GEAR PUMP HYDRAULIC TESTING AND SIMULATION

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Hydraulic systems are increasingly present in all segments of our production chains such as agriculture, construction, transportation and other industrial areas. The key component of every hydraulic system are pumps. With newly developed gear pump testing rig, long-term tests of five gear pumps simultaneously are being tested. One pump is tested with medium test dust, one is tested with real wear particles extracted from a filter and the last three are simulating a real hydraulic system with cleanliness of 20/19/17 according to standard ISO 4406. In addition, simulation in program Automation Studio of durability test of gear pump is presented at different pressure operating points. Findings of this research contribute to sustainable development of hydraulic gear pumps and improving efficiency of whole hydraulic systems.

Keywords:
gear pumps,
cleanliness,
test rig,
efficiency,
pressure operating points

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1 Intruduction

The service life of a hydraulic system is the duration throughout which the system is used economically and efficiently and is able to maintain desired temperatures, pressures, and flow rates. There are numerous parameters that affect efficiency. Among the most important are the hydraulic oil, the temperature and the cleanliness of the oil. One of the most important parameters for oil life is the amount of contaminants in the oil, as well as the influence of pressure, oxidation, pressure, radiation shear, and other oil additives that can trigger a chemical reaction [1–4]. More than 70 % of breakdowns in industry workflow are caused by contaminants in hydraulic fluid, with 60 to 70 % of all breakdowns due to solid particles [5]. For implementing condition-based maintenance of hydraulic components and consequently whole systems the cleanliness of oil is of utmost importance [6–8].

Within the hydraulic system there are a number of particles — some created by wear and tear, others that have entered as contaminants from the surroundings or have existed since the system's inception. The dimensions and composition of these particles exert substantial influence on system performance, particularly when the gaps between moving surfaces closely match the particle size [9]. Particles between contact surfaces and near surfaces promotes wear [10]. Most common wear mechanisms are three body abrasion [11] and erosion [12].

ISO 12103-1 is standard for Arizona test dust that is used for testing filters because it has repeatable distribution of particle size and number [13]. There are four types: A1 ultrafine, A2 fine, A3 Medium and A4 coarse. Medium test dust (MTD) is commonly used for accelerating testing of hydraulic equipment. Test dust is more abrasive than common contaminants found in the hydraulic systems which promote wear of hydraulic components [14]. Volumetric efficiency depends on the decrease of initial flow rate of the pump, usually due to wear of the sealing surfaces of elements in components [15]. Wang et. al [16] predicted the remaining useful life (RUL) of a hydraulic gear pump using an accelerated test of the useful life of a gear pump. This method effectively improved the operating efficiency of the hydraulic system and reduced the frequency of failures Gear pumps were previously studied by Ranganathan et al. [17] and Frith [18] using test dust. It was found that the most influential factors for flow rate degradation were chemical composition, hardness, size distribution, and number of particles that caused wear of critical sealing elements. There are numerous simulations of gear pump flow rates done by Rundo [19], Casoli [20], Malsavi [21] and others points to the utility of such tools.

The particles inevitably damage every component in the hydraulic system and cause wear. This wear on the sealing surfaces manifests itself in the form of visible leakage, which leads to a reduction in the volumetric efficiency of the system. A review of the literature indicates that particles commonly present in a hydraulic system are less harmful to the system than test dust. There is some evidence that test dust can efficiently accelerate wear to reduce the time required for long-term testing of hydraulic components. However, to determine the acceleration time, parameters such as particle concentration (oil cleanliness), temperature, pressure and others should be considered. In addition, there is no direct correlation in the literature between the effects of wear particles and test dust on hydraulic component wear. Three durability tests were conducted on hydraulic gear pumps in the laboratory: one without additional contaminants, one with wear particles, and one with test dust. The study presents the design and the test rig itself, the simulation of the flow rates of the gear pumps, and the real measurements of the flow rates and the comparison of the volumetric efficiencies of the pumps.

2 Materials and methods

Three hydraulic test rigs were assembled in laboratory. We tested the effect of oil cleanliness in the hydraulic system on the durability of the system itself and compared the effect of wear particles and test dust (MTD) to normal operating hydraulic system without additional contaminants (without adding contaminants). Fig. 1 shows all test conical reservoirs of test rigs (on the left) and hydraulic valves used for the load of the gear pumps (on the right). Initially, 30 L of ISO VG 46 hydraulic oil was added in first unit that was tested without additional contaminants. Other two test rigs had both 13 L of oil in each unit, one was tested with wear particles and the other with test dust. The flow through the gear pump was measured with stopwatch and weighting of oil. Later the flow rate was determined due to the density of ISO VG 46, which is 0,8551 kg/L. Gear pumps have a displacement of 3,5 cm^3/rev. and a maximum pressure up to 290 bar. In the neutral position of the directional 4/3-way valve all ports are closed. The oil temperature of tank, that was used for testing the gear pump without additional contaminant was around 63±5 ° C. The thermostat was used to control the operation of the cooler set to 70 ° C.

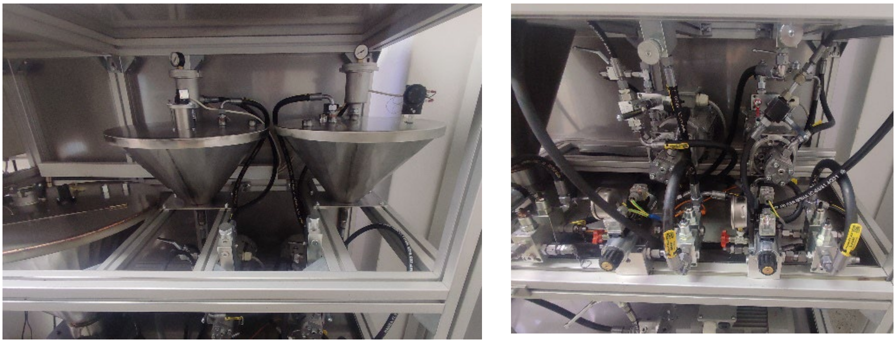


Figure 1: Conical reservoirs (on the left) to prevent the contaminants to settle down and 4/3-way valves with pressure relief valves for the load of the gear pumps (on the right).
Source: own.

The pump (Fig. 2, pos. 1) draws oil from the conical reservoir (Fig. 2, pos. 12), designed as such due to the non-settling nature of particles. This oil is then pushed through the check valve (Fig. 2, pos. 3) into a manually operated 3/2 valve (Fig. 2, pos. 5). Subsequently, it passes through a solenoid-controlled directional valve (Fig. 2, pos. 4). Depending on the valve's position (parallel or diagonal), the oil flows through either working line A or B towards the pressure relief valves (Fig. 2, pos. 7), the cooler (Fig. 2, pos. 10), the priority valve (Fig. 2, pos. 11), and the filter (Fig. 2, pos. 8) back to the reservoir.

The priority valve can be adjusted to maintain a specific pressure differential. This feature allows the valve to either direct oil flow through the filter, ensuring higher cleanliness, or bypass the filter, leading to lower cleanliness levels. The testing process involves setting the system load pressure (the two pressure relief valves) to 220 bar. The hydraulic oil in all three test rigs was initially filtered to achieve cleanliness levels of 16/15/13 or less. Throughout the tests, the temperature was consistently monitored and maintained within the range of 63±5 ° C.

Cleanliness is assessed by manually collecting oil samples, with control of the 3/2 valve at (Fig. 2, pos. 5), and through measurements at the pressure relief valve's measuring connection. Each cycle lasts for 0.5 seconds, during which the solenoid "a" is activated, positioning the spool in the 4/3 directional valve to achieve a parallel configuration. Following this, the solenoid "b" is engaged, returning the spool to its initial diagonal position.

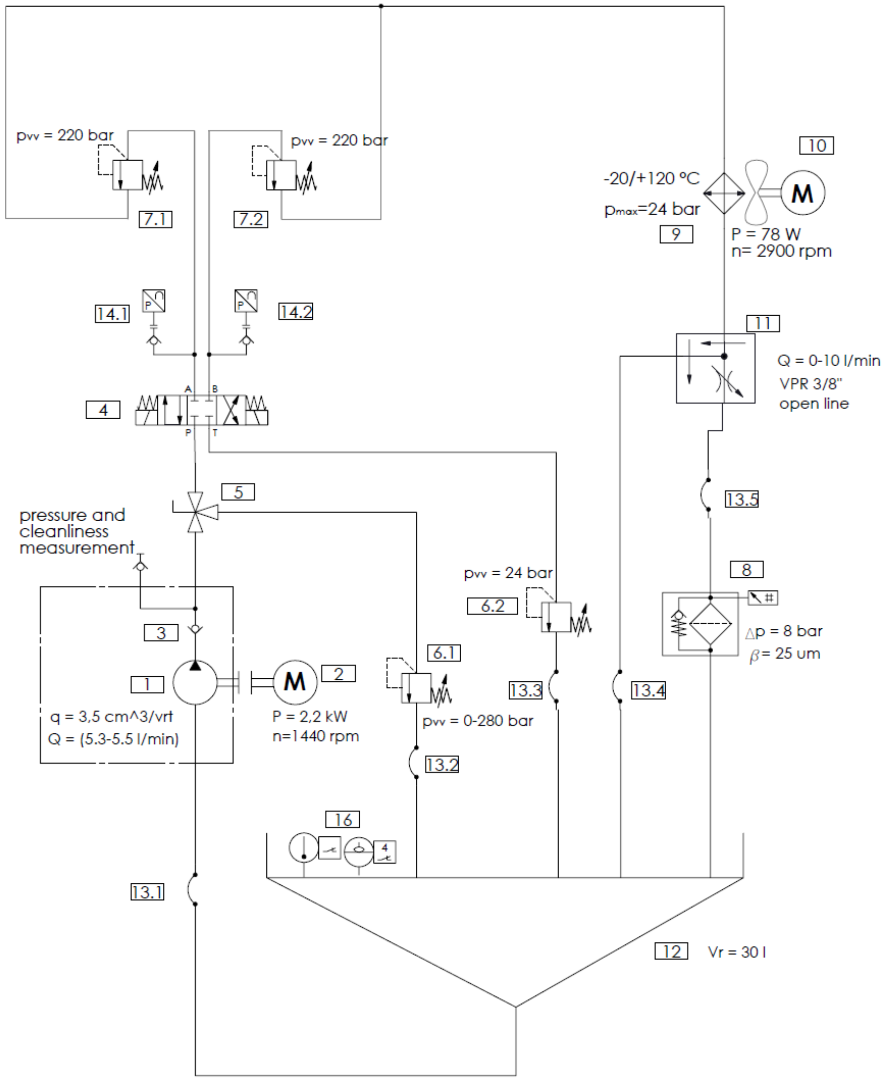


Figure 2: Hydraulic scheme of one test rig for gear pump tested with wear particles and test dust with conical reservoir.

Source: own.

Hydraulic simulation in Automation studio was performed for the gear pump without additional contaminants. The pump flow rate was stable throughout the test. Simulation was designed to simultaneously measure single gear pump at different pressure points (Fig. 3) where the volumetric efficiency curve was specifically defined for the case (Fig. 4).

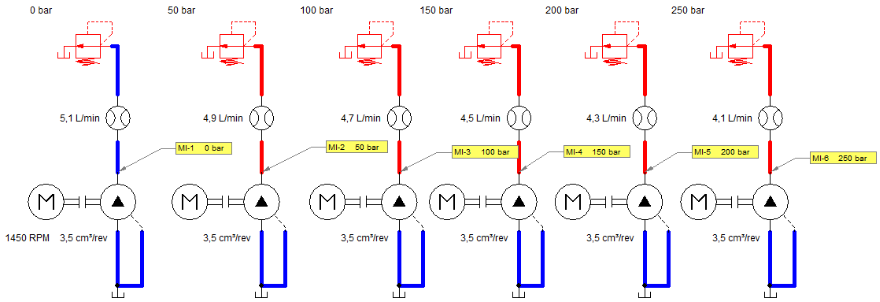
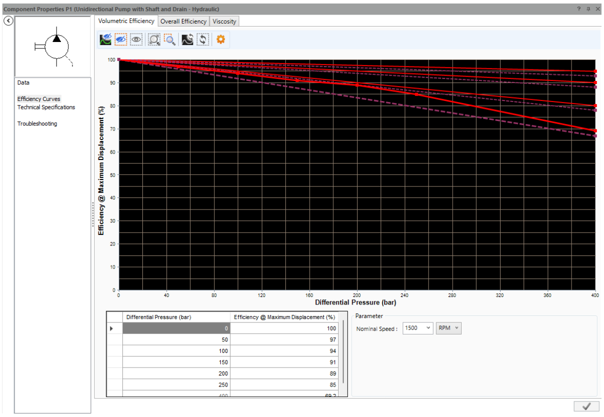


Figure 3: Hydraulic simulation of single gear pump at 5 different pressures (0 bar, 50 bar, 100 bar, 150 bar, 200 bar, 250 bar).

Source: own.



Differential Pressure (bar)	Efficiency @ Maximum Displacement (%)
0	100
50	97
100	94
150	91
200	89
250	85
400	69.2

Nominal Speed : 1500 RPM

Figure 4: Specifically defined volumetric efficiency curve for simulating gear pump flow rates at different pressure points.

Source: own.

3 Results

Three different conditions of oil contamination were introduced in three separate hydraulic test rigs testing gear pumps with aluminium housings. First, the gear pump without any contamination was tested (Fig. 5). The highest measured flow rate was at a pressure of 0 bar, where the pump was not loaded, and averaged 5.28 L/min. At a pressure of 50 bar, the average flow rate was 5.13 L/min, at 100 bar it was 5.00 L/min, at 150 bar it was 4.85 L/min, at 200 bar it was 4.74 L/min, and at 250 bar the flow rate dropped to 4.75 L/min. The gear pump was tested for 576 hours and the volumetric efficiency did not decrease significantly.

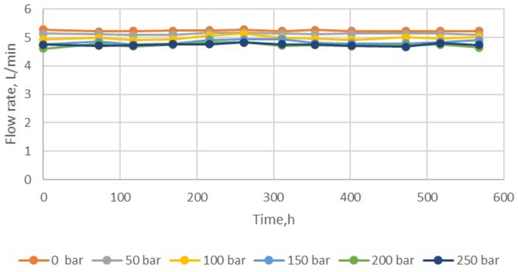


Figure 5: Flow rate of gear pump tested with without the addition of contaminant.

Source: own.

The gear pump tested with wear particles lasted only 19 hours (Fig. 6). At the beginning of the test, we added 8 g of wear particles to 13 L of hydraulic oil, so the concentration of wear particles and oil was 0.615 g/L. At the beginning of the test, the flow rates were similar to the gear pump tested without any additive. The gear pump tested with wear particles had a flow rate of 5.17 L/min at a pressure of 0 bar. At a pressure of 50 bar, the flow rate was 4.99 L/min, at 150 bar 4.81 L/min, at 200 bar 4.66 L/min and at 250 bar 4.57 L/min. After only 19 hours of testing, the flow rate dropped to 4.75 L/min at 0 bar, at 50 bar the flow rate was 2.67 L/min, at 100 bar the flow rate was 0.46 L/min, and at 150 bar and above the flow rate was 0 L/min (not measurable). The oil cleanliness during the test was 22/22/21.

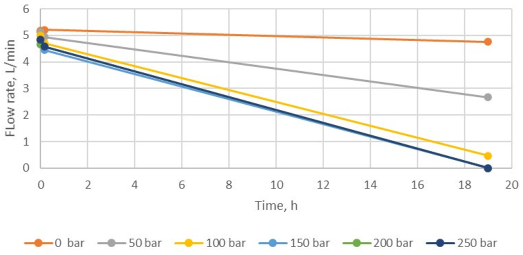


Figure 6: Flow rate of gear pump tested with wear particles with concentration of 0.615 g/L and oil cleanliness 22/22/21.

Source: own.

The gear pump tested with test dust was operated for 70 hours. At the beginning of the test, 0.208 g of test dust was added, resulting in a test dust and oil concentration of 0.016 g/L (Fig. 7). At the beginning of the test, the flow rate at 0 bar was 5.12 L/min. At a pressure of 50 bar, the flow rate was 5.03 L/min, at 100 bar 4.81 L/min, at 150 bar 4.77 L/min, at 200 bar 4.67 L/min and at 250 bar 4.71 L/min. The oil cleanliness during the test was 21/20/18 and level of impurities in oil increased through time due to wear of components. At the end of the test cleanliness was 22/21/19.

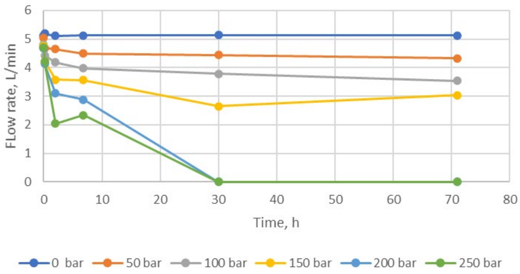


Figure 7: Flow rate of gear pump tested with test dust with concentration of 0,016 g/L and oil cleanliness 21/20/18.

Source: own.

The simulation of the flow rate of the gear pump lasted 5.33 s. The increase of the flow rate depends on the setting of the cracking pressure of the relief valve. Fig. 8 shows that the higher the cracking pressure of the valve, the more delayed is the stabilization of the flow rate and the faster is the transition. The flow rate at 0 bar pressure was 5.07 L/min. At 50 bar the flow rate was 4.88 L/min, at 100 bar 4.68 L/min, at 150 bar 4.48 L/min, at 200 bar 4.28 L/min and at 250 bar 4.09 L/min.

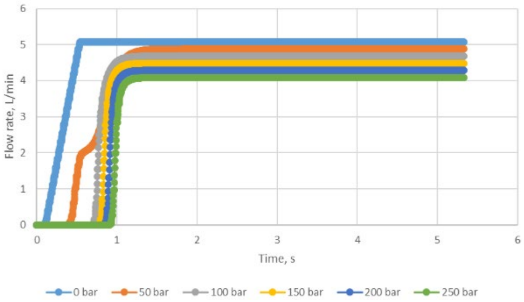


Figure 8: Flow rate simulation of gear pump without additional contaminant.

Source: own.

4 Discussion and conclusion

The difference between three long-term tests of gear pumps is obvious. The gear pump tested without any contamination had a high volumetric efficiency of 100 % at 0 bar, 97 % at 50 bar, 91 % at 100 bar, 89 % at 200 bar, and 85 % at 250 bar.

These efficiencies do not change significantly throughout the test, while the flow rates of the gear pumps tested with wear particles decrease tremendously. In 18 h of operation, the efficiency of the tested gear pump with wear particles drops from 100 % to 92 % at 0 bar, from 99 % to 51 % at 50 bar, from 96 % to 9 % at 100 bar, and from 90 % to zero at the other pressure measurement points (150 bar, 200 bar and 250 bar). In 30 h of operation of the gear pump tested with test dust, the efficiency at 0 bar unusually increased from 99.7 % to 100 % (the reason for this is uncertainty in the measurements), but then dropped from 97 % to 86 % at 50 bar, from 93 % to 73 % at 100 bar, and from 92 % to 51 % at 100 bar. The other efficiencies dropped from about 90 % to zero at both 200 bar and 250 bar.

It can be concluded that the more the gear pump is worn, the greater the differences between the efficiencies at higher pressure. The reason for these phenomena is the higher leakage between the worn sliding/sealing surfaces of the hydraulic elements. In this case, between the housing and the gears. The gear pump tested with wear particles failed after 18 hours and the gear pump tested with test dust failed after 30 hours. Due to the time period between the last measurements, we cannot determine the exact time of failure. When analysing the efficiencies at 100 bar (wear particles 8 %, test dust 73 %) and 150 bar (wear particles 0 %, test dust 51 %), some differences can be observed. It can be assumed that the gear pump tested with wear particles was more damaged (worn) than the gear pump tested with test dust.

Simulating the flow rate of the tested gear pump without additional contaminants shows reasonably comparable results. The simulated flow rates and efficiencies are lower than the real measurements. This is due to the fact that the temperature of the oil is constant at 25 °C. Therefore, the viscosity is higher than in the real environment and the flow rates are slightly lower. Simulating the efficiency curves of hydraulic gear pumps can greatly improve fault diagnosis in the hydraulic system and reduce costs by replacing the gear pump at the most critical time.

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