Designing Gear Pump Bodies Using FEM

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1 Introduction

Gear pump make the largest and the most frequently applied group of fluid power energy generators [1]. It results from a number of the pumps' qualities, such as:

- high delivery rate and a wide pressure range,
- high volumetric and total efficiency,
- high durability and reliability,
- small size and low weight of the pumps in relation to the generated delivery and pressure (power).

The development work done so far in the field of gear pumps has focused predominantly on the design and research of the gears [2–7] as well as on the research into the flow processes and phenomena observed in the channels and clearances of the pump [8–15]. They also concern the collaboration of the gear pumps with the hydraulic system [19] and the reduction of noise produced by the working gear pumps [17–19]. The work on the designing of the gear pump bodies, however, has been done less often. Traditionally, the bodies have normally been formed intuitively, so that they would tightly enclose a gear set, a system of channels and clearances. Various design solutions of that type have been shown in [1]. The approach to the problem changed with the start of mass production of gear pumps amounting to several thousand pieces made yearly by individual manufacturers. It became necessary to undertake fundamental research on the strength and durability of the pump bodies to ensure high technical specifications, long life and reliable operation, at the small size and weight of the bodies and In [20], a method of calculating the stress and strain in the external gear pump body by means of classical mechanics and strength of materials was presented. The considered pump body was a commonly used shape of a right prism with a base similar to a rectangle. The load range of the body was verified. In [21, 22], an original method of calculating strength of the gear pump body using FEM was presented. It was applied in the design of high-pressure gear pumps [23]. FEM is now widely used to analyse the strength of hydraulic machines. An example is work [24], where an analysis of deformation in a gear pump body caused by pressure and temperature was carried out in order to determine the axial clearance in the pump. Another example are works [25, 26], in which FEM was used to define the stress and strain of POM

E. Rusiński and D. Pietrusiak (Eds.): CAE 2018, LNME, pp. 767–783, 2019.
https://doi.org/10.1007/978-3-030-04975-1_89

cycloidal gears working in a gerotor pump body. This method was also used to design the bodies of large capacity external gear pumps [27].

Based on the literature overview, the following observations can be formulated:

- there are no clear design criteria for evaluating the design solution of the pump body
- the publications dealt with some previously selected shapes of external gear pump bodies, and generally the strength calculations concerned the determination of the load capacity range of those bodies, but did not deal with the internal gear pumps and gerotor pumps,
- there are no clear conclusions concerning the impact of the design solution and the operational parameters on the strength of the pump body.
- it is noted that the commonly used method of strength analysis of bodies is the FEM method

In this situation, the following objectives were adopted:

- to determine the rules of designing the bodies of the three basic kinds of gear pumps, namely the external gear pump, the internal gear pump, and the gerotor pump, and in this:
  - formulating criteria for designing gear pump bodies
  - analysis of various body shapes and determination of basic shape
  - determination of the impact of the pump construction solution, and in particular the use of a particular type of toothing and the arrangement of internal channels on the shape and dimensions of the body
- carrying out strength analysis of bodies with basic shapes using the FEM method, in order to determine the impact of the construction solution and operational loads on the state of their stresses and deformations
- modification of the basic shape to reduce stress and deformation of these bodies.

2 Analysis of the Shape of the Gear Pump Bodies, Determining the Basic Shape

The body of the gear pump must fulfill the basic design criteria, i.e. constructional and technological. From a structural point of view, the body should be shaped in such a way that it encloses a set of gears together with the shaft and bearings and a set of channels and flow gaps. The cooperation of these teams with the body will ensure the process of converting mechanical energy into a hydraulic one. The course and efficiency of this process depends on the shape and dimensions of the body. Therefore, the body should meet the strength conditions, i.e. the stresses in the body and its deformations should be less than the allowable stresses and deformations, i.e. $\sigma < \sigma_{allow}$, $d < d_{allo}$. From the technological point of view, the body should have a simple shape, and its implementation should be possible by available and cheap production methods.

Figure 1 presents typical shapes of the three basic types of gear pumps:

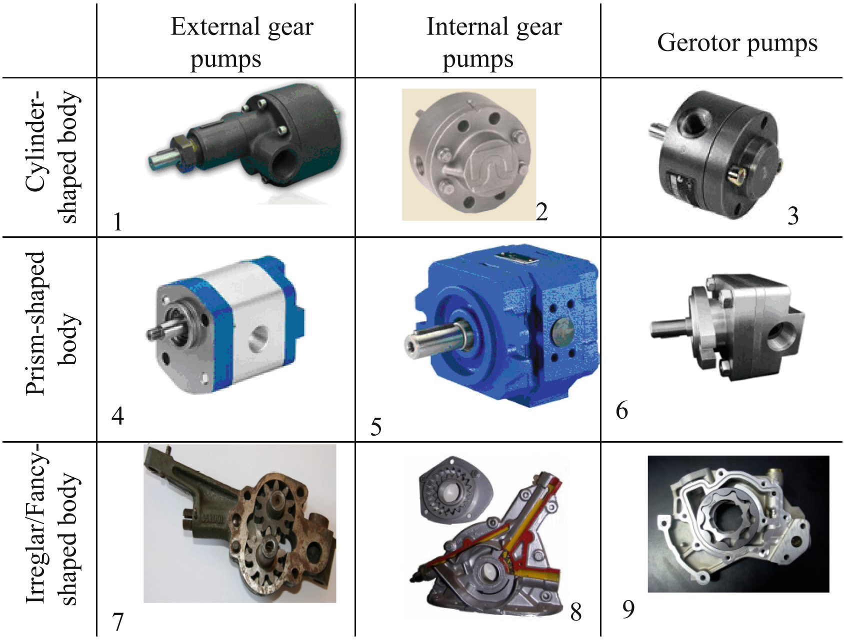

Fig. 1. Classification f gear pumps by the type of gearing and body shape [38]

- external gear pumps – no. 1, 4, 7
- internal gear pumps – no. 2, 5, 8
- gerotor pumps – no. 3, 6, 9

An analysis of the examples presented of the pumps allows for the distinguishing of three typical body shapes:

- a cylinder-shaped body – no. 1, 2, 3
- a rectangular base prism-shaped body with optional curves - no. 4 or a square base prism-shaped body – no. 5, 6
- irregular- or fancy-shaped body – no. 7, 8, 9

Cylindrical or prism body shapes meet the specified design and technological criteria. The bodies with fancy shapes meet the given criteria to a lesser extent, especially due to the complex shape and higher technological requirements. The fancy shape normally results from the necessity to integrate the units with other fluid power systems. It concerns, for instance, the lubrication or cooling systems for cars and other vehicles, or power steering of vehicles and heavy-duty machines.

For further analysis, the so called basic shape of the gear pump body was assumed. It was decided that in the case of all kinds of the pumps it was a right prism with a rectangular like base (no. 4) or a square like base (no. 5, 6). The basic body is divided into three parts: in the front part there is a mounting flange, in the central part there is a gear system, whereas the hydraulic mounting ports are located either in the central or in the back part, perpendicular to the drive shaft axis. The three parts of the body are jointed by screws.

Taking a prism as the basic shape for the pump body, the Fluid Power Research Group from Wroclaw University of Technology designed and manufactured models of three gear pumps which are shown in Fig. 2, and they are:

- external gear pump [18] – Fig. 2b, c
- internal gear pump [21] – Fig. 2e, f,
- gerotor pump [22] – Fig. 2h, i.

Following the assumptions, the pump bodies were designed respectively to the accepted gear systems and the internal channel systems collaborating with them, where all the units were designed for the same delivery rate, which enables the comparing of the units.

Figure 2 shows that going from the external gear systems to the internal gear systems, and then to the gerotor systems allows for the reduction of the cross-section of

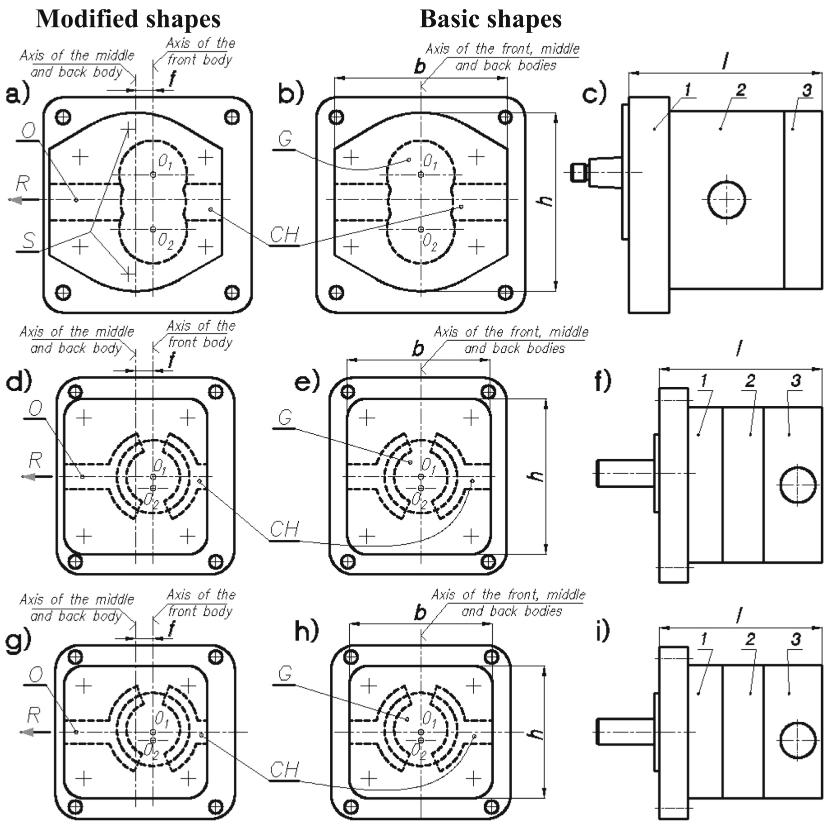

Fig. 2. Profiles of the basic and modified shapes of gear pump body units: a, b, c: external gear pump d, e, f: internal gear pump g, h, i: gerotor pump

the body (the prism base). It results from the comparison of *b* and *h* in Fig. 2b, e, h of the analysed pumps.

Figure 2 also shows that the channel systems (CH) and the bearing systems of the internal gear pumps and gerotor pumps placed in the side bodies enables the reduction of the overall length of the body (the height of the prism), in relation to the external gear pump. This stems from the comparison of dimensions *l* marked on individual silhouettes of the pumps – Fig. 2c, f, i.

The body of basic shapes should be subjected to strength analysis in order to determine the state of stress and deformation.

3 FEM Strength Analysis of the Gear Pump Bodies

3.1 Geometrical Models, Loads and Restraints

The basic shape gear pump bodies depicted in Fig. 2 were subject to strength analysis. Figure 3 shows the geometrical models of the pumps with the loads and restraints distribution.

The very models of the bodies were marked with a bold solid line, while in their background, additionally, by means of a dashed line, pump elements (gears, shafts, bearings and screws) collaborating with the bodies and generating the loads were presented. In the figures of each of the models, both the mechanical and hydraulic loads systems, as well as the restraints system were marked.

As shown in Fig. 3a, the geometrical model of the external gear pump body consists of three parts, namely of the front body (1), the central body (2) and the back body (3) jointed by screws (4). Inside the body, gears with shafts (5) and bearings (6) are located. It was assumed that the models of the bodies, similar to the real pumps, are made of a 150 MPa strong aluminium alloy, and it was decided that they would be working in the elastic range. The pump loads result from the operational principle of the pump.

The mechanical loads (marked in the Fig. 3a with arrows) are generated by torque T working on the pump's shaft and by cramp Q of the screws acting on the body. The torque generates intertooth forces F_{i1} and F_{i2} which load the shafts and the bearings. At the same time, the gears and the body are worked on by the hydraulic load in a form of pressure p. It generates forces F_{p1}, F_{p2} loading the gears. The summary of F_{i1}, F_{p1} and F_{i2}, F_{p2} results in generating forces F_{s1}, F_{s2} which load the gears and the bearings of the pump. Forces F_{s1} and $F_{s2,}$ however, generate bearing thrust p_{b1}, p_{b2} working on the inner surface of the central body (2).

Torque T applied on the pump shaft is calculated according to [1] from the formula:

$$T = \frac{q}{2\pi}\Delta p$$

where: q – pump delivery per 1 revolution of the shaft, $\Delta p \approx p$ – outlet pressure from the pump

Intertooth forces $F_{i1,}$ F_{i2} are calculated according to [2] from the formula:

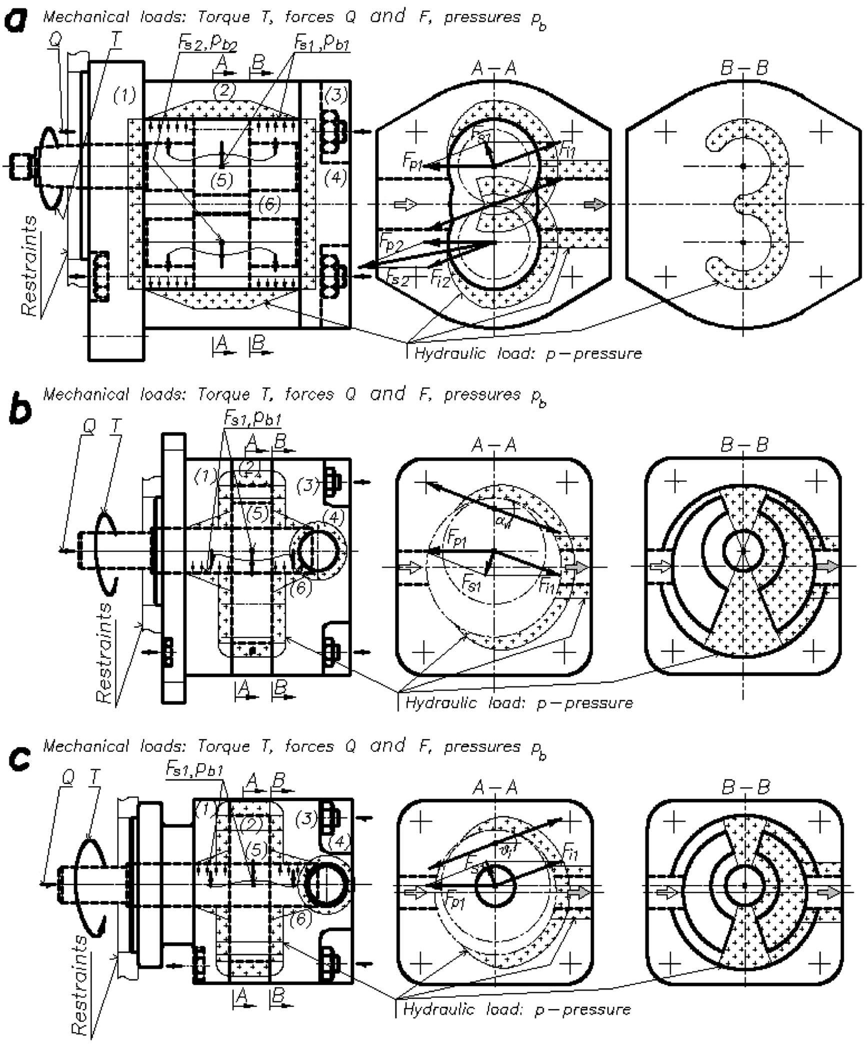

Fig. 3. Geometrical models of the gear pump bodies with the loads and restraints distribution a - external gear pump b - internal gear pump c - gerotor pump

$$\mathrm{F}_{\mathrm{i1}} = F_{i2} = \frac{2T}{d_{w1} * \cos \alpha_t}$$

where: d_{w1} – pitch diameter of the gears, α_t – pressure angle

Pressure generated forces F_{p1}, F_{p2} are calculated from the formula:

$$F_{p1} = F_{p2} = p * d_a * b$$

where: d_a – outside diameter of the gear, b – gear width.

As it has been mentioned above, the hydraulic loads (marked in the Fig. 3a with crosses) are generated by pressure *p* produced by the pump. The pressure acts on the central, back and front bodies. In the central body, it works on the ports in which gears revolve, as well as on the chambers and the outlet from the pump. Pressure *p* works also on the surface between the body and the bearings. The pressure distribution has been assumed following [1]. Pressure *p* works also on the front areas of the front body (1) and of the back body (3), on the areas limited by sealings.

Following the structure of the geometrical model of the external gear pump body, the geometrical models of the internal gear pump and gerotor pump bodies were built, which is illustrated by Fig. 3b, c.

In order to determine the mechanical and hydraulic loads, the same rules and formulas were used. An exception is the formula used for the determining of intertooth force F_{i1} in the gerotor, which was determined following [2] from the dependence:

$$F_{i1} = \frac{2T}{d_{w1} * \sin\vartheta'_1}$$

where: ϑ'_1 – intertooth force angle

The assumed restraints distribution was the same for all the pump models. The restraints are made by vertical panels to which the front bodies of the pumps were fixed by means of screws.

Thrusts p_{b1}, p_{b2} of the bearings on the central body are determined as:

$$p_{b1} = \frac{F_{s1}}{2 * d_b * l};\ p_{b2} = \frac{F_{s2}}{2 * d_b * l}$$

where: d_b – bearing diameter, l – bearing length

Mechanical loads generated by force Q the screw clamp are determined as:

$$Q = \frac{M}{0{,}5 * d_s * tg(\gamma + \rho')}$$

where: M – screw torque, d_s – thread diameter, γ – helix angle, ρ' – apparent friction angle

3.2 Numerical Models, Research Programme

Using [28–31], numerical models of the bodies for the three analysed pump types were developed. The views of the bodies with the assumed finite element mesh are presented in Fig. 4a, b, c. For the making of the mesh, the 1st-order HEXA elements of the same size were used (Fig. 4d). A HEXA element is a cubic element. The element features eight nodes and is a 1st-order element. In each node, the element has three degrees of freedom which are shifts ux, uy, uz relative to the coordinate axes.

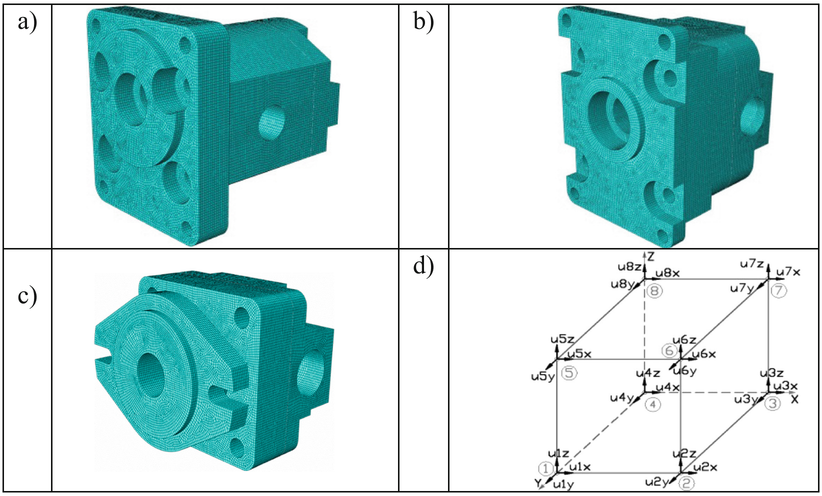

Fig. 4. Discrete models of the gear pump bodies a) external gear pump b) internal gear pump c) gerotor pump d) HEXA element

Figures 2 and 4 show that the capacity of the bodies varied. It was the highest in the case of the external gear pump, average for the internal gear pump, and the lowest for the gerotor pump. Respectively, the number of the finite elements in the mesh of the pumps at the same size of the elements decreased, and was;

- 559445 for the external gear pump,
- 392348 for the internal gear pump,
- 315314 for the gerotor pump.

The stress and strain analysis was carried out by means of the ABAQUS program version 6.11-2.[1]

The research included the strength analysis of three selected basic models of gear pump bodies (the external gear pump, the internal gear pump, the gerotor pump). Based on that, the nature and values of the stress and strain in particular bodies needed to be determined and compared.

While examining the models of the pumps, they were loaded with:

- mechanical loads, namely torque T = 8, 16, 24, 32 Nm and screw clamping force Q = 35 Nm
- hydraulic loads, namely working pressure p = 5, 10, 15, 20 MPa.

[1] The license of the program number 05UWROCLAW has been made available to the FPRG [4] by Wroclawskie Centrum Sieciowo-Superkomputerowe of Wrocław University of Technology.

3.3 Findings and Discussion

Figure 5 presents characteristic results of the stress and strain analysis of the basic shape gear pump bodies loaded with the total mechanical load by torque $T = 16$ Nm and screw clamping forces $Q = 35$ Nm, as well as with the hydraulic load by pressure $p = 10$ MPa. The bodies of all the pumps have been presented in the same way, namely in a general view (Fig. 5a, c, e) and section (Fig. 5b, d, f). In the view of the bodies, the areas of maximum stress and strain have been marked.

The analysis of the state of stress shows that in the bodies of all pump types, the highest values of reduced stress σ occur around the outlets. The areas have been marked in Fig. 5 with letter O both in the pump views and in their sections. The

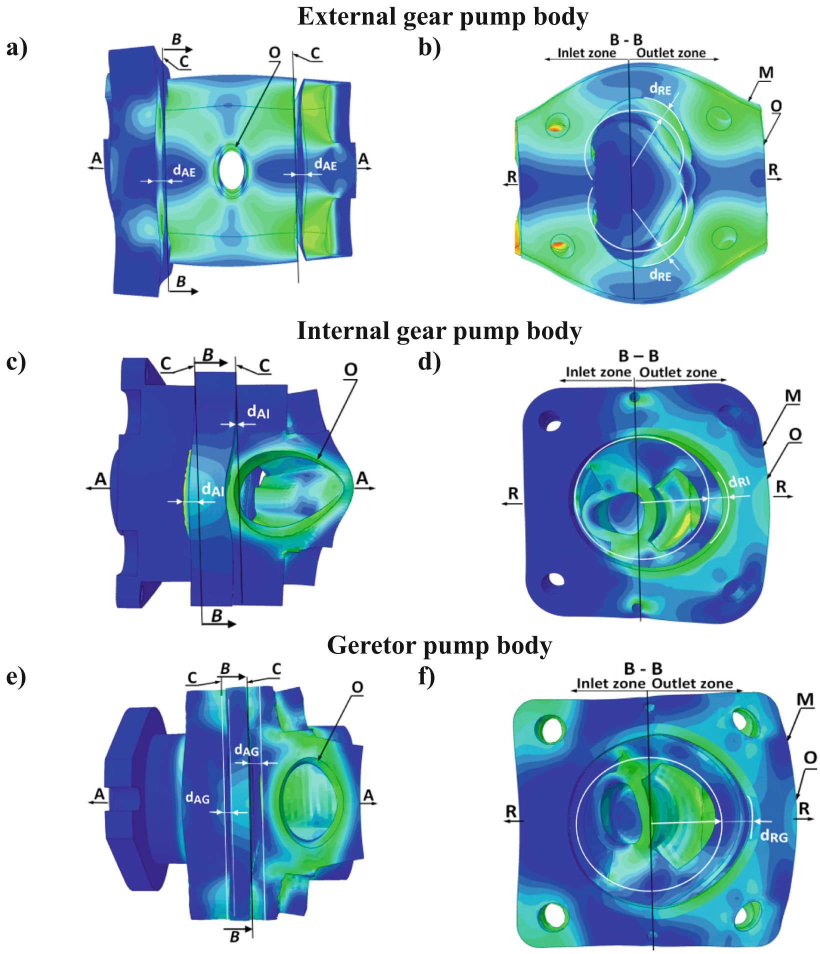

Fig. 5. Distribution of stress and deformations in the basic shape gear pump bodies, a; b) external gear pump σ_E, d_{RE}, d_{AE}, c; d) internal gear pump σ_I, d_{RI}, d_{AI}, e; f) gerotor pump σ_G, d_{RG}, d_{AG}.

maximum stress σ observed around outlets O is taken as a criterion for the assessment of the pump body effort. Trying to determine the source of the stress, it is needed to return to Fig. 3 which illustrates that high pressure p works on the node including the outlet chamber, the channel, the outlet of the pump, resulting in the load concentration in that node and in high stress σ.

Based on the FEM analysis, the relation between stress σ and pressure p loading the pump, namely σ = f(p), for the three considered pump types was determined. The relations are presented in Fig. 6a, b, c. The figure shows that stress σ grows along with an increase in working pressure p according to the directly proportional dependence, for all the pump types.

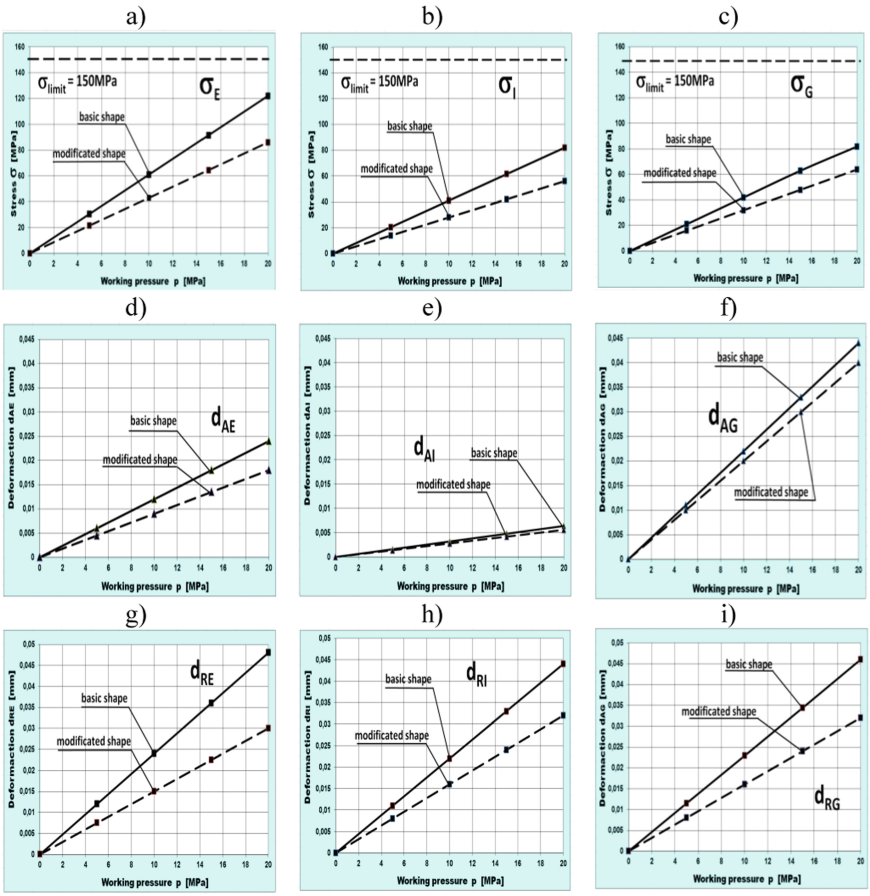

Fig. 6. Stress and strain characteristics of the basic shapes (continuous line) and the modified shapes (dashed line) gear pump bodies depending on the working pressure. a–c) stress characteristics $\mathbf{s_{A,I,G}}$ = f(p), d–f) axial deformation characteristics d_{AE}, d_{AI}, d_{AG} = f(p), g-i) axial deformation characteristics d_{RE}, d_{RI}, d_{RG} = f(p).

The analysis of the impact of the pumps' design on the stress, however, allows for the observation that the stress values in the internal gear pump body and in the gerotor pump body are lower than in the external gear pump body where hydraulic and mechanical loads concentrate around the outlet.

When assessing the state of stress, it is noted that in all basic bodies shapes, with their loading with working pressure p = 0–20 MPa, the stresses did not exceed the allowable values. This stress condition can be written as:

$$\sigma_E,\ \sigma_I, \sigma_G > \sigma_{allow} = 150\,\mathrm{MPa}$$

Analysing the deformations presented in Fig. 5, it can be noticed that the bodies of all the pumps become deformed both axially (A) and radially (R). Consequently, axial deformations d_A and radial deformations d_R of the body occur. Axial deformations d_A and radial deformations d_R are assumed as the criteria for the assessment of the state of the pump body deformations.

Analysis of Fig. 5a, c, e shows that in the case of all the pump types, the highest values of axial deformations d_A are visible mainly in the contact areas C between the central and front bodies as well as between the central and the back bodies. Similarly, analysis of Fig. 5b, d, e shows that the highest values of radial deformations d_R are visible in all cases in the central body M around the outlets of all the examined pumps.

Hence, reconsideration of Fig. 3 makes it possible to observe that the cause of the axial deformations is the activity of high pressure on the front face of the front and back body of a particular pump.

Based on Fig. 3, it is also noted that a cause of radial deformations d_R is the impact of high pressure onto the inner surface of the central body of a particular pump.

On the basis of the FEM analysis, the dependence of axial deformations d_A and radial deformations d_R on pressure p loading the pump, namely d_A = f (p), and d_R = f (p), was determined. Those relations are shown in Fig. 6d–i. The figures show that in all types of pumps deformations d_A and d_R grow with an increase in pressure p according to the directly proportional dependence. Comparing the deformations shown in Fig. 6d–i, it is noted that radial deformations d_R are greater than axial d_A, and that is observed for all types of pumps.

Based on the literature [1, 2, 5, 15, 20, 22, 27, 32, 33], the allowable axial d_A and radial d_R deformations of the pump bodies, which are d_A = 0,020 mm, d_R = 0,040 mm, can be determined. When assessing deformations, it is noted that in all basic bodies shapes, with their loading with working pressure p = 0–20 MPa, axial and radial deformations exceed the allowable values. This stress condition can be written as:

$$d_{AE},\ d_{AI},\ d_{AG} > d_{Aallow} = 0,020\,\mathrm{mm}$$
$$d_{RE},\ d_{RI},\ d_{RG} > d_{Rallow} = 0,040\,\mathrm{mm}$$

This ultimately leads to the conclusion that the basic shape of pump bodies should be modified in order to reduce their axial and radial deformations.

4 Determining the Modified Body Shape of Ear Pump and Their Strength Analysis

4.1 The Principles of Modifying the Shape of the Body

Modification of the basic shape of a pump body consists in reshaping of the whole solid of the body or of its main nodes while maintaining the compactness of the solid and the smallest possible dimensions. As a result of the modification, the internal stress in the elements of the body as well as the elements' deformations should be reduced.

Some global modification can be performed, which includes changes of the regular shape of the body into an irregular or even fancy shape of the body. But also, some local modification can be made, which consists in changing the design, its shape or dimensions of individual nodes of the body.

Global and local modifications have been made to the basic gear pump bodies. The bodies with modified shapes are shown in Fig. 2a, d, g.

The global modification made according to [34–36] consisted in a shift of the central and back bodies in relation to the front body towards the outlet zone of the pump by distance f (see Fig. 2a, d, g). Thus, the central and back bodies of the pump in the area of the typical heavily loaded chambers, channels and outlets O, became thicker along radius R. That should result in a reduction of stress σ in the outlet O zone, and, more importantly, of a reduction in the radial deformations of the central and back bodies. At the same time, it should be pointed out that the axis of the drive shaft of the pump, as well as gears' axes O_1, O_2 remain situated symmetrically in relation to the front body and to the flange and mounting holes. This ensures smooth installation of the pump and its connection to the motor. It should also be noted that the pump body as a whole still has the form of a prism with a cross-section similar to a square, and unchanged overall dimensions.

The local modification was carried out according to the type of pump. In the case of the external gear pump, the number of screws jointing the bodies was increased. 6 screws were used instead of 4 screws typically used before. Two additional screws S were introduced at the top and bottom of the body (see Fig. 2).

The local modification in the case of the internal gear pump consisted in the decreasing of the diameter of outlet O of the back body (see Fig. 2d). In that way, the area affected by high pressure was reduced, and, consequently, the forces deforming the body in axial direction A were reduced. A similar modification of outlet O was made in the gerotor pump (see Fig. 2g).

4.2 Findings and Discussion

The modified shape bodies of the three considered pump types were subject to the FEM strength analysis. The loads and restraints distribution, as well as the numerical models, were developed according to the rules similar to those for the basic shape bodies. Also, the research programme adopted for the modified shape bodies was the same as the one for the basic shape bodies.

Figure 7 shows the results of the stress and strain analysis of the modified shape pump bodies loaded with torque T = 16 Nm, Q = 35 Nm and pressure P = 10 MPa.

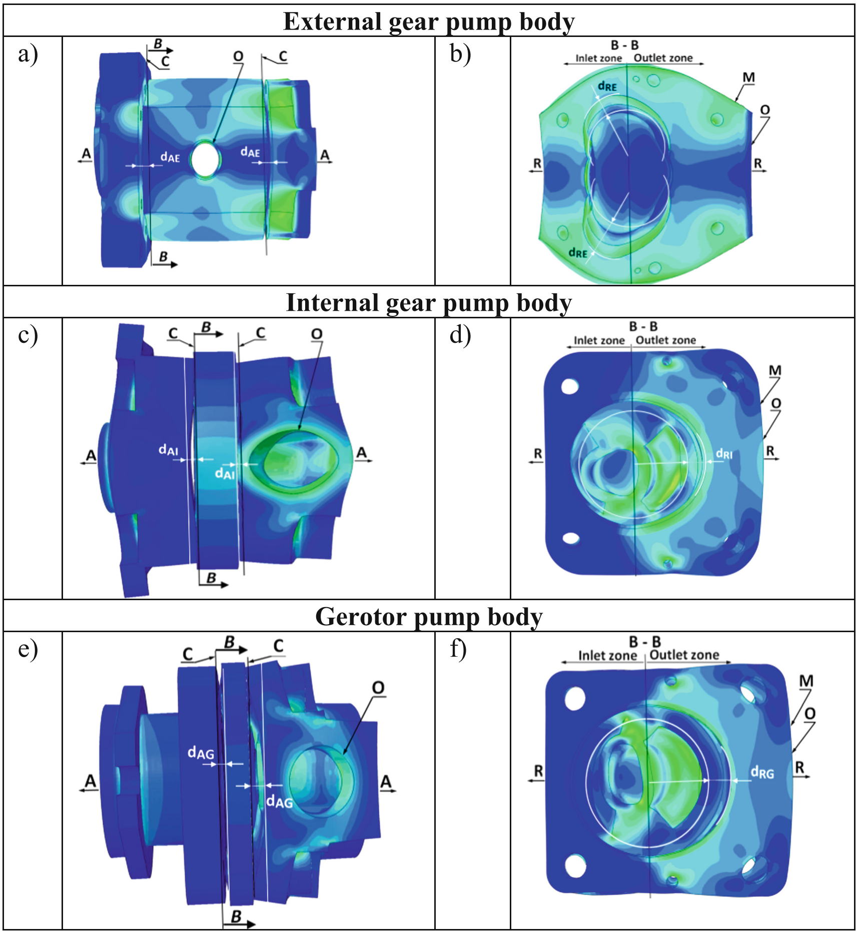

Fig. 7. Stress and strain distribution in the modified shapes pump bodies a; b) external gear pump c; d) internal gear pump e; f) gerotor pump

The analysis of the stress distribution shows that the greatest reduced stress σ, as in the case of the basic shape, occurs on the outlet side of the bodies, in the area of outlet O.

Treating stress σ as the criterial stress, its interrelation to pressure *p* loading the pump, namely σ = f(p) for three modified kinds of bodies was studied. The interrelation is shown in Fig. 6a, b, c in a form of the dashed line. Figure 6a, b, c show that the stress increases along with an increase in pressure *p* according to the direct proportional dependence for all the modified bodies.

When assessing the state of stress, it is stated that for all bodies with modified shapes, when their working pressure is p = 0–20 MPa, the stresses do not exceed the permissible values. At the same time, the stresses were lower than the stresses in the basic body.

Analyzing the state of deformation, shown in Fig. 7, it is stated that modified bodies, similarly to basic bodies, are deformed in the axial (longitudinal) A and radial (transverse) R directions. In Fig. 7a, c, e show that still the greatest axial deformation d_A occur on contact surfaces C between middle and front body, as well as between middle and back body. In turn, Fig. 7b, d, f shows that the largest radial deformations d_R occur in the middle body near outlets O. Figure 6 shows that axial d_A and radial d_R deformations also increase in direct proportion to the pressure p that loads the pump bodies.

When assessing deformations, it is noted that in all bodies with modified shapes, axial and radial deformations are smaller than in bodies with basic shapes, in the range of the same loads p = 0–20 MPa. It was particularly advantageous to reduce radial deformations that fell below the permissible values, which can be written:

$$d_{RE},\ d_{RI},\ d_{RG} < d_{Rallow} = 0,040\ \text{mm}$$

Axial deformations d_{AE}, d_{AI}, d_{AG} were also reduced, but to a lesser extent. This signals that further modification of the bodies should be made.

5 Conclusion

Realizing the objectives of the article, the author's method of designing gear pump bodies has been developed, which includes:

- formulation of basic design criteria
- determining the basic shape of the body and its strength analysis
- modification of the basic shape and its strength analysis
- adopting the shape of the final body.

It has been established that the basic shape can be a prism with a rectangular or square base, divided into three parts, which are jointed by screws. The body of this basic shape can be used for pumps with external, internal and gerotor meshing. The subsequent transition from the external to internal and gerotor meshing allows reducing the dimensions of the basic body. Strength analysis of the bodies were carried out using the FEM method using the ABAQUS program. It was found that the stresses in the basic bodies were smaller than the allowable ones, but the deformations of the bodies were greater than the allowable ones. A global and local modification of basic bodies was carried out and bodies with modified shapes were obtained. Repeated strength analysis showed that stresses and deformations in the modified bodies visibly decreased in relation to the stresses and deformations occurring in the basic bodies. This means that the use of shape modification of the bodies is useful in the design process. It is noted that the introduction of asymmetrical shapes may be useful in design practice. At the same time, it is noted that the shape modification process is "open" and can be continued to achieve further reduction of stresses and deformations.

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