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# ADAPTIVE DESIGN OF A 3D MODEL MAGNETIC DRIVE MICROPUMP FOR AN EXTENDED LIFE CYCLE AND LOW MAINTENANCE

**Ionuţ Gabriel GHIONEA, Constantin Gheorghe OPRAN, Adrian Lucian GHIONEA, Saša ĆUKOVIĆ, Cristian Ioan TARBĂ**

***Abstract:*** *This paper presents preliminary research results on innovative and adaptive redesign of a magnetic drive micropump with helical gears. We have illustrated some issues of a gears micropump and particularly its worn components caused by the running process in its hydraulic installation. The design procedure and all the ideas involved in the research to improve actual performances relied on a relevant theories and hydraulic principles which affect the performance of the micropump. We have considered real industrial applications of magnetic drive micropumps, such as: textile and ceramic tiles printing, Diesel engines emission controls, fuel additives injection, etc. Accordingly, the micropump has to deal with corrosion inhibitors, lubricants, fuel, anti-icing additives, dyes, detergents, static dissipating additives, so its wear status drew our attention towards an improved and innovative version with the same or higher volumetric efficiency and flow displacement. Our research procedure highlights the competitive advantages of innovative 3D design with technological parameters, numerically simulated, proving the correctness and advantages of the new design variant. Presented innovative CAD solution allows us to develop a series of micropumps, compact, robust, reliable and with low manufacturing costs.*

***Key words:*** *magnetic drive gear micropump, innovative CAD model, constructive solutions.*

## 1. INTRODUCTION

The hydraulic pumps, in all their variety, represent an economic and technical requirement for researching, designing and manufacturing projects in the industrial and academic environments. Important applications in aerospace, automotive, medical systems, energy plants, electrotechnics, food supply and refrigerate systems, consumer goods, etc., need special hydraulic pumps, with a complex factor of innovation and variable functional parameters adapted to custom requirements [1, 2]. These pumps drive a specific fluid characterized by: flow, pressure, viscosity, presence of particles, for applications like: fuel transfer, injection and/or mixing, recirculation in a circuit, dosing of substances with medical applications, mixing of additives or chemicals, printing on textiles and ceramic tiles, etc. Even they should fulfill complex requirements in their hydraulic installations, these magnetic drive pumps should also be of constructive simplicity, reliability, lack of leakage to the outside, low weight, small size and a wide range of applicability.

The authors of the paper carry out research in the GEX UPB project 57/2017 for the redesign and development of an innovative experimental prototype of a magnetic drive micropump for an extended life cycle and low maintenance.

In the research, the authors want to implement a new gears pair with a different module, polymeric composite materials for the gears, but also for the new friction disc and the aspiration component in order to reduce the wear that emerges from the micropump operation.

The research proposed by this paper raises a number of challenges, such as: conception of an improved micropump variant using 3D models of the components and of the assembly, numerical and technological calculations and simulations [3, 4].

There are some risks linked to the new variant: the magnetic field creation system required to drive the micropump may be different from the original system; high costs for manufacturing, experiments, validation and implementation of the new model; certain constructive and functional requirements (adjustments, tolerances of the pump components) may not be fulfilled, leading to a decrease of the prescribed pressure and flow; the modified pair of gears proposed by research may be inappropriate for generating the fluid flow with high efficiency [5, 6, 7]. Also, the choice of the polymeric composite materials may require laborious injection molding research, there could appear difficulties in balancing the resulting forces from pump operation etc.

## 2. HOW DO THE MAGNETIC DRIVE PUMPS WORK?

Generally, the magnetic drive pumps are energy efficient and require no complex seals or lubricants for operation. They use a wide variety of fluids including chemicals, acids, water, lubricants, fuels, anti-icing additives and even biological. Because there is no mechanical seal on a magnetic driven pump, the chance of external leaks or contamination of the internal fluid from an external source is eliminated.

In a magnetic drive pump, there are two main types of magnets. Permanent magnets are attached to the pump assembly, as seen in Fig. 1 [8]. The driving magnet, which drives the inner driven magnet, is attached on a second shaft operated by an electric motor [1, 8]. When this motor turns in a certain direction, it spins the driving magnet. As a result, the created magnetic field from the motor's magnet causes the driven magnet connected to one gear to spin and rotate the other gear(s).

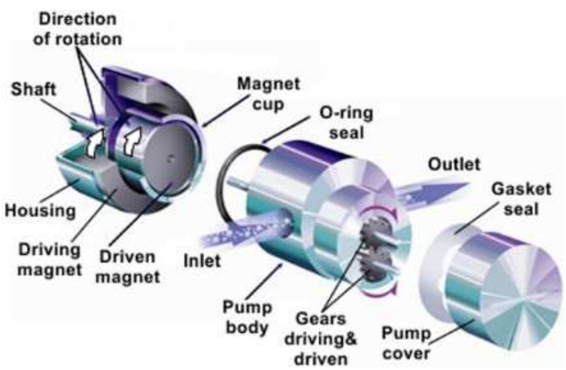


**Fig. 1.** Main components and inside view of a gear pump with magnetic drive

The magnetic field created by these two magnets allows the full torque of the electric motor to be transmitted to the pump. As the shaft of the motor does not extend into the interior of the pump, there is no need for a shaft seal ring, which is a very important aspect and a characteristic of magnetic pumps. Without it, the probability of leakage, commonly encountered in other classic hydraulic pumps, due to the wear of the shaft seal ring is eliminated.

Generally, the maintenance cost is reduced because it is not necessary to verify and maintain the shaft seal ring. However, periodically, a check of the pump is necessary because the wear may occur on the moving components and with the other fixed components with role of closure and support (the pump body, the driving gear shaft).

The resulted rotation produces a pressure that drives the fluid aspired by the pump through the inlet to the outlet apertures, outside the pump's housing

## 3. MICROPUMP'S CURRENT STATUS

The paper presents results of the wear evaluation of a hydraulic gear micropump assembly with helical teeth. The applied diagnostic considers constructive and functional particularities, but especially data of the cylindrical gear and of the micropump body components, used materials, the quality of the sliding and contact surfaces, the gears holes fits with their rotation and support shafts.

This micropump (fig. 2) has some worn components. The wear has been observed at the arrival of the micropump at our university Composite Products Laboratory provided by the service department of the Axflow Romania. The pump is withdrawn from the service, its type is GB-P25.JF5S.A, by Micropump Inc. [8].

According to the manufacturer specifications, the pump flow rate varies between $Q_p$ = 1.65 - 3.1 l/min corresponding to the drive speed variation of $n$ = 2850 - 5500 rev/min.

In operation, the micropump can generate a maximum pressure of 8.6 bar (125 psi), and it can be integrated into a hydraulic system where the maximum pressure is 21 bar (300 psi). The temperature during its operation may vary in the range of -46$^0$ to 177$^0$ C. The viscosity of the hydraulic medium is 0.5 - 1500 cps.

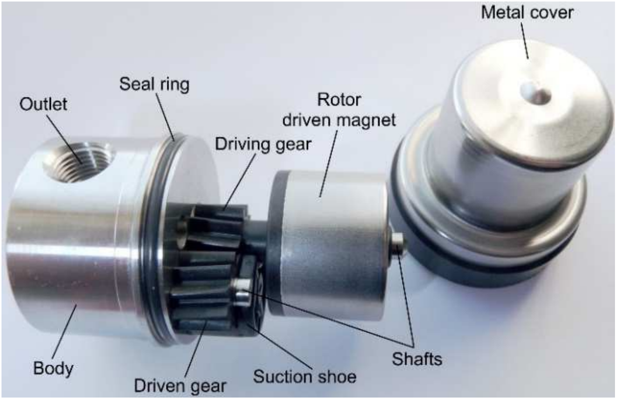


**Fig. 2.** Magnetic drive micropump in our laboratory

Applying maintenance to the analyzed micropump requires knowledge of the above mentioned and constructive parameters. It is necessary to evaluate the main functional conditions, as follows: the noise level, the decrease of the efficiency, proposals for some constructive changes.

Other requirements are also important, such as: calculating the forces in the gears and the rotation couplers allowable stress, choosing the gears material, determining the deviations size of shape and of position for the surfaces that definine each pump component.

The micropump analysis method is based on direct measurements of its components and the application of specific modeling and simulation techniques [9, 10, 11].

The analyzed pump shows some wear on its components, the most affected are the body and the gears shafts. Figure 3 illustrates the wear distribution and its results generated by the functioning. There are noticeable wears on the body's flat surface in the contact spot with the gears and with the suction shoe element.


**Fig. 3.** Worn areas on the micropump body

The areas with the most pronounced wear (W1 and W2) are located on the body's flat surface around the two shafts: under the driving and the driven gears.

Following the pump's driving under laboratory conditions and carrying tests with fluid's viscosity characteristics in the range prescribed by the manufacturer, it was noted an increase in the drive torque (by 12-18%) and a pressure variation.

There was also an increased wear of the pinion's shaft due to the forces in the gear. By wearing the flat surface of the micropump body and the shafts in the contact areas with the gears, there is no longer a condition of parallelism between the gears shafts, the increase of the radial and frontal clearances. The observed wear has led to increased gearing forces and to a deficient contact between the teeth flanks. At the same time, there were remarked vibrations; noise and heating were increased inside the micropump.

The pump has a suction shoe in the suction area (inlet aperture) to guide the fluid flowing into the micropump to be placed in the gaps/cups between the teeth, driven and discharged into the hydraulic system through the outlet aperture [8]. We observed a wear zone (W3) in the contact area between this suction shoe and the flat surface of the body. This is due to the vibrations resulted from the micropump operation.

The results of the functional analysis of the studied micropump revealed that the suction element has created a clearance on the vertical direction.

Figure 4 represents the 3D model of the micropump in the actual constructive solution shown in Fig. 2. Its assembly was parametrically modeled in CATIA v5 on the basis of laboratory measurements and operating requirements.

The main component of the assembly is the body in which the two aspiration and discharge holes are diametrically created, corresponding to the inside flat surface. The holes for mounting and supporting the gears shafts and the suction shoe element are also identified.

Inside the body, mounted at the end of the pinion shaft, there is placed a helical spring used to lock this shaft when the micropump assembly is closed with the metal cover [1, 8].

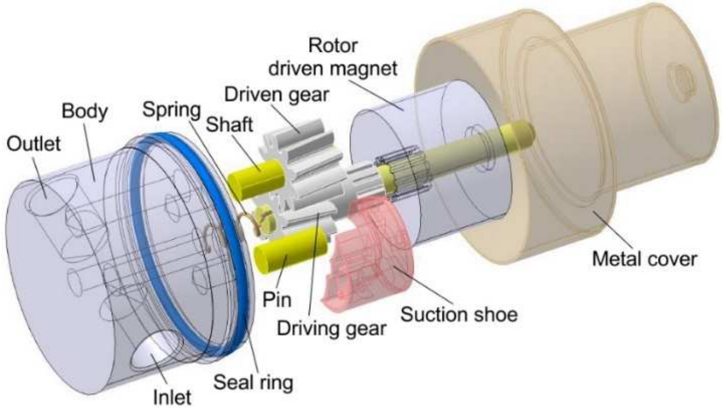


**Fig. 4.** 3D model of the actual micropump

The micropump gears are driven by the rotor magnet, assembled and oriented on the pinion shaft through a splined rigid coupling. The fluid aspiration into the micropump is done through the suction shoe element. The assembly closes with a metal cap, which is sealed with the body using a seal ring.

Both from the 3D model of Fig. 4 and the representation of Fig. 5 it is identified the shape and position of the suction shoe element with respect to the two gears and the inlet and outlet apertures. This element is placed on the short shaft of the driven gear and also contributes as a part of the pinion shaft bearing.

During operation, the fluid is aspired into the micropump, filling its entire cavity, from where it is discharged into the hydraulic system. The fluid stored in the cavity is permanently driven by the two gears and thus creates a flow of fluid which produces a decrease of the discharge pressure, an additional stress on the moving parts and an increase of the necessary driving torque [12, 13].

From a constructive point of view, the number of teeth (Fig. 5) between the aspiration and discharge areas is also important, with the role of sealing the fluid. An important role in the performance of the micropump is done by the radial clearance between the driven gear and the inner wall of the metal cap. This ensures that a certain amount of fluid corresponding to a set radial clearance is in the area between the head of the teeth and the suction shoe. They influence the efficiency of the micropump.

Also, from a functional point of view, the decrease of flow (and efficiency) is also determined by the front-end clearance and the size of the front surface bounded by the diameters of the two gears in engagement and the suction shoe element. This is the area where the required pressure for the fluid aspiration is created.

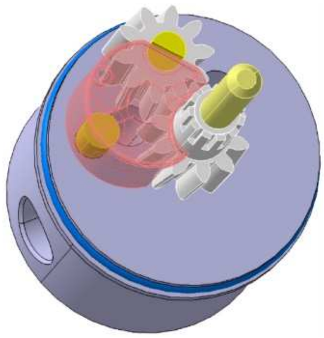

**Fig. 5.** Actual constructive solution using the suction shoe element

In the micropump's cavity, in areas with a lower pressure, accumulating residues from the hydraulic medium can contribute to the wear of the moving parts. They may be trained and partially obturate the discharge port and the gear, or may be deposited causing damage to the micropump and to the hydraulic system.

## 4. PROPOSED MODEL OF THE NEW MICROPUMP

Our research led us to propose some innovative and constructive solutions to improve the characteristics of the micropump. The most important modification was applied to its gears, the original pair to be replaced. The new gears will also be helical and corrected (displaced) and with another module size. This change results in an increase of the pump flow, keeping the same distance between the gears shafts and the same drive speed [14].

The modification of the two gears in engagement (Fig. 6) required some specific calculations based on the literature [15, 16, 17], their 3D modeling in the frame of the assembly.

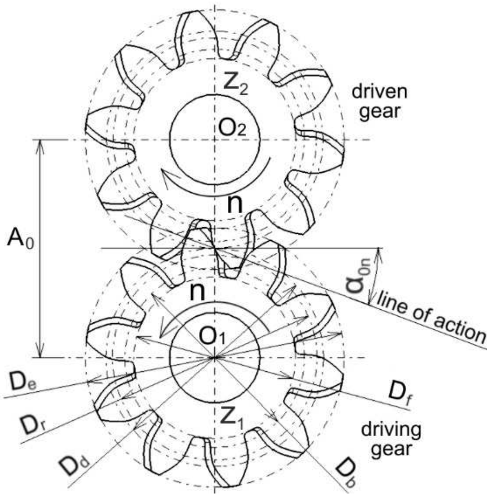


**Fig. 6.** Geometrical elements of the cylindrical gear with helical teeth in the frontal plane

Considerations were given to conditions of engagement and possible interferences between the redesigned gears and the other components in the available space of the micropump cavity.

For the representation phase of the micropump assembly, the following values of the constructive parameters were set (Table 1), namely: $z$ - number of teeth, $m$ - teeth module [17, 18, 19], $\xi$ - correction parameter, $\beta_0$ - reference angle of inclination of teeth (helix angle), $\alpha_0$ - engagement angle, $b$ - gears width, $A_0$ - center distance, etc. The gears teeth flank line is a constant-pitch cylindrical helix. Both gears must have the same helix angle. However, the helix direction must be opposite, a left-hand helix mates with a right-hand helix.

The option to maintain a cylindrical gear with helical teeth with involute and displaced profile provides some functional advantages, the most important being: reducing the wear of the gears teeth, noise, vibration and pulsation. In the initial study phase, we also considered for analysis a spur gears variant of the micropump.


**Fig. 7.** Representation of geometric elements and forces of cylindrical gear: [$P_N$] – normal plane, [$P_F$] – frontal plane, $F_t$ – tangential force, $F_a$ – axial force, $F_r$ – radial force, $F_n$ – normal force, $G_{cd}$ – generator of the pitch circle cylinder, flanks line: $E_{cd}/E_{cs}$ – cylindrical helix right-hand/left-hand, $n_1 = n_2$ – speeds of the meshing gears, $z_1 = z_2$ – number of teeth of each gear, *pc* – pitch circle of virtual gear, *epc* – elliptical pitch circle in normal plane [$P_N$], $r_{e1}$ – radius of the equivalent gear $z_1$

Helical gears operate more smoothly and quietly compared to spur gears due to the way the teeth interact [15, 19, 20]. The teeth on a helical gear are cut at an angle to the face of the gear. When two of the teeth start to engage, the contact is gradual starting at one end of the tooth and maintaining contact as the gear rotates into full engagement. Tooth strength is improved because of the elongated helical wraparound tooth base support. Contact ratio is increased due to the axial tooth overlap.

Helical gears thus tend to have greater load carrying capacity than spur gears of the same size. Spur gears, on the other hand, have a somewhat higher efficiency.

While the micropump is active, the use of the helical teeth determines the creation of the following forces [15, 21]: $F_t$, $F_a$, $F_r$ and $F_n$, represented in Fig. 7. To determine their values, it is necessary to set the parameters defining the micropump operation, namely: flow, pressure, torque and power of the electric drive.

The drives used by Micropump Inc. are innovative brushless DC motors with no moving parts for great efficiency and reliability. They are fitted with variable speed operation, a tachometer and error output, and with fwd/rev capabilities [8], being specific to the operation of these hydraulic micropumps.

The forces values are very important in the dimensioning of the gear teeth, with reference to the number of teeth, the normal and frontal modules, gears width, teeth length, inclination angles in the frontal and normal planes, etc. (Tab. 1), precision grade (recommended 5/6), deviation of the tooth direction, the chosen material and the quality of the teeth flanks [22, 23].

The parameters [14, 16, 19] noted in Table 1 correspond to those in Fig. 6 and 7. Increasing the number of teeth increases the coverage grade, reduces vibration and noise, reduces slipping in gearing, increases efficiency and reduces wear.

*Table 1.*

**Calculus stages and basic data of gears with helical teeth.**

| **Parameter** | **Calculus relation** | **Value** |
|---|---|---|
| $z_1$, $z_2$ – number of teeth for each gear | chosen values | $z_1 = z_2 = 10$ teeth |
| $\beta_0$ – the reference angle of the teeth inclination. It is the angle between the tooth line tangent and the cylinder generator with pitch diameter $D_d$ | recommended value | $\beta_0 = 8^0 40' = 8.67^0$ |
| $m_n$ – normal module of the gear | standardized value | $m_n = 1.125$ mm |
| $m_f$ – frontal module of the gear | $m_f = \dfrac{m_n}{\cos\beta_0}$ | $m_f = 1.138$ mm |
| $\xi_{n1}$, $\xi_{n2}$ – the normal specific displacement of the teeth profiles | chosen value | $(\xi_{n1} + \xi_{n2}) > 0$ positive specific displacement<br>$\xi_{n1} = \xi_{n2} = 0.2$ mm |
| $\alpha_{0n}$ – normal angle of the reference profile, pressure angle | standardized value | $\alpha_{0n} = 20^0$ |
| $\alpha_{0f}$ – frontal angle of the reference profile, operating pressure angle | $\tan\alpha_{of} = \dfrac{\tan\alpha_{0n}}{\cos\beta_0}$ | $\alpha_{0f} = 20^0 13' = 20.22^0$ |
| $p_{0n}$, $p_{0f}$ – the normal/frontal pitch of the reference rack | $p_{0n} = \pi \cdot m_n$ ; $p_{0f} = \pi \cdot m_f$ | $p_{0n} = 3.534$ mm;<br>$p_{0f} = 3.575$ mm |
| $\xi_{f1}$, $\xi_{f2}$ – specific frontal displacement of the profiles | $\xi_{f1,2} = \xi_{n1,2} \cdot \cos\beta_0$ | $\xi_{f1,2} = 0.1976$ mm |
| $A_0$ – reference distance between the gears shafts | $A_0 = 0.5 \cdot m_f \cdot (z_1 + z_2)$ | $A_0 = 11.38$ mm |
| $\xi_{0f}$ – the specific frontal coefficient of displacement | $\xi_{0f} = 2 \cdot \dfrac{\xi_{f1} + \xi_{f2}}{z_1 + z_2}$ | $\xi_{0f} = 0.039$ mm |
| $\alpha_{rf}$ – frontal engagement angle | $\mathrm{inv}\,\alpha_{rf} = \xi_{0f} \cdot \tan\alpha_{of} + \mathrm{inv}\,\alpha_{0f}$ | $\alpha_{rf} = 24^0 55' = 24.92^0$ |
| $A$ – distance between the gears shafts | $A = A_0 \cdot \dfrac{\cos\alpha_{0f}}{\cos\alpha_{rf}}$ | $A = 11.77$ mm |
| $D_{r1,2}$ – working pitch diameter | $D_{r1,2} = m_f \cdot z_{1,2} \cdot \dfrac{\cos\alpha_{0f}}{\cos\alpha_{rf}}$ | $D_{r1,2} = 11.72$ mm |
| $D_{d1,2}$ – pitch diameter | $D_{d1,2} = m_f \cdot z_{1,2}$ | $D_{d1,2} = 11.38$ mm |

| | | |
|---|---|---|
| $D_{e1,2}$ – outside diameter | $D_{e1,2} = D_{d1,2}+2\cdot m_n\cdot\left(1+\xi_{n1,2}\right)$ | $D_{e1,2}$ = 14.08 mm |
| $D_b$ – base diameter | $D_b= D_d\cdot \cos\alpha_{0n}$ | $D_b$ = 10.69 mm |
| $D_f$ – root diameter<br>$h_a^*$ – addendum<br>$c^*$ – radial reference clearance coefficient | $D_f = D_d - 2\cdot m_n\cdot\left(h_a^* + c^* - \xi_{n1}\right)$ | $h_a^*$ = 1.1, $c^*$ = 0.4<br>$D_f$ = 8.45 mm |
| $b$ – gears width<br>$l$ – the teeth length (cylindrical helix) on the pitch circle | $l = \dfrac{b}{\cos\beta_0}$ | $b$ = 8.9 mm;<br>$l \approx$ 9.05 mm |
| $n_1 = n_2$ – speed of the meshing gear<br>$M$, $P$ – torque and power of the electric drive<br>$F_{t1,2}$ – tangential force<br>$F_{r1,2}$ – radial force<br>$F_{a1,2}$ – axial force<br>$F_{n1,2}$ – normal force | $P = \dfrac{M\cdot n}{9550}$<br>$F_t = \dfrac{M}{D_d/2}$<br>$F_r = F_t\cdot\tan\alpha_{0n}$<br>$F_a = F_t\cdot\tan\beta_0$<br>$F_n = \dfrac{F_t}{\cos\alpha_0\cdot\cos\beta_0}$ | according to the Table 2 |
| $q_p$ – the specific capacity equal to the total volume of the two gears couplings in one rotation<br>$Q_p$ – micropump's flow | $q_p = 2\cdot\pi\cdot l\cdot m^2\cdot z$<br>$Q_p = q_p\cdot n$ | $q_p$ = 720.6 mm³/rev<br>$Q_p$ = 4 l/min<br>($n$ = 5550 rev/min) |

According to [2, 8, 15], the $M$ and $P$ values are indicated and determined in dependence to viscosity and pressure of the trained hydraulic medium. Using the calculation relations results values for the gearing forces, shown in Table 2. It is noted that the obtained values for the micropump's operating parameters are considered to be optimal and in accordance with its purpose and fields of use, the speed range, the hydraulic environment viscosity, the delivered flow rate and the reduced gear dimensions.

The basic parameter of the micropump is the specific flow $q_p$, also called specific capacity, equal to the total volume of the two gears cups on one rotation and the theoretical flow, which is determined by calculation relations and checked by measurement under laboratory conditions.

*Table 2.*

**Calculated values for the drive's necessary power and forces in gear**

| No. | Viscosity [cps] | Speed [rev/min] | Torque [mN×m] | Drive power [W] | $F_t$ [×10⁻³ N] | $F_r$ [×10⁻³ N] | $F_a$ [×10⁻³ N] | $F_n$ [×10⁻³ N] |
|---|---|---|---|---|---|---|---|---|
| 1 | 1 (water) | 2850 | 70 | 20.89 | 12.28 | 4.47 | 1.87 | 13.22 |
| 2 | 1 (water) | 4000 | 120 | 50.26 | 21.05 | 7.66 | 3.21 | 22.40 |
| 3 | 1 (water) | 5550 | 160 | 92.98 | 28.07 | 10.22 | 4.28 | 29.87 |
| 4 | 100 (oil) | 2850 | 140 | 41.78 | 24.56 | 8.94 | 3.75 | 26.14 |
| 5 | 100 (oil) | 4000 | 180 | 75.39 | 31.58 | 11.49 | 4.82 | 33.61 |
| 6 | 1500 (oil) | 1150 | 230 | 27.70 | 40.35 | 14.69 | 6.15 | 42.94 |

The calculation of the theoretical flow rate for the proposed gear is detailed in several specialized papers [13, 24, 25].

The actual flow of the micropump differs from the theoretical flow, being influenced by several factors [6, 25]. The most important are: flow decrease due to frontal clearance (recommended 20 - 150 μm) and to the radial clearance (recommended 10 - 35 μm), hydraulic fluid viscosity, admissible wears of the gears contact surfaces with the body's flat surface (max 0.08 mm) [10].

According to papers [6, 14], to determinate the micropump's flow it is considered that the front surface of a gap between two adjacent teeth is approximately equal to the surface of one tooth in the same plane.

In order to evaluate the specific flow rate of the re-designed micropump with a new gear pair, there were considered [15, 26, 27] the following data extracted by measurements from the 3D model: section of the gap in the normal plane (4.606 $mm^2$), and in the frontal plane (4.727 $mm^2$), the length of the cylindrical helix considered on the pitch cylinder of the gear (9.05 mm = the tooth length on helix) corresponding to the gear width (8.9 mm). Considering these data, the geometric volume of a cup results in 41.7 $mm^3$, so the calculated flow rate of the micropump is $Q_p$ = 2 l/min ($n$ = 2850 rev/min) - 4 l/min ($n$ = 5550 rev/min).

Compared to the specific capacity of the current/actual micropump ($q_{pa}$ = 568.6 $mm^3$/rev [8]), we achieved an increase of 22.22% to $q_p$ = 720.6 $mm^3$/rev.

Another proposed change on the micropump assembly is the design of a new suction element (Fig. 8), which has the role of separating the aspiration zone of the fluid by the driving gear through the inlet port to the outlet port. Thus, a significant reduction in torque is required to drive the fluid from the entire cavity of the micropump. The fluid in the cavity is, however, still trained by the driven gear to ensure the lubrication of the other moving parts [23], but at a much smaller rate.

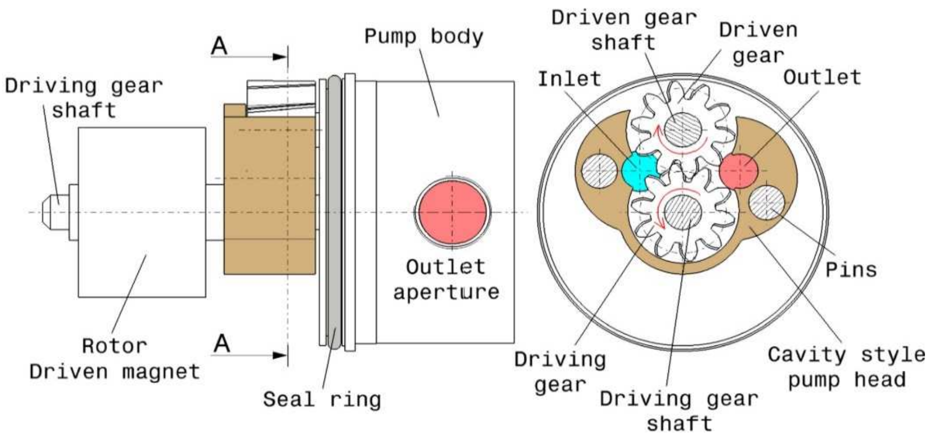


**Fig. 8.** View and position of the cavity style pump head

In case of using this new suction element, named cavity style pump head, a new hole has to be made in the pump body and a new support pin will be added. Thus, this element is oriented and supported by two pins and by the driving gear shaft, also having a bearing purpose for this shaft.

Due to its larger dimensions and to the new type of assembly, the suction element is less prone to vibration and radial clearance. The suction element has, of course, the two inlet and outlet pockets. This creates a maximum pressure due to the driving of the fluid by the driving gear [2, 21]. The fluid is in the cups between the gears' teeth and the new suction element.

The driven gear helps to create the necessary pressure between the two apertures of the micropump by driving the fluid also into the cups between the teeth and the metal cap. Any residuals or impurities in the micropump cavity are prevented from reaching back the discharge area due to the closed construction of this new suction element (Fig. 9).

The 3D model of the proposed micropump is completed by an assembled friction disc in contact with the flat surface of the body.

**Fig. 9.** Proposed constructive solution using the cavity style pump head element

The disc has all the holes necessary for mounting of the gears shafts, the two pins that support the new suction element and the inlet and outlet holes, corresponding to the two apertures. The disc has the role of taking over the wear that affects the body of the micropump in the current version, being a simple and easy way to replace the component during service and maintenance intervals.

Through its flat surface, the disc can also be used with micropumps that already have an acute wear of the body, increasing lifetime and operation in the hydraulic system.

Figure 10 shows another component added to the micropump assembly, a support disc for the end of the driving gear's shaft.

This second disc is assembled inside the metal cap and, through its central hole, guides and sustains the end of that shaft. This reduces its displacement from the already existing small guide on the metal cover.

The support disc is also a replaceable component.

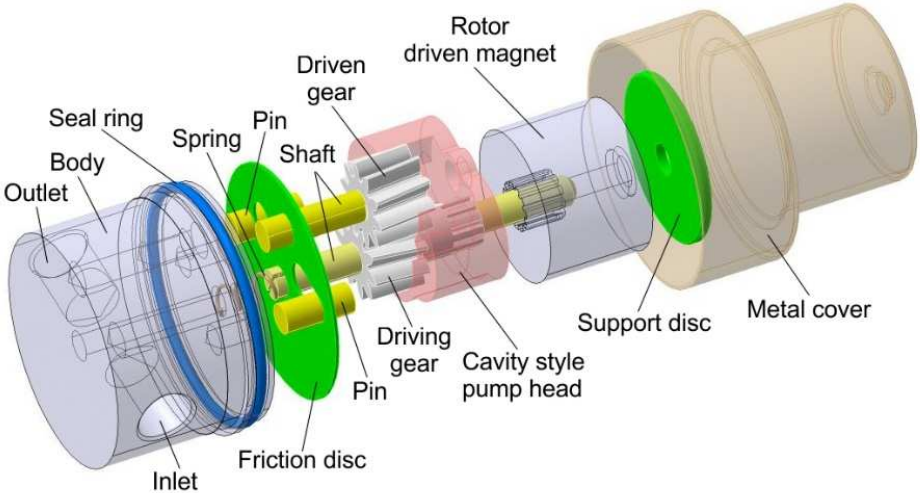


**Fig. 10.** The 3D model of the proposed micropump

Regarding the materials from which the components of the micropump are proposed to be manufactured, the 316 steel type can be retained for the metal components, as in the current micropump. It is an austenitic chromium-nickel stainless steel containing molybdenum. This addition increases general corrosion resistance, improves resistance to pitting from chloride ion solutions, and provides increased strength at high temperatures.

This steel of 316 type has also an improved corrosion resistance against hydrochloric, sulfuric, acetic, formic and tartaric acids, acid sulfates and alkaline chlorides [28].

The gears, the new suction element and the two discs are proposed to be machined from the POM material. For this micropump we propose the use of Ultraform [29], a co-polymeric POM created by the BASF company with excellent sliding friction properties.

Ultraform is the company’s brand name for polyoxymethylene product range, it comprises versatile engineering plastics having a wide array of properties designed to be used in manufacturing complex components capable of withstanding high stresses.

This material is suitable for most applications in the automotive manufacturing, medical technology, for machine construction and various household applications. The main advantages of the POM material are: high rigidity and stiffness, good resilience properties, excellent resistance to chemicals, dimensional stability, low water absorption, low tendency to creep, excellent sliding friction properties.

The Ultraform product line contains various grades for processing by means of extrusion and injection molding. Some grades have a high melt strength and high molecular weight, being optimized for the extrusion of thin-walled and thick-walled tubes and panels, hollow profiles and semi-finished products having wall thicknesses of up to 50 mm and more.

This material can be manufactured into gears [29, 30], friction discs, bearings and other machine elements without classic cutting procedures. It can be processed rapidly, without deposits and are also easy to demold, with a very good surface quality.

## 5. CONCLUSIONS

The research conducted in the project is part of the research on current topics of increasing the performance and lifetime of products and optimizing their maintenance.

Also, there were defined the shape, size and position of the contact surfaces that are influenced by the wear process during operation. For this purpose, some constructive changes have been established and proposed for analysis and we have redesigned some components of the micropump, while others are kept like in the original/actual variant.

It is also important, from a functional point of view, the analysis of the modification influence of some parameters imposed on the hydraulic environment (viscosity, flow, absence or presence of particles etc.).

In the next stages of the prototype, as well as in the execution, assembling and testing, it will be important to analyze various constructive solutions that will ensure functional, operational and maintenance requirements.

These may result in functional modifications of the micropump: the distance between the gears shafts, the eccentricity between the outside circles of the teeth and the shaft's holes in the pump body, deviations from the gears shafts perpendicularity, from the planar surfaces of the gears and of the micropump's body.

Future research in the field of this type of micropumps will include approaches to redesigning the body and cavity in order to add and ensure the mixing and dosing functions of two compatible fluids to pump them into the hydraulic system.

It is also expected a complex study for the parameterized creation of a micropump family in order to provide a wider range of flows and pressures, as well the possibility to flow a variety of fluids with different viscosities.

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## Proiectarea adaptivă a modelului 3D al unei micropompe cu antrenare magnetică în vederea extinderii perioadei de utilizare şi pentru o mentenanţă redusă

**Rezumat:** Lucrarea prezintă unele rezultate preliminare ale cercetărilor privind reproiectarea inovatoare și adaptivă a unei micropompe cu antrenare magnetică cu dantură înclinată. Autorii au evidenţiat câteva aspecte ale angrenajului micropompei şi, în special, a componentelor uzate datorită funcţionării acesteia în instalația hidraulică. Procedura de proiectare a modelului 3D al micropompei şi toate ideile implicate şi utilizate în cercetare pentru a îmbunătăți performanțele actuale ale acesteia s-au bazat pe teorii și principii relevante din domeniul hidraulicii. S-au considerat aplicațiile industriale reale ale micropompei cu antrenare magnetică, cum ar fi: tipărirea pe suport de tip materiale textile, plăci ceramice, controlul emisiilor motoarelor Diesel prin injecția cu aditivi de combustibil etc. În consecință, micropompa trebuie să aibă capacitatea de a antrena inhibitori de coroziune, lubrifianţi, combustibili, coloranţi, detergenţi, fluide de degivrare, aditivi, etc. Astfel, starea de uzură observată a unor componente ale micropompei a condus cercetările şi către o versiune îmbunătăţită si inovatoare cu aceeaşi eficienţă volumetrică sau mai mare şi debit modificat. Procedura de cercetare subliniează avantajele competitive ale designului inovator 3D cu parametri tehnologici simulați numeric, dovedind corectitudinea și îmbunătățirile noii variante de proiectare. Soluția CAD inovatoare prezentată ne permite să dezvoltăm o serie de micropumpe compacte, robuste, fiabile și cu costuri reduse de fabricație.

**Ionuţ Gabriel GHIONEA**, Phd. Eng., Lecturer, University Politehnica of Bucharest, Faculty of Engineering and Management of Technological Systems, Department of Machine Manufacturing Technology, E-mail: ionut.ghionea@imst.pub.ro, Phone: +40722926249, Address: Bucharest, Romania, Spl. Independenţei nr. 313, district 6.

**Constantin Gheorghe OPRAN,** Phd. Eng., Professor, University Politehnica of Bucharest, Faculty of Engineering and Management of Technological Systems, Department of Machine Manufacturing Technology, Composites Products Laboratory, E-mail: constantin.opran@ltpc.pub.ro, Address: Bucharest, Romania, Spl. Independenţei nr. 313, district 6.

**Adrian Lucian GHIONEA**, Phd. Eng., Dr.H.C., Professor, University Politehnica of Bucharest, Faculty of Engineering and Management of Technological Systems, Department of Machines and Manufacturing Systems, E-mail: adrianghionea@yahoo.com, Phone: +40744110472, Address: Bucharest, Romania, Spl. Independenţei nr. 313, district 6.

**Cristian Ioan TARBĂ**, Phd. Eng., Lecturer, University Politehnica of Bucharest, Faculty of Engineering and Management of Technological Systems, Department of Machine Manufacturing Technology, E-mail: cristian.tarba@imst.pub.ro, Address: Bucharest, Romania, Spl. Independenţei nr. 313, district 6.

**Saša ĆUKOVIĆ**, PhD, Scientific Associate, University of Kragujevac, Faculty of Engineering, E-mail: cukovic@kg.ac.rs, Phone: +38134335990, ext. 643, Address: Kragujevac, Serbia, Sestre Janjic 6, 34000.