rigs have been provided here because gear transmissions are checked for being leak-proof during other tests, when the device under test is filled up with oil.

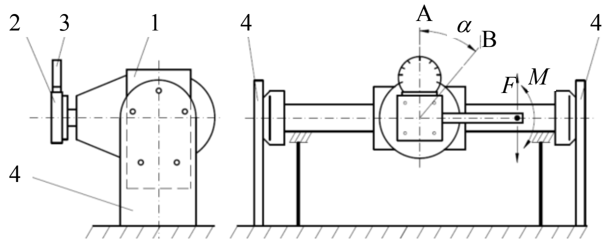

Figure 19. Schematic drawing (test principle) of a test rig for the measuring of backlash / angular backlash in the final drive of a drive axle: 1 − drive axle under test; 2 − instrument base; 3 − protractor (e.g. optical); 4 − bracket to immobilize the hubs of the drive axle under test; F − force applied to the instrument base lever; M − drive axle gear-mesh loading torque; A, B − outermost positions of the drive axle input shaft in mesh when loaded with a torque of ±M; α − angular backlash measured

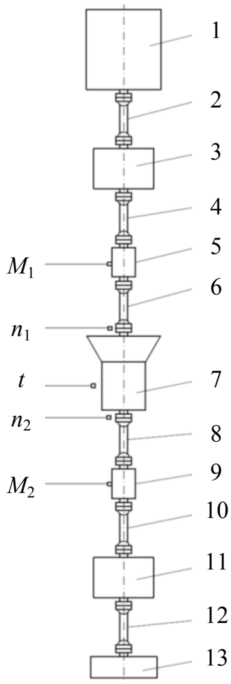

Figure 20. Schematic drawing of a test rig for determining the efficiency of gearboxes: 1 − electric motor; 2, 4, 6, 8, 10, 12 − transmission shafts; 3, 11 − test rig gear transmissions (optional); 5, 9 − torque meters; 7 − gearbox under test; 13 − brake. Basic quantities measured, related to the gearbox under test: M_1 – gearbox input torque; M_2 – gearbox output torque; n_1 − gearbox input speed; n_2 − gearbox output speed; t − gearbox oil temperature

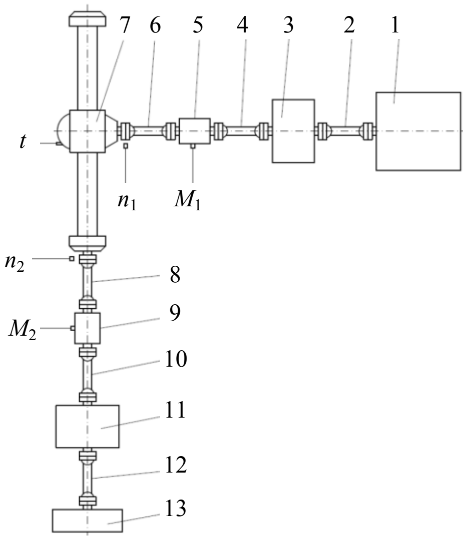

Figure 21. Schematic drawing of a test rig for determining the efficiency of drive axles: 1 − electric motor; 2, 4, 6, 8, 10, 12 − transmission shafts; 3, 11 − test rig gear transmissions (optional); 5, 9 − torque meters; 7 − drive axle under test; 13 − brake. Basic quantities measured, related to the drive axle under test: M_1 – drive axle input torque; M_2 – gearbox output torque (on the drive axle hub); n_1 − drive axle input speed; n_2 − gearbox output speed (of the drive axle hub); t − drive axle oil temperature

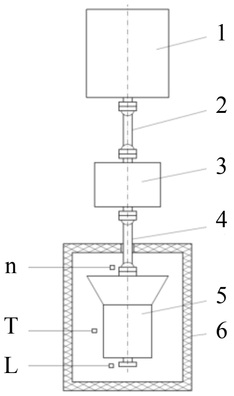

Figure 22. Schematic diagram of a test rig for gearbox / drive axle noise examination: 1 − electric motor; 2, 4 − transmission shafts; 3 − test rig gear transmission (optional); 5 − device (gearbox or drive axle) under test; 6 – anechoic chamber. Basic quantities measured, related to the device under test: n − input speed of the device under test; T − oil temperature in the device under test; L − intensity of the sound emitted by the device under test during operation

Concept of test rigs intended for performance testing of power transmission units

The test rig of this type is characterized by the following basic features:

– intended use;
– defining the type of the tests to be carried out;
– location in the laboratory and the area occupied;
– test rig component units, with their specifications and quantity;
– degree of complexity of the test rig construction;
– measuring equipment;
– achievable operating parameters.

Two schematic diagrams of the test rigs have been presented as examples in Figures 23 and 24. The facility shown in Figure 23 is intended for the carrying out of tests aimed at determining the following performance characteristics of power transmission units:

– maximum stabilized oil temperature;
– no-load drag (torque and power);
– static gear-change forces (including control mechanism functioning checks);
– gear engagement and synchronization times and synchronization forces;
– backlash / angular backlash;
– leak-tightness;
– operating noise (optional).

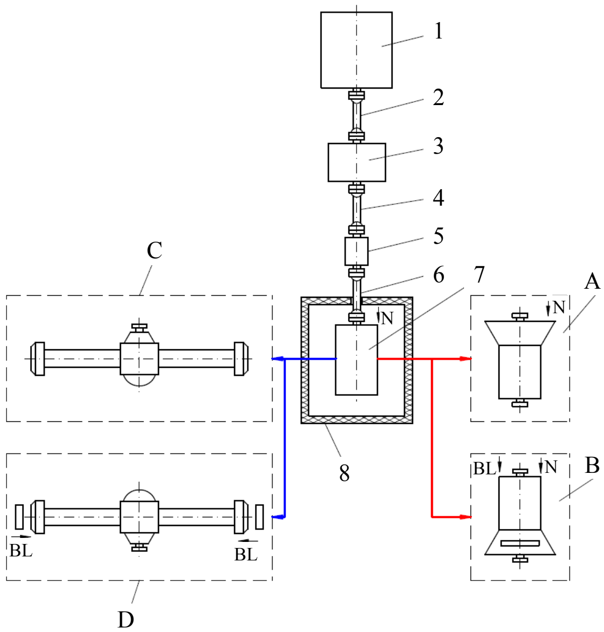

Figure 23. A concept of multifunctional test rig No. 1 for performance testing of power transmission units: 1 − electric motor; 2, 4, 6 − transmission shafts; 3 − test rig gear transmission; 5 − torque meter; 7 − device under test; 8 − anechoic chamber (optional); N − system to drive the device under test

A – gearbox; examination of:
maximum stabilized oil temperature;
no-load drag and drag power;
static gear-change forces (the N driving system: disengaged);
operating noise (optional).
B – gearbox; examination of:
gear engagement and synchronization times;
angular backlash (the BL output shaft lock: engaged).
C – drive axle; examination of:
maximum stabilized oil temperature;

no-load drag and drag power;
operating noise (optional).

D – drive axle; examination of:
angular backlash (the N driving system: disengaged; the BL drive axle hub lock: engaged).

Figure 24 shows a test rig intended for the carrying out of tests aimed at determining the following performance characteristics of power transmission units:

– efficiency;
– backlash / angular backlash;
– static gear-change forces;
– leak-tightness.

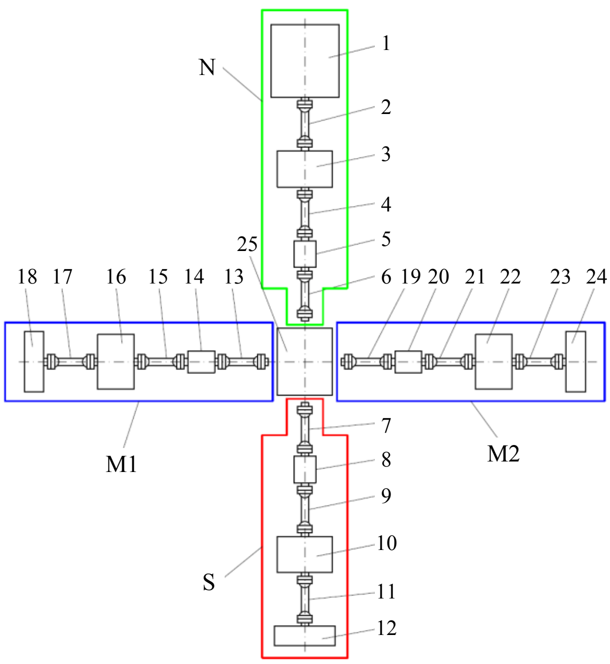

Figure 24. A concept of multifunctional test rig No. 2 for performance testing of power transmission units: 1 – electric motor; 2, 4, 6, 7, 9, 11, 13, 15, 17, 19, 21, 23 – transmission shafts; 3, 10, 16, 22 – test rig gear transmissions; 5, 8, 14, 20 – torque meters; 12, 18, 24 – test rig brakes; 25 – device under test (gearbox or drive axle). N – test rig driving part; S – test rig power receiving part, used for gearbox testing; M1, M2 – test rig power receiving part, used for drive axle testing

In the latter case, the concept adopted does not include the possibility of carrying out all the planned performance tests on one general-purpose facility because of significant diversity of the equipment needed for individual tests, although this is also feasible.

Some of the tests, thanks to their simplicity, can be performed on both test rigs. On the other hand, the facility shown in Figure 23, if some measuring equipment is changed, provides a means of carrying out endurance tests as well (the endurance testing is not covered by the scope of this article).

Conclusions

The engineering design basis for the construction of a facility for performance testing of gearboxes and drive axles has been presented. In consideration of the quality of the tests to be carried out, such a facility should be designed taking into account the following requirements:

1) The putting into service of an automated inertial test rig represents a significant development and modernization in the existing research potential, enabling the carrying out of dynamic tests representing the real operation of power transmission systems and, therefore, the build-up of the stock of possible orders for the test jobs that are needed but currently cannot be performed at the laboratory.

2) The response to any changes in test implementation systems, including the response to any interference, should be many times faster than it is in the case of the systems being manually controlled by human operators.

3) Results of a large number of measurements, with their accuracy greatly exceeding that attainable by human senses, should also be taken into account.

4) The test rig should be capable of implementing rapid changes, impossible for direct reproduction when the system is manually controlled by a human operator.

5) It should be possible to build, apply, and use complex test rig control algorithms and to implement (in accordance with the task being carried out) modifiable algorithms of system operation without the necessity of changing the basic structure of the test rig.

6) Direct programming of the test rig operation and of selected individual test parameters should be possible.

7) The system should provide a possibility of data recording as well as information archiving and reproduction.

8) It should be possible to save individual measuring configurations in the test rig computer system (i.e. to prepare finished test "scenarios" ready to use).

9) A possibility of automatic control of peripheral equipment, such as printers, alarm systems, etc., should be provided.

10) A possibility to develop the supervision systems in case of need should be provided as well.

From work safety and ergonomics perspectives, the following should be considered important as regards rig testing:

1) Automatic equipment of test rigs may operate for a long time without interruptions, while human brain functions on impulses and it tires out quite quickly.

2) After many hours of work, especially at night-time and at raised temperatures prevailing in laboratory rooms, the human reaction time becomes much longer and the external stimuli (visual, auditory, and thermal) are perceived differently by tired and rested operators.

3) Automatic test rig operation and automated supervision of it provide a possibility of considerable shortening the periods of required presence of personnel in the test rig area, thereby reducing accident risk.

4) Incomparably faster operation of automatic systems as regards measurements and information acquisition helps to avoid personnel's exhaustion.

In economic terms, the following factors are essential:

1) possibility of carrying out various test jobs that are in demand but currently cannot be undertaken at the laboratory because of the unavailability of an adequate test rig;

2) quick and highly productive operation of automated test rig systems, offering the possibility of great savings in labor costs;

3) quick detection of any problems that might occur during tests and possibility for the entire facility to be stopped in time to prevent severe damage, thus providing reliable identification of reasons for the problem, and eliminating excessive costs caused by the necessity of repeating tests or to paying compensation to the client;

4) savings in electric energy consumption thanks to the use of drive systems capable of returning the excess electricity to the power grid and to the use of automated supervision systems rationally controlling the test rig operation;

5) possibility of reducing the test rig construction costs by applying AC electric machines as motors and brakes to the system, such that some test rig gear transmission units could be eliminated;

6) automatic protection of the test rig, the costly components of which are exposed to the risk of damage.

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