

METHODS OF VERIFICATION OF MAIN TRANSMISSION GEAR BOXES ON TEST BENCH USING VIBRATION MEASUREMENT

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Abstract

The methodology of research and the inspection of the main transmission gearboxes of multi axles armoured vehicle ROSOMAK under operating conditions are described in this article. Tests are carried out on the bench to allow force transmission changes to the extent similar to the loads that occur during vehicle operation. Transmission gear boxes are driven by an internal combustion engine 359 through a reducer (standard engine gearbox). This reducer increase torque and decrease shaft speed similar to condition in the vehicle. On the other side of the transmission, multiplying gearbox coupled with a brake dynamometer were placed. It increases speed according to requirements of the work field of the dynamometer W-230. The combustion engine made possible to measure vibration of transmission gearboxes under the high torque. The torque is greater than obtainable by using electric motors. The value of the inertia moment loading axle shaft was determined using a simplified kinematic diagram. During the tests, vibration were measured at selected points of the housing using different methods of attachment of vibration transducers. First, the screw for assembling the vibration sensor was glued to the hull of gearbox. Second, the sensor was connected with gearbox by permanent magnets. Third, the sensor was used as a hand probe. The results of vibration measured by these methods were compared. Temperature of the gearbox was also measured by using thermocouples and thermovision methods. Selected results are presented and analysed of vibration transmission and its temperature at selected working conditions.

Keywords: multi wheeled vehicle, transmission gearbox, vibration measures, test stand

1. Introduction

The threat for vehicles coming from improvised explosive devices is increasing. Mainly, powertrain and vehicles chassis are exposed to the damage. Often after a mine explosion, those mechanisms have no visible damage, but their actual condition is unknown. Hence the need to inspect these mechanisms during repair of the vehicle. Therefore, bench and methodology of the technical condition of the main gearbox and differentials driving axles of AMV ROSOMAK on the bench was developed, providing a mapping service of operating conditions of those gearboxes.

Typically, bench testing of transmission consists of mechanical efficiency measurement, internal friction torque of the torque differential distribution to the drive shafts and fatigue tests [4, 5]. They are implemented on measuring stations using high-power electrical machines, as a drive motor, and dynamometer. In presented work, verification of the driving axle technical condition on the built bench with internal combustion engine with higher power density than common electric motors. Driving axles were powered by diesel fuel ignition S-359 engine, and

loaded by Schenck W-230 dynamometer in so called “open loop power circuit“. This allowed to load transmissions with torque similar to the operating point.

2. Solution of the test bench

The torque of AMV ROSOMAK is transmitted from the engine through automatic transmission(SB) with torque converter (PH)to transfer box (SR) with inter axial differential MMR (Fig. 1). The torque is split into two sides - front driving axles connected in series MN1 and MN2 and rear axles MN3 and MN4. The final drives of driving axles use bevel gears with a gear ratio 1:33. All the differentials are fitted with locking gear.

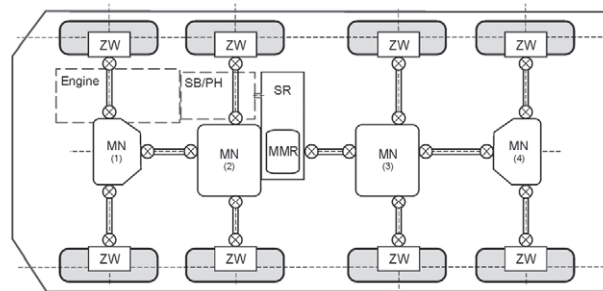


Fig. 1. Block diagram of the drive system AMV: MN1...MN4 – axles 1...4, SB/PH – torque converter/transmission, SR – transfer gear, MMR –gear between the differential, ZW – hub reduction gear

ROSOMAK gear conveyor axes loads calculation for operating speed of the engine crankshaft (1500 to 2100 rev/min) showed that with a steady torque distribution to the various axes drive torque at the input of each gear is in the range from 239 Nm (for driving the gear VII –step-up gear, the transmission input shaft speed of 2942 rev/min) to 3337 Nm (gear I, at the transmission input shaft speed 221 rev/min).

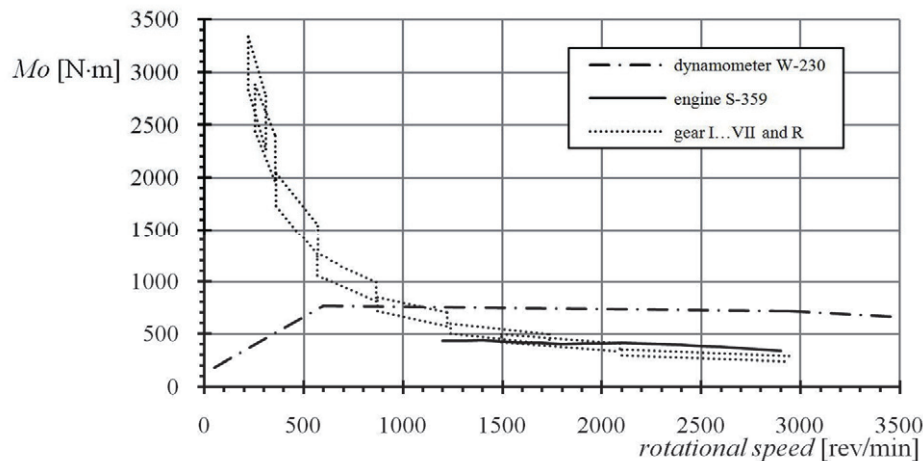


Fig. 2. A comparison of the external S-359 motor, brake performance dynamometer W-230, and field gear unit gear in individual gears

Load field (torque versus speed) of each gear set for different gear ratios shown in Fig. 2 by the dotted line. This allowed a comparison of the calculated torque on the transmission input from the external characteristics of the S-359 engine and the characteristics of the field work dynamometer W-230. In the case of direct drive driving axles by S-359 engine, the torque driving the gears would be smaller than the present moment while riding on the most widely used conveyor gearbox

ratios (I to IV). On the other hand, the characteristics of the brake at low speeds do not allow obtain real moments of gearbox load.

Hence, in order to increase available torque, an additional five-speed gearboxes from ZF were applied (SB1 and SB2, Fig. 3).

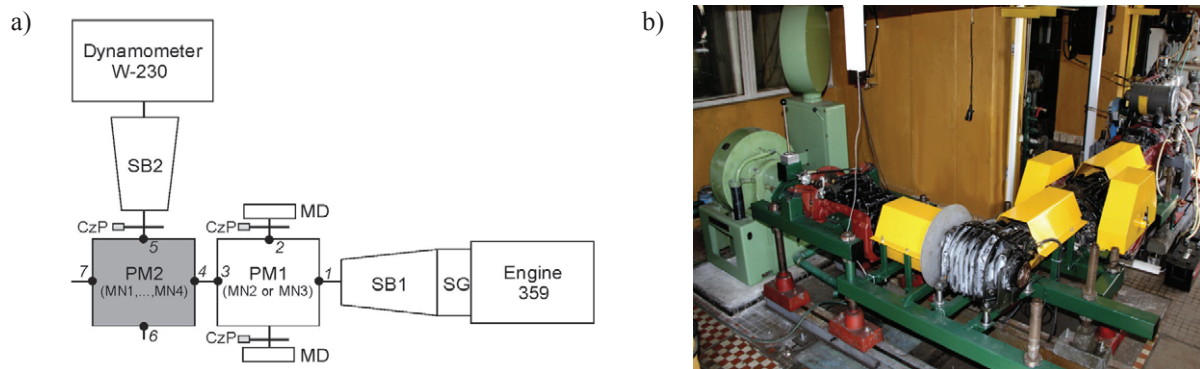


Fig. 3. The test bench: a) general scheme of completion, b) overview: SG – clutch, SB – gearbox, PM – driving axle, MD – additional weight, CzP – speed sensors, 1...7 – measurement of the vibration signal

The first gear box SB1 is placed between the engine S-359 and the test transmission axis, in such a way as to obtain a reduction of transmission input shaft speed and increase torque. The second gearbox is mounted on the brake, between the second transmission gear and W-230 dynamometer. Gearbox SB2 was placed in the reverse arrangement, so that the rotational speed increased brake shaft according to the characteristics of the brake. Such a solution, depending on the adjustable transmission ratio enables:

- adjust the characteristics of the engine and dynamometer to the performance of the gear unit,
- loading tested gearbox similar to the operating moment of the driving axle (Tab. 1).

In the research condition selection, direction of rotation of the transmission input shaft on the bench was considered. Due to the opposite direction of rotation of the drive shaft while moving forward (see Fig. 1), the construction of the outer main gear labelled MN1 and MN4 is different. They use roller pinion attacker respectively for the right and counter-clockwise line of teeth. This approach was necessary to avoid a situation in which the low bearing loads, power interdental attacker would cause retraction of the shaft to the gearbox and lock up in result [5, 6]. Knowing this risk, research were carried out (for gear PM2, Fig. 3):

- for the front transmission MN1 – in first gear,
- for the rear transmission MN4 – in reverse.

The test conditions on the test bench are shown in Fig. 3, providing the opportunity to use the full power of the driving motor, being evaluated for the speed range 1500...2800 rev/min engine 359 and summarized in Tab. 1.

Built test bench allowed the simultaneous examination of two transmission gears. The first transmission axle PM1 (axles MN2 or MN3), transmits the torque from the transmission gearbox SB1 to gearbox PM2. The outputs of the differential gear were loaded by rotating masses MD. This method of driving axle load, includes the load effects on the driving axle mechanisms not only the engine torque, but also the moment of inertia hub reduction gear and rotation of transporter wheel.

The value of the inertia moment loading drive axle shaft (shaft and the crown) was determined using a simplified kinematic diagram (Fig. 4) planetary gear forming hub reduction gear in the AMV Rosomak wheels.

The moment of inertia of the elements associated with the mechanisms driving axle was determined using the equation for the kinetic energy of the planetary gear, to separate the impact of individual components of hub reduction gear total moment of inertia, and making the reduction to the axis of the shaft to the sun gear.

Tab. 1. The test conditions with transmission gear on the test bench

SB1 ratio gearbox [-]	Speed range on the driving axle PM1 input [rev/min]	Torque range on the driving axle PM2 input[Nm]	Possible gearbox ratios SB2 [-]
I gear	135...252	4738...3939	I gear or reverse
II gear	245...457	2618...2176	I or II gear or reverse
III gear	421...786	1521...1265	I...III gear or reverse
IV gear	720...1344	889...739	I...IV gear or reverse
V gear	1145...2137	559...465	I...V gear or reverse
reverse	237...442	2707...2251	I or II gear or reverse

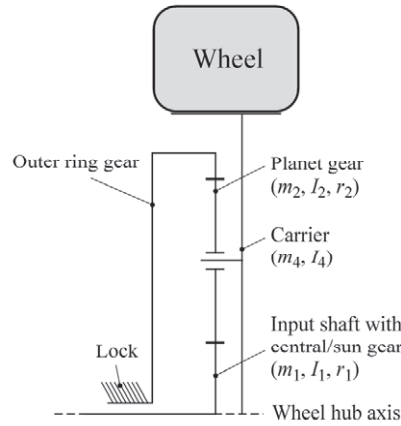


Fig. 4. Simplified schematic of a kinematic planetary wheel transporter reduction gear ROSOMAK

For the individual components of hub reduction gear kinetic energy equation used in the calculation are in the form:

$$E_{K1} = \frac{1}{2} \cdot I_1 \cdot \omega_1^2, \quad (1)$$

$$E_{K2} = \frac{1}{2} \cdot I_2 \cdot \omega_2^2 + \frac{m_2 \cdot V_j^2}{2} = \frac{1}{2} \cdot I_2 \cdot \omega_1^2 \frac{r_1^2}{4 \cdot r_2^2} + \frac{1}{2} \cdot m_2 \cdot \frac{\omega_1^2 \cdot r_1^2}{4}, \quad (2)$$

$$E_{K4} = \frac{1}{2} \cdot I_4 \cdot \omega_j^2 = \frac{1}{2} \cdot I_4 \cdot \frac{\omega_1^2}{(1 - i_w)^2}, \quad (3)$$

where:

E_{K1} – the kinetic energy of the sun gear shaft,

E_{K2} – the kinetic energy of the satellite,

E_{K4} – the kinetic energy of the yoke,

i_w – internal leverage $i_w = -\frac{r_1 + 2 \cdot r_2}{r_1}$,

ω_1 – the angular velocity shaft of the sun gear,

ω_2 – the angular velocity of the satellite,

I_i – moment of inertia of a mass element m_i radius r_i (Fig. 4).

Eventually the moment of inertia reduced to the shaft of the sun gear transmission was described with the formula:

$$I_C = I_1 + 4 \cdot \left(I_2 \cdot \frac{r_1^2}{4 \cdot r_2^2} + m_2 \cdot \frac{r_1^2}{4} \right) + I_4 \cdot \frac{1}{(1 - i_w)^2}. \quad (4)$$

The values needed to determine the moment of inertia are determined by the corresponding direct measurements. Design details of closed mechanisms, on which weight of elements were

determined, were taken from the transporter service manual [7, 8]. Mass measurements of tires and complete wheels confirmed the correctness of the calculation, and the maximum difference between the value determined by calculation and measured (weighed) does not exceed 1%.

Moment of inertia was set to $1.555 \text{ kg}\cdot\text{m}^2$ and developed inertial weight about $1.1 \text{ kg}\cdot\text{m}^2$. The adopted lower value includes, for example, the deflection of tyre, leads to a reduction of the moment of inertia (smaller radius of the wheel) and moreover, put requisite tension in free ends of the drive shafts, which can have a positive impact on vibration reduction occurring in this area.

3. Methodology and research program

The test bench with the measuring instrumentation can conduct the following driving axle tests of AMV ROSOMAK (the tests algorithm is shown on Fig. 5):

- preliminary validation in gears at idle speed,
 - long-lasting test under a load corresponding exploitation conditions,
 - vibration measurement of the input and output shaft bearings, using vibration analyser (analyser Emerson CSI 2130 with the accelerometer A0760GP are shown on Fig. 6),
 - measuring the temperature of the oil in the gear housing with oil temperature sensor mounted in the drain plug,
 - recording infrared temperature distribution in the most loaded areas of gears housings.
- The analyser and transducer were calibrated periodically using hand held calibrator 304C06.

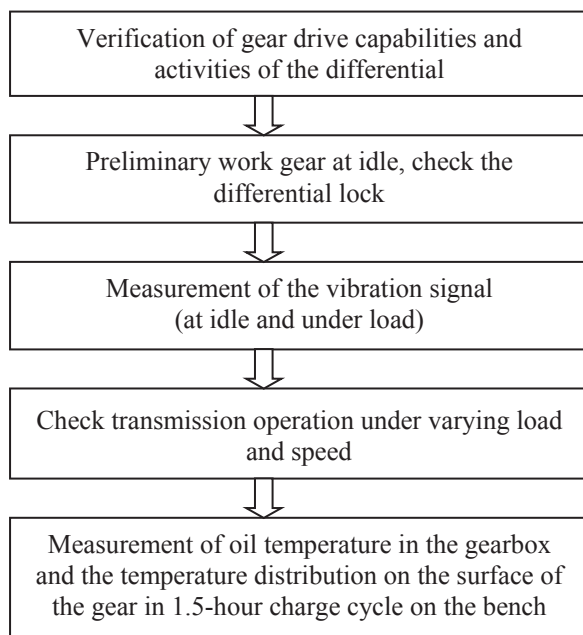


Fig. 5. Transmission gearbox testing algorithm

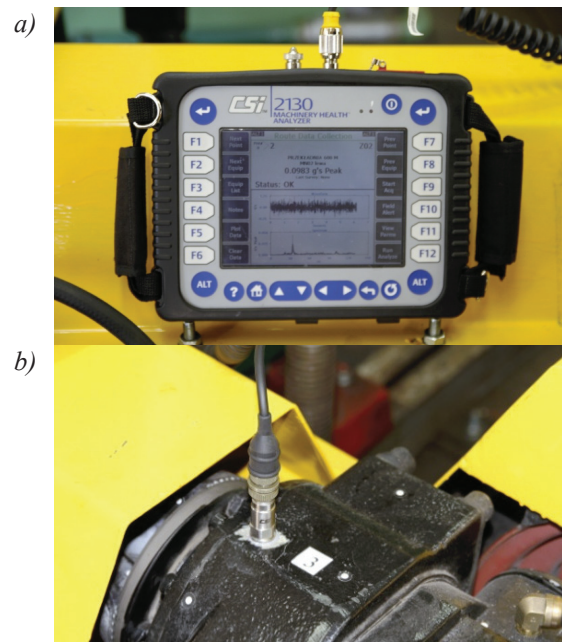


Fig. 6. Equipment for vibration measurements: a) analyser, b) transducer

Conditions for registration of the vibration signals, the parameters of the engine and gearbox drive, were set taking into account the characteristics of vibration analyser (Emerson CSI 2130), and the expected form of vibration waveforms and spectra recorded vibration signals. The values of the rotational speed of the drive shaft at the tested transmission entry, due to the speed of the engine and the gear ratios, were respectively selected SB1. The scope of the analysed frequency bands of vibration signal were set so that the harmonic signal recorded, corresponding to the rotational speed of the drive shaft (connected component of the unbalance shafts), was equal to frequency of 16 lines of obtained spectrum, and harmonic corresponds to the main gear teeth of examined gearbox, entering into tooth contact, falling in the frequency spectrum obtained 384

lines.

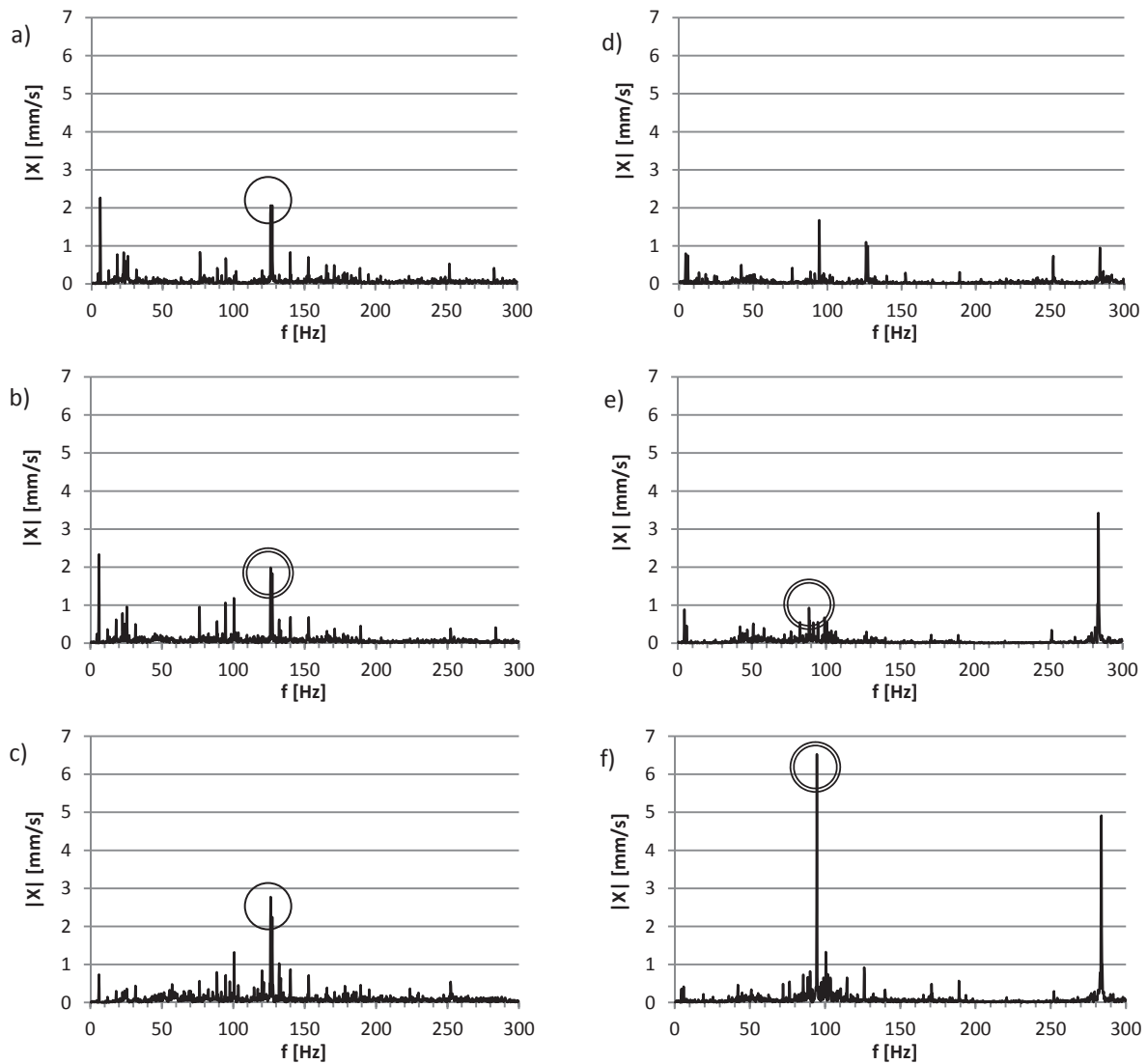


Fig. 7. Sample spectra of vibration signals (position sensor according to Fig. 3): a) the transmission gear PM1(MN2, or MN3) near the inputs shaft bearing from the source drive(point 1), b) the transmission PM1 gear near the right output bearing driveshaft (point 2), c) the transmission PM1 gear near the output shaft bearings for transmissions PM2 (point 3), d) the transmission PM2 gear near the input shaft bearing from the transmission gear PM1 (point 4), e) the transmission PM2 gear near the output bearing right driveshaft, loadable by the brake(point 5), f) the transmission PM2 gear near the left output bearing driveshaft(point 6)

Last, 1600 spectral line coincides with the upper limit of the analysed band. The adoption of these types of simple assumptions about the signal recorded, was caused due to the lack of detailed information about the internal structure of tested driving axles. This refers to the kind of used differentials and locking elements of these mechanisms, the main gear wheel bearings, bearing components differentials, bearings, drive shafts, etc.

Sample spectra of vibration signals at the speed of the engine corresponding to the maximum torque n_M , are shown in Fig. 6. Treating engine and transmission system SB1 as vibration source, it can be seen that at this speed a slight increase in the amplitude, characteristic vibration components at the output of the transmission can be seen (marked with a circle in Fig. 6a and 6c). Also beneficial effects of using a flywheel are disclosed when comparing to other typical amplitude-term storage (marked with a double circle) on the output drive shafts internal gearbox (Fig. 6b) and the outer gearbox (Fig. 6e – from the brake and Fig. 6f – driveshaft free).

The last stage of the study was to test the value and temperature distribution for which the

transmission heats up during operation. It consisted of 1.5 hours of performance measurement cycle, divided into several phases in gear with different mass and rotational speed of the engine. Detailed conditions of these tests are summarized in Tab. 2.

Tab. 2. Conditions for registration of the oil temperature and thermal spectra in selected areas of gear power units of AMV

Measurement time[minutes]	10	10	10	10	10	10	10	5	10	5
Engine conditions	$M_{o_{max}}$	$M_{o_{max}}$	$N_{e_{max}}$	n_{bjal}	$M_{o_{max}}$	$M_{o_{max}}$	$M_{o_{max}}$	n_{bjal}	$N_{e_{max}}$	$N_{e_{max}}$
<i>$M_{o_{max}}/N_{e_{max}}$ – engine under load respectively with a maximum torque / maximum power, n_{bjal} – operation of the engine at idling speed</i>										

At the end of each phase oil temperature and the measured temperature distribution of gear housing in transmission were recorded by thermal imaging camera. Example results of measurements of the temperature distribution at different places in the gear are shown in Fig. 7. Detailed results of measurements of the thermal load and vibration spectra are shown in separate publications.

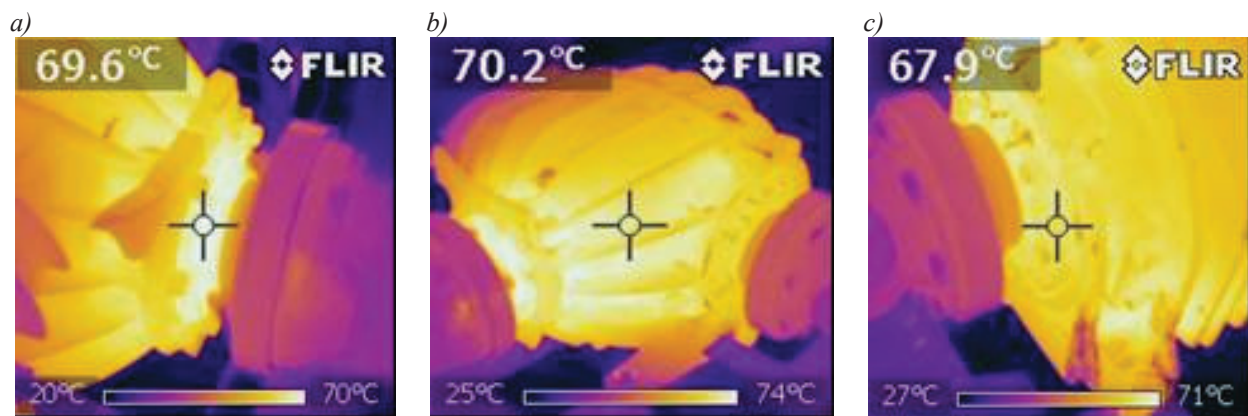


Fig. 7. The temperature distribution of the gear unit showing the busiest areas during the transmission operation: a) roller bearings attacker, b) a differential with wheel packing, c) less loaded bearing driveshaft

4. Conclusions

1. Built and presented test stand allows testing of AMV ROSOMAK driving axes in conditions similar to loads under real operating conditions. This is achieved through the use of an internal combustion engine with the gearbox in the drive gear and a significant increase in torque test drive transmissions. In this method, however, the drive gear has a disadvantage because it increases „background“ vibrations which may affect the test gear.
2. An important advantage is the possibility to register the position of gear vibration during idle and under load. Vibration measurements are possible by using sensors placed around the input shaft bearings, output gear and other places of housing.
3. Test bench allows measuring oil temperature inside the gears and the temperature distribution on the gear housing, thus revealing loaded areas. This also allows to check the correct lubrication, and a preliminary assessment of the technical elements of a confirmed transmission of vibration measurements.
4. Described by multi-rate transmission gear, complemented by measuring backlash and friction torque of the differential, and tracking the possible adjacent of gears, gives the ability to perform a comprehensive assessment of axles and classify for further use.
5. Currently, development work for better diagnosis of transmission properties is underway, including internal resistance measurements of gears and vibration on idling. Tests are

conducted on the dynamometer bench with an electric motor.

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