Influence and Effects of Pressure Variation on the Life Span of External Gear Pumps

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Abstract: *The article analyzes the effects of variable pressure on the life of hydraulic gear pumps using modern numerical simulation tools (CAE & CAD). It presents the results of a static and a dynamic analysis (fatigue) as well as experimentation, validating these simulations, with the ultimate goal of improving the operating time of the tested pump.*

Keywords: *External gear pumps, CAE & CAD, numerical simulation, fatigue, dynamic analysis.*

1. Introduction

External gear pumps are the simplest and most common oil hydraulic pumps operated by an electrical motor. Their success is accounted for by several advantages, like their extreme lightness, mechanical simplicity, wide-range viscosity, optimum suction, their wide range of flow rates, adaptability to any position and space and, finally, their cost, which makes them one of the cheapest types on the market. [1]

Yet, these pumps have some drawbacks too, i.e. a design focused on sustaining around 150 - 280 bar, their unsuitability for high flow applications, rather loud noise and a rather poor overall efficiency.

Nonetheless, the strong demand for these hydraulic generators prompts manufacturers to carry out research and improve their products by employing special materials, accurate heat treatments, minimum coupling tolerance between wheels and surface precise finish in order to reduce the drawbacks mentioned above.

External gear pumps (Figure 1) are essentially made up of two twin gearwheels geared with each other that are held in a totally smooth stator housing to prevent leakages between moving and fixed parts; the 8-shaped bearings (Figure 5), held in the stator along with gears, counterbalance side hydraulic thrusts by means of dedicated seals. [2]

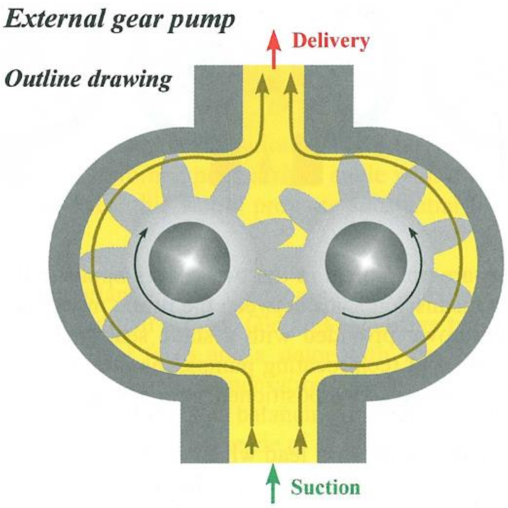

Fig. 1. External gear pump [2]

Axial and radial compensations are made possible by placing two balancing or compensation bearings opposite the plane faces of the gearwheels (Figure 2) and axially floating between the covers and the gears themselves (Figure 3).

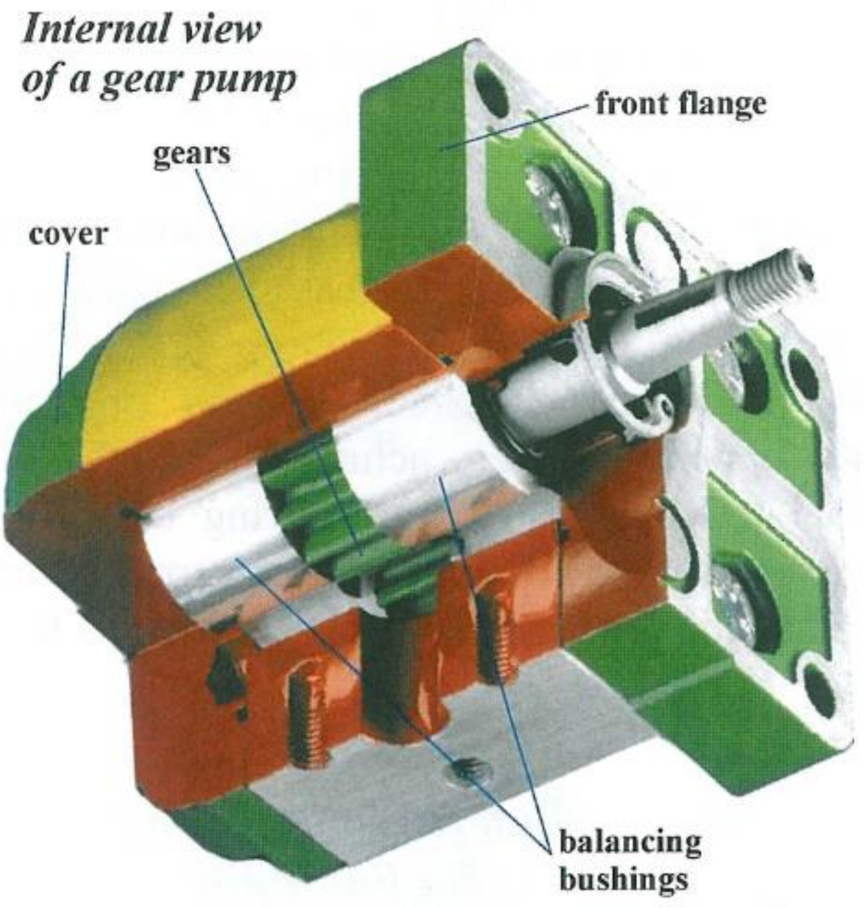

Fig. 2. Internal view of a gear pump [2]

In axial and radial balance (Figure 4), the pressurized fluid, pushed through tiny and accurately measured openings between the outlet and the bushings, exerts a thrust on their two back parts (cover sides). This keeps them solidly connected to the wheel but it allows an adequate lubrication thanks to an accurate design of the thrust force. As a result, gears perfectly mesh with bushings while the lubricating film prevents the faces of the parts from wearing out; the spindles solidly connected to the gearwheels do not need more bearings and the single bearing, if used, is positioned over the end of the transmission shaft.

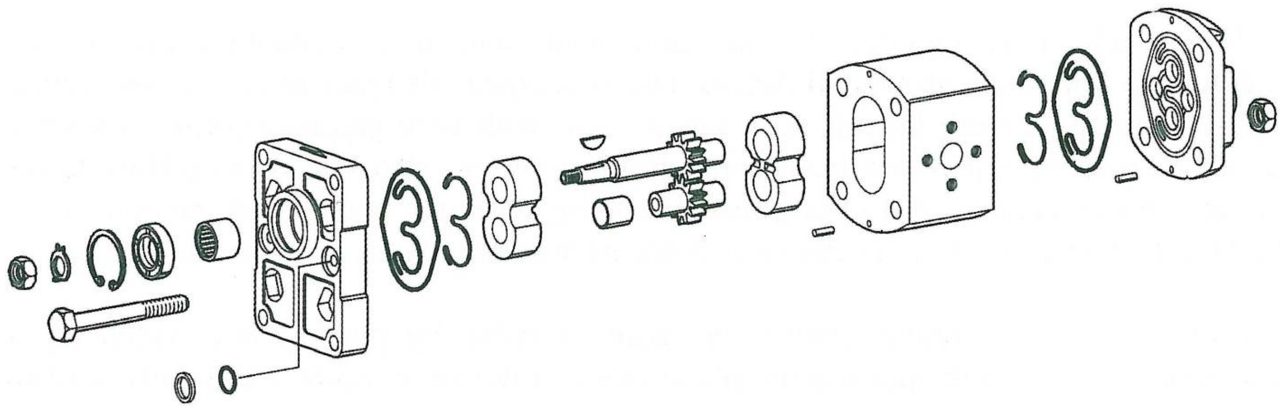

Fig. 3. Exploded view [2]

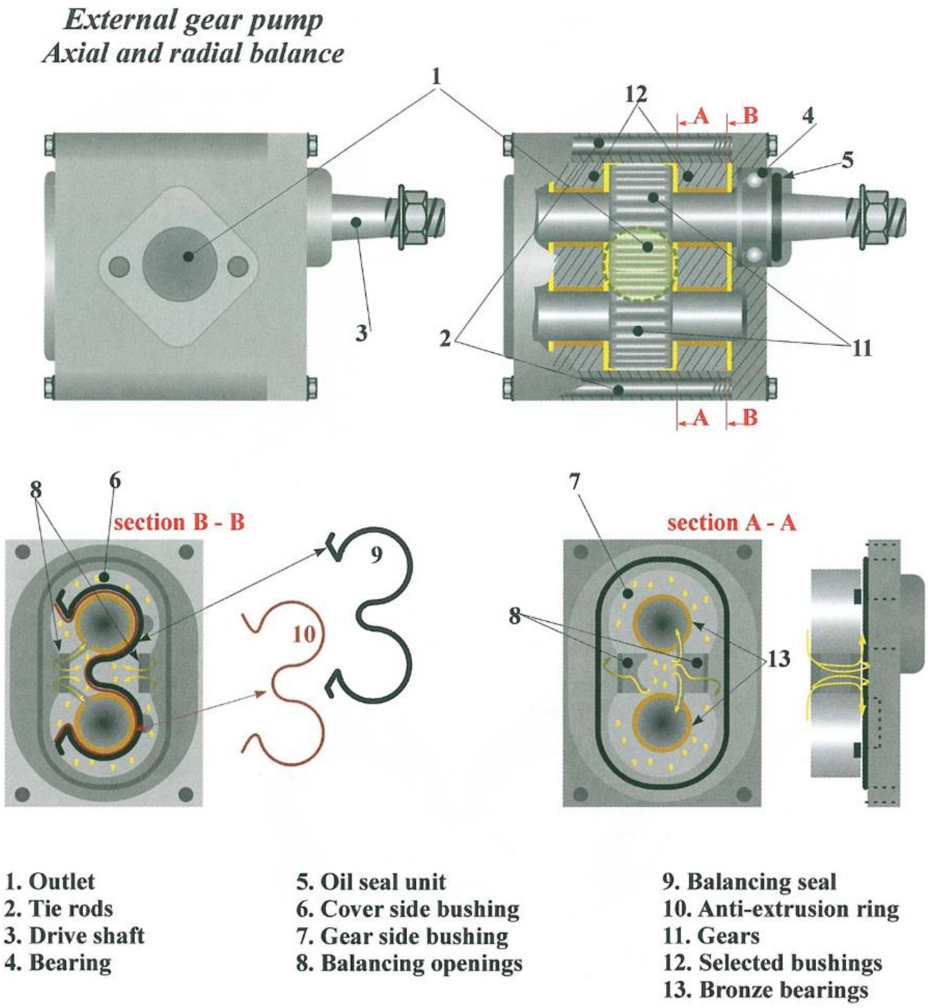

Fig. 4. Axial and radial balance [2]

Radial balance design must also consider the fact that teeth in mesh cannot fully expel oil. As a result, tiny fluid drops are 'squeezed' between the engaging wheels, thus entailing (depending on the incompressibility of the liquid) local overpressures that act in a radial manner vis-a-vis gear shafts. In addition, during gear disengagement, the volume between the teeth suddenly increases even before the contact with the sucked fluid.

Consequently, the central part of bushings must have some interstices in order to discharge this fluid; these are the only points where delivery and suction areas come into contact and the overpressure-prone fluid discharges in micro-areas subjected to early vacuum.

Fig. 5. Cover side bushing [2]

Bushing openings (Figure 5) play another important role as they allow leakages to pass from the delivery area to the suction area where the leaked fluid mixes with the fluid from the tank.

The “3-shaped” seal sets the balance area and separates the suction area from the delivery area. It is supported by an anti-extrusion ring, with the same shape as the seal, so as to avoid the extrusion of the seal parts where it is not supported due to clearance.

The pump leading gear is generally set to revolve clockwise; anticlockwise revolution occurs when the back cover is disassembled, and wheels are inverted (i.e. the leading wheel is replaced by the driven wheel and vice versa). [2]

2. Numerical simulation of pressure effects on the pump body

Given that type of pumps has radial and axial compensation, the forces discharging into the body are relatively small relative to the pressure action on the pump body. Therefore, the pressure action on the pump body will be analyzed using the CAE & CAD simulation software, Solidworks 2019.

2.1 Static analysis of the pump body

For static analysis with finite element, the 3D model was executed of the pump body (Figure 6) according to the execution documentation. The pump cap is not analyzed because for this construction it shows no significant deterioration but consider its presence during the simulation. The pump body shown below is made of an aluminum alloy which is poured under pressure into the mold.

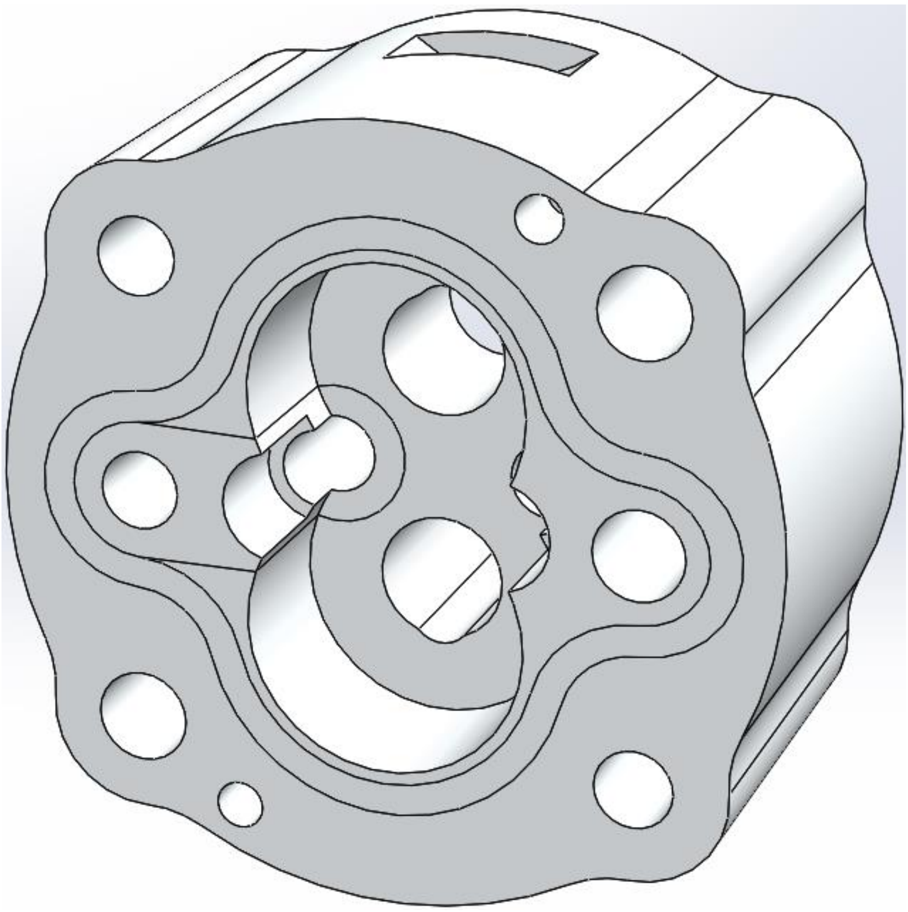

Fig. 6. The 3D model of the pump body

Figure 7 shows the mesh and the variation of its dimensions in the areas of interest, and in figure 8 the surfaces on which the pressure of 25 MPa acts are indicated with red arrows.

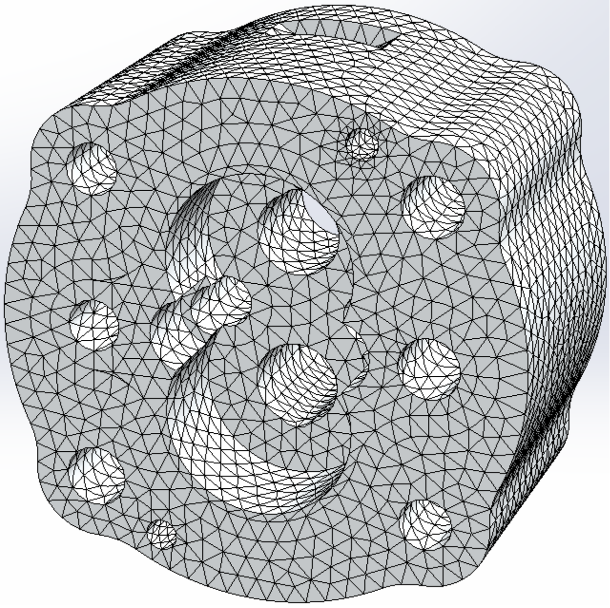

Fig. 7. The mesh

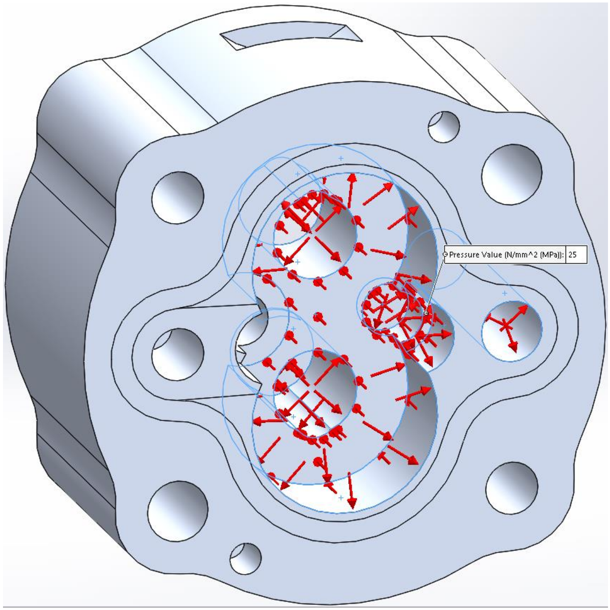

Fig. 8. Surfaces on which the pressure acts

Following the von Mises yield criterion analysis (Figure 9), the maximum stress on the body is 124 MPa, and it does not deform permanently up to a value of 140 MPa.

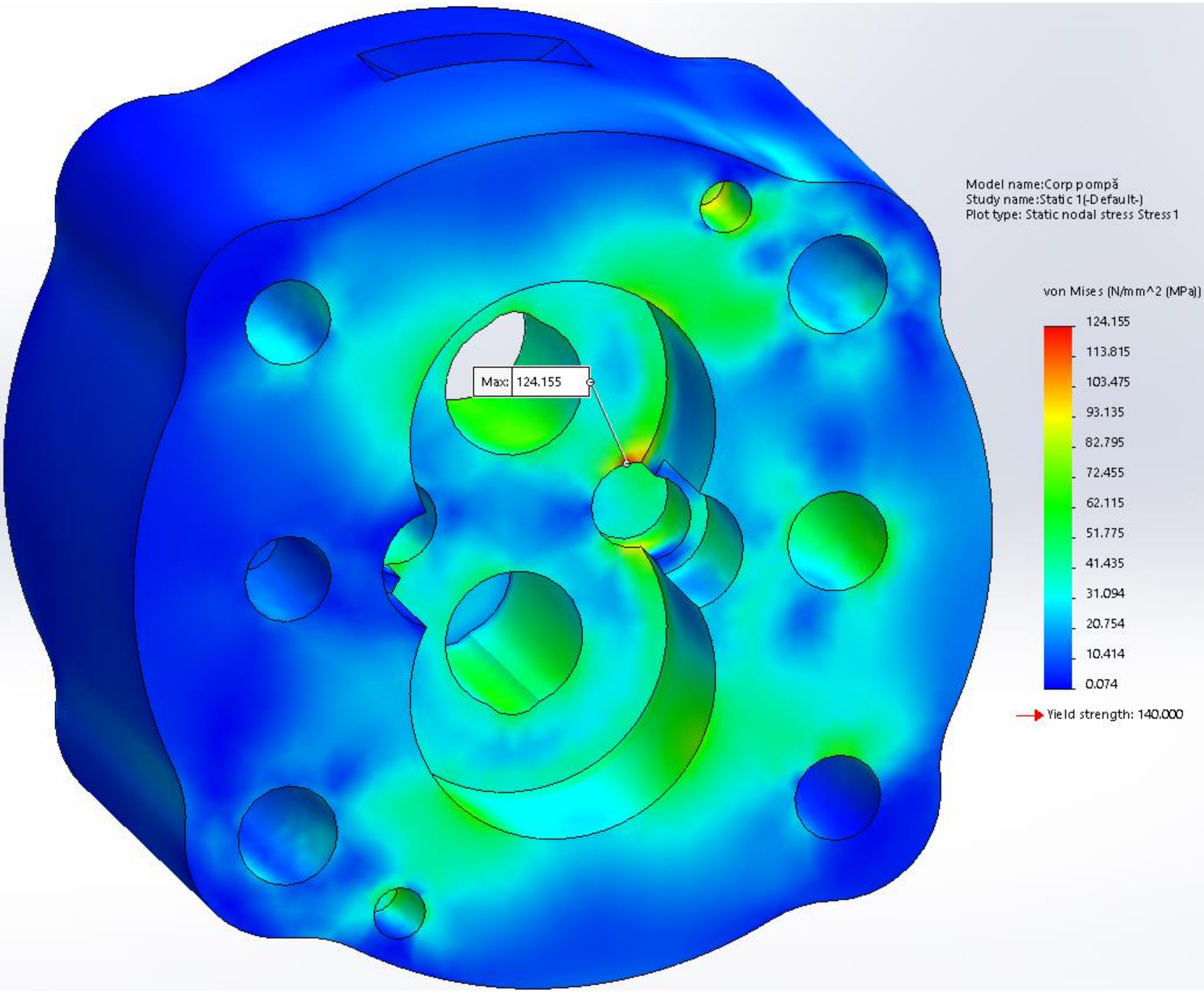

Fig. 9. The von Mises yield criterion analysis

Figure 10 shows a volume of 0.41% of the element's geometry of the pump body that is stressed with a value greater than 70 MPa.

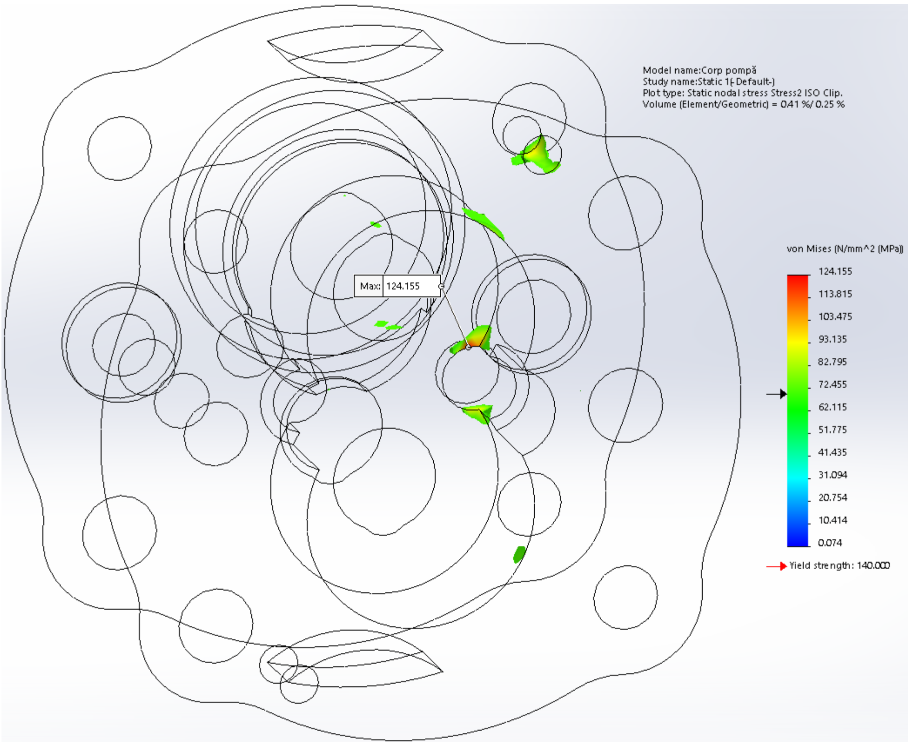

Fig. 10. The von Mises yield criterion analysis – Iso Clipping

Figure 11 shows the strain in the material of the pump body. On this diagram one can easily see the areas with the most mechanical demands.

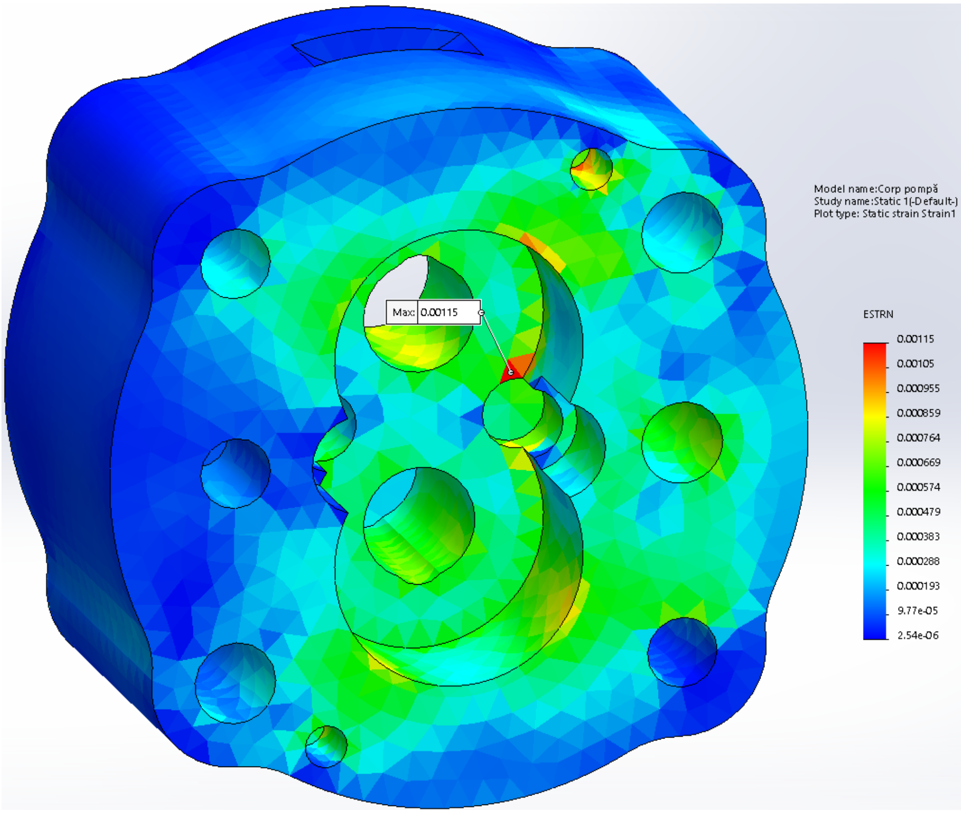

Fig. 11. The strain analysis

The deformations of the pump body are shown in Figure 12, where it can be seen that in the pump discharge zone the deformation has a maximum value of 0.0131 mm; on the same figure it can also be observed that the pressure deformation is not symmetrical compared to any main plane of the body, asymmetry due to the way the body cap is fastened.

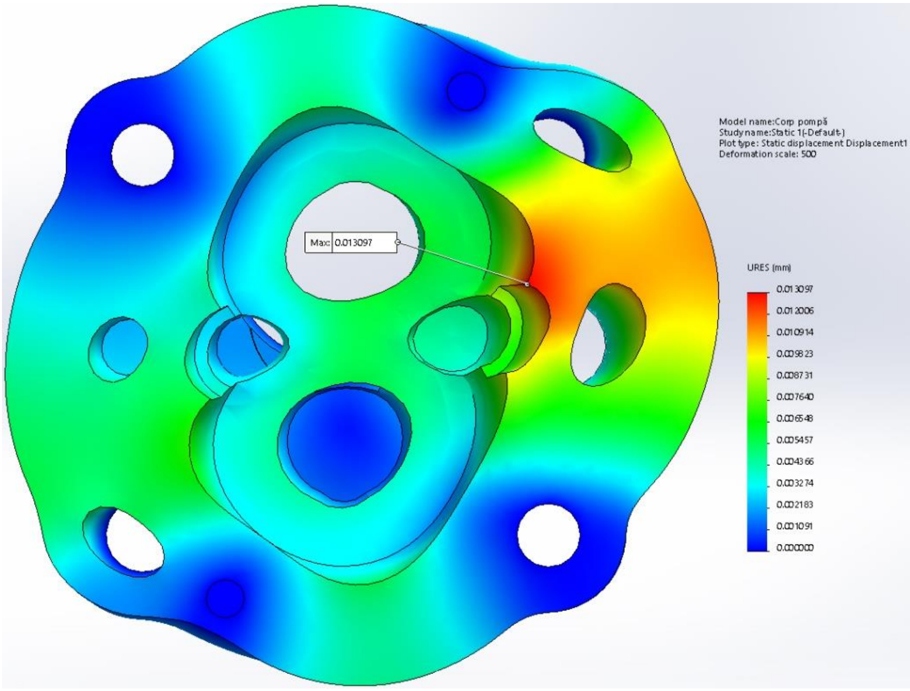

Fig. 12. The deformations of the pump body

After the static analysis of the pump body it can be concluded that it is well sized, no effort exceeds the critical value but there are minor problems with the stress concentrators in the pump discharge zone.

2.2 Dynamic analysis of the pump body

This subchapter is dedicated to the fatigue analysis of the pump body, the initial conditions of the simulation remain the same as in the previous subchapter, different from this is the load cycle, which in this case is pulsating-null cycle (R = 0), as exemplified in Figure 13.

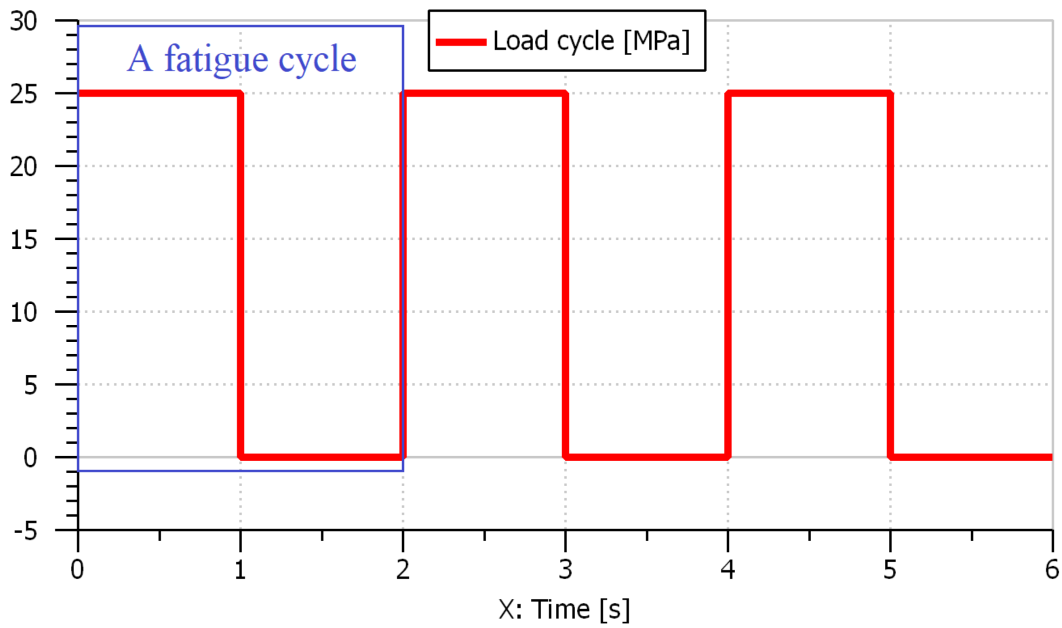

Fig. 13. The load cycles

After performing the analysis in Figures 14 and 15, the pump body resists along a number of 10.777.972 load cycles until in the discharge zone the destruction of the material reaches 100% where the material yields.

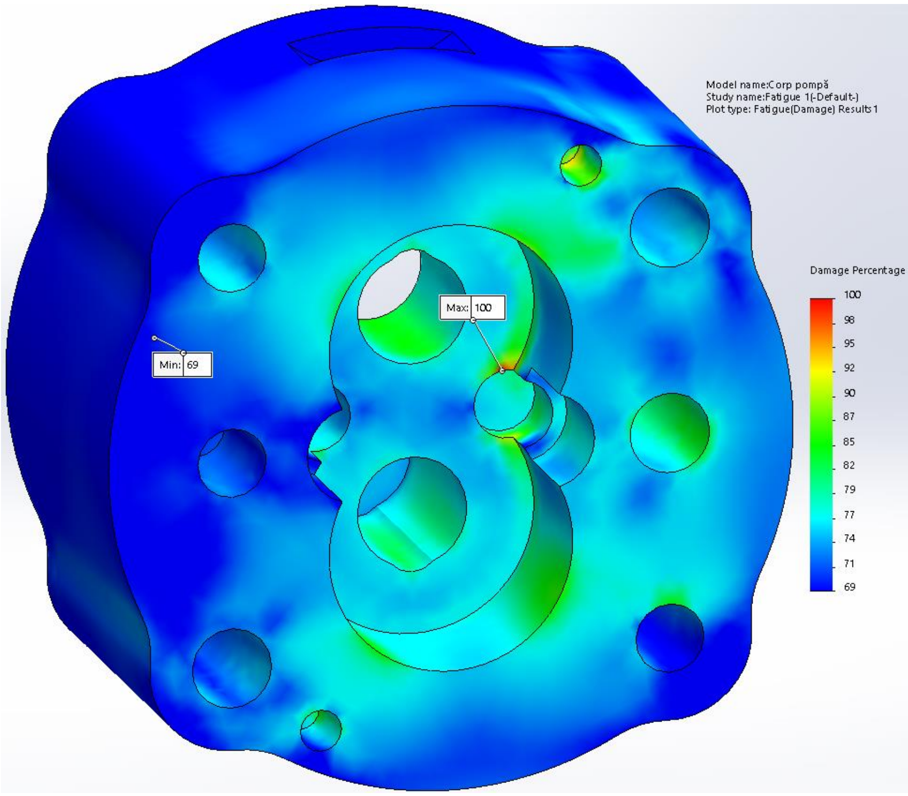

Fig. 14. Variation of material damage due to variation in pressure

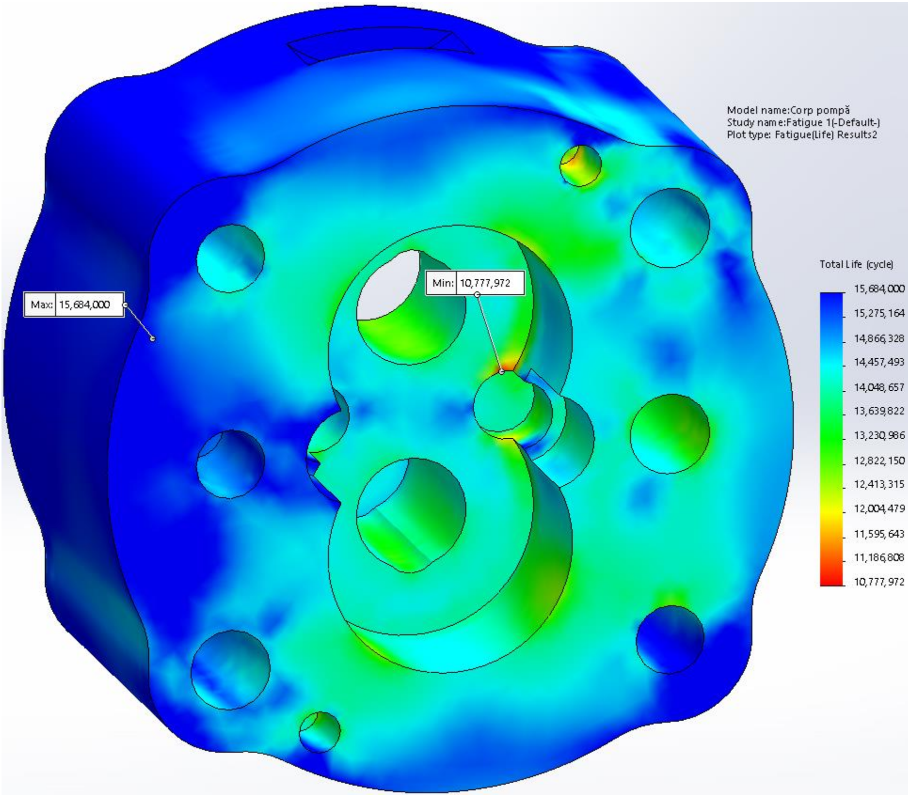

Fig. 15. Total life (fatigue)

3. Experimenting and validating numerical simulation results

Experimenting took place in the HESPER S.A. pump testing laboratory, where the pump was subjected to the same mechanical stress cycle as in Figure 13. The pump was driven at a speed of 3000 rpm for about 240 days.

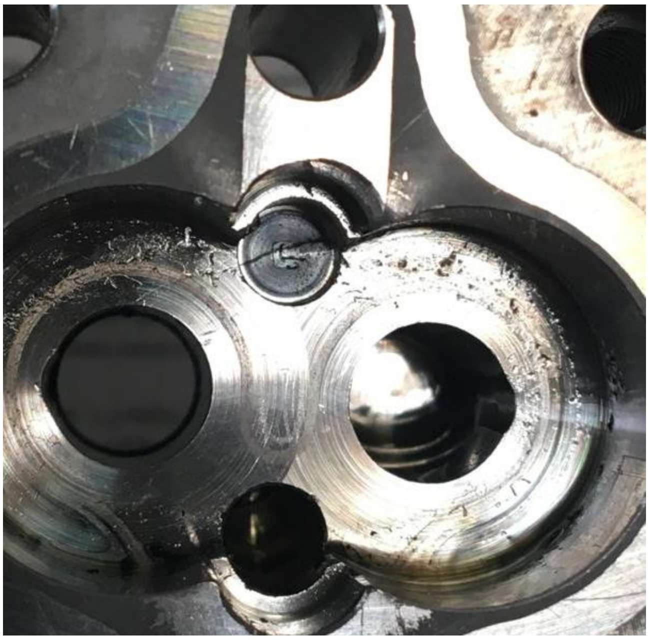

Fig. 16. The fissure

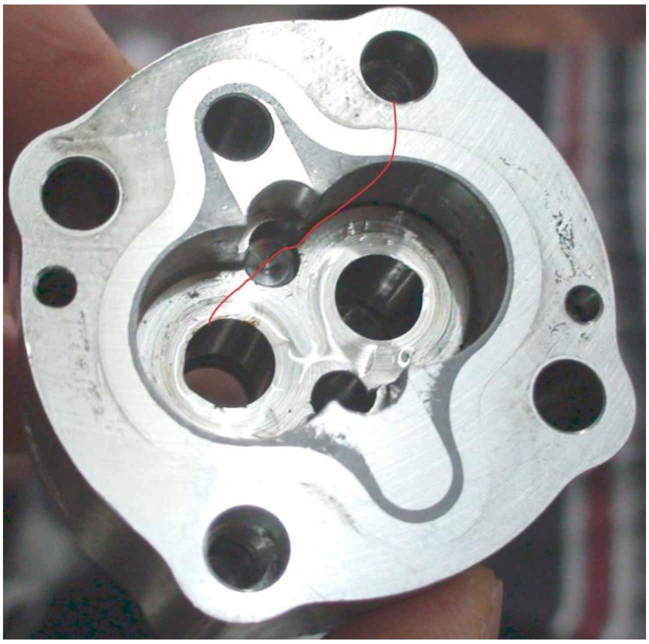

Fig. 17. The fissure route

After 10.413.376 cycles, a longitudinal fissure on the flange was observed in the discharge area, the crack (Figure 16 and 17) starting from the pump collar to the pump clamping screw (the crack is visible from the inside). When the pump was dismantled, it was found that the pump was working properly (Figure 18), the pinion shafts were free of wear, the gear toothing and the pump cover did not show significant wear. [3]

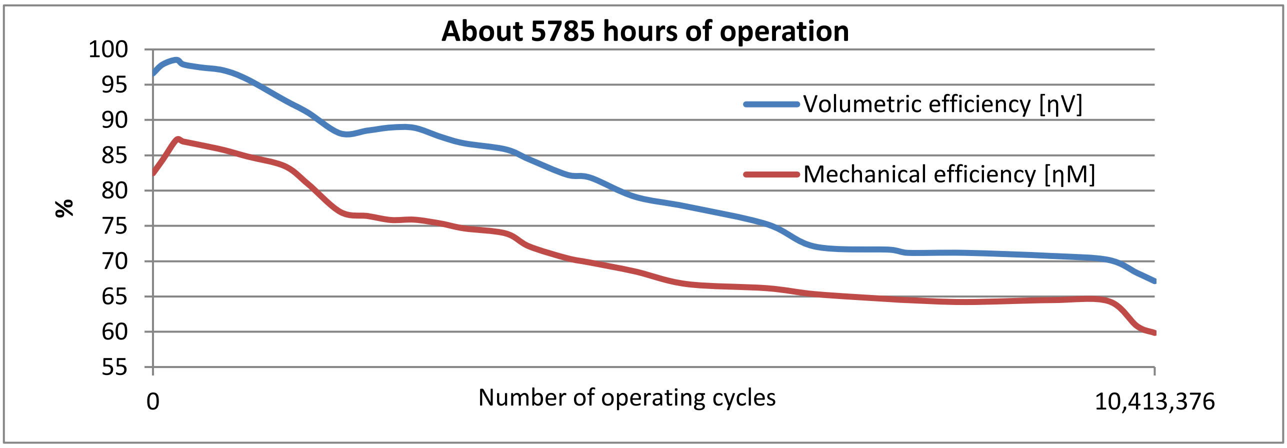

Fig. 18. The evolution in time of the pump's volumetric and mechanical efficiency

4. Conclusions

- It can be concluded that the numerical simulation is validated by the experimental results, with a difference of only 3.4%. Moreover, it can be noticed that both in simulation and in reality, the areas with stress concentrators coincide. The simulation can be used later to check different constructive variants to improve pump reliability.

- Most likely, the crack has emerged from the pump discharge zone due to a crack priming produced by mechanical machining by cutting and expanding outward. The development of this crack can be stopped by practicing a small diameter hole on the cracking route, or it would be much better if the machining in the troubled area would be done with a spherical head cutter that would reduce the number of stress concentrators and surface quality.
- The reliability of this hydraulic pump is very good, but there is still room for improvement in the future.

Acknowledgments

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