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AN INVESTIGATION OF THE IMPACT OF PUMP DEFORMATIONS ON CIRCUMFERENTIAL GAP HEIGHT AS A FACTOR INFLUENCING VOLUMETRIC EFFICIENCY OF EXTERNAL GEAR PUMPS

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Abstract. Gear pumps are widely used in transport machines not only in hydraulic drive systems, but also in lubrication and fuel systems. A positive displacement pump converts mechanical energy into pressure energy stored in the liquid, which it transports to the hydraulic cylinder or motor. The efficiency of this process depends on the efficiency of the system components, including the efficiency of the pump, which depends on the amount of internal leakage through the gaps between the high and low pressure sides. The larger the gaps, the lower the volumetric efficiency. This article presents an investigations of impact of pump deformations on circumferential gap height. The article presents a three-dimensional model of an external gear pump using Finite Element Method (FEM) during operating conditions. The model reflects pumping operation at discharge pressures up to 32 MPa and it was validated against strain measurements of pump casing. The simulation results indicate that the pump casing becomes deformed due to pressure, causing a significant increase in the height of the circumferential gap. The increase of the discharge pressure from 8 to 32 MPa causes more than twofold local increase in the height of the circumferential gap. The obtained results indicate that for the correct modelling of the flow generated by gear pumps, it is necessary to consider the change in the size of the gaps resulting from the deformation of the pump.

Keywords: circumferential gap, finite element method, gear pump, leakage, pump deformation, simulation, volumetric efficiency.

Notations

Abbreviations:

CFD – computational fluid dynamics;
FEM – finite element method;
HYGESim – hydraulic gear machines simulator;
MPC – multi-point constrain;
PTFE – polytetrafluoroethylene.

Variables and functions:

A – cross-section of the orifice;
b – width of the tooth tip;
C – contraction coefficient depending on the shape of the gap;
d_m – mean diameter of the surface area of the back nut;
d_s – pitch circle diameter;
E – Young's modulus;
F – tensile force;
h – gap height;
i – number of tooth;
M – assembly torque;
M_p – torque on the pump shaft;
n – rotational speed of the pump;
p_i – load pressure at i-th tooth past the intake port;
p_t – discharge pressure;
q – specific delivery;
Q – actual flow generated by the pump;
Q_c – circumferential gap flow;
Q_{ideal} – ideal flow generated by the pump;
Q_{leak} – flow due to leaks;
$R_{0.2\,\min}$ – minimum yield stress;
$R_{m\,\min}$ – minimum tensile strength;
r – average distance of nodes forming the teeth tips and nodes forming the inner casing surface from the axis of rotation of the gear in initial, unladed position;
r_s – radius of the pitch circle;

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v – velocity at the gear tips;
γ – spiral angle;
Δp – differential pressure;
η_v – volumetric efficiency;
μ – friction coefficient;
ν – Poisson's ratio;
ρ – fluid density;
ρ' – apparent friction coefficient;
ϕ – angle of rotation of pump shaft.

Introduction

Gear pumps are used in transport machinery for many applications. They can be found in fuel systems and lubrication systems of internal combustion engines. They are also used in circulating lubrication systems where the lubricant is oil. However, they are most often used in a hydraulic drive.

Gear pump belongs to the group of gear displacement machines, among which, apart from external gear machines, there are also internal gear machines and satellite machines. They convert mechanical energy into pressure energy. Gear pumps have a simple design, high efficiency, high reliability and low price. In a gear pump, the liquid is typically transported with a pair of gears. As they rotate, the liquid is transported from the intake port to the discharge port. A low-pressure zone is formed at the pump inlet and causes the working liquid to be drawn. The liquid is subsequently transported in the intertooth chambers towards the outlet port. The liquid is pumped as a result of being displaced from the intertooth chambers, as the teeth in successive pairs mesh with each other.

Gear pumps can be classified following several criteria. The pump may generate one or more than one streams (multiple pumps). The type of gears, on the other hand, allows such pumps to be classified as either internal or external gear pumps. This research focuses on single external gear pumps.

Despite their simple design and operating principle, gear pumps have some problems, which affect their key operating parameters. Significant pressure differences between the discharge and the intake areas cause some of the pump elements to become displaced. As a consequence, gaps form inside the pump and cause the discharge area to lose its tightness.

1. Gaps in external gear pumps as a factor influencing the volumetric efficiency

Loads due to pressure cause the gears, bearings and the bearing blocks to be displaced towards the intake area, and in effect a circumferential gap is formed between the casing and the gears. Another area prone to leaks is located in the faces of the gears and of the bearing blocks, where an axial gap is formed. The size of the gap is of significance for the delivery, efficiency or maximum discharge pressure values of the gear pump. Pump efficiency, on the other hand, translates directly into the efficiency of the entire hydraulic system. This fact applies to all types of displacement pumps, and therefore the literature indicates a trend to improve both the efficiency of the pumps (Antoniak *et al.* 2019; Borghi *et al.* 2009; Kollek *et al.* 2014; Patrosz 2021; Śliwiński 2018; Zardin *et al.* 2019) and the efficiency of the entire system (Karpenko, Bogdevičius 2017; Lisowski *et al.* 2021; Karpenko *et al.* 2022).

Publications by Ghazanfarian, Ghanbari (2015), Rituraj *et al.* (2018), Schiffer *et al.* (2013) and Wahab (2010) describe pumps in the fuel delivery system, in which the working liquid has a significantly lower kinematic viscosity (below 10 mm^2/s) than for example mineral oil used in hydrostatic drive systems (typically approximately 68 mm^2/s at 40 °C). This is of importance, as the Hagen–Poiseuille equation indicates that low viscosity increases internal pump leaks (assuming laminar flow).

Schiffer *et al.* (2013) shows a gear pump used in direct fuel injection systems. The pump can be offered as a replacement for a traditional pump arrangement employed in such systems (a precharge pump together with a high-pressure pump). The article demonstrates the negative influence of the internal gap size on the volumetric efficiency of the pump. In order to determine the efficiency, the internal gap flow was measured. The research was based on a model of the circumferential gap with a constant height and with one moving wall, in which the flow is the sum of the flows due to pressure differences between the sides of the gap (Hagen–Poiseuille) and to the movement of the gear with respect to the casing walls (Couette). The analysis demonstrated that the total size of the gaps in the pump has a decisive influence on the volumetric efficiency and on the generated flow rate. The authors indicate that the implemented compensation of the axial gap is insufficient to reach a pressure difference from 4 to 5 MPa. An additional requirement is to use a compensation of the circumferential gap, which is observed between the teeth tips and the casing. The pump was provided with an elastic sealing of the circumferential gap in three design variants. The circumferential compensation referred to by the authors as V2.2 allowed discharge pressures of 4 MPa at a volumetric efficiency of 75%.

Wahab (2010) address a similar issue. The tested gear pump is used in a fuel injection system of a turbocharged engine not only to pump the fuel, but also as a fuel metering device. It replaces the valve system regulating the fuel charge. The publication presents a model for predicting the flow rate generated by the pump. The leaks through the gaps were identified depending on the flow type. For turbulent flow, it was the (turbulent) Bernoulli equation, and for the laminar flow, it was the orifice equation for rectangular gap with sharp edges. The article shows the expected flow rate at discharge pressures from 0 to 4.8 MPa. The maximum relative error between the expected flow rate and the experiment was 18%.

Rituraj *et al.* (2018) question the assumption of laminar flow through the circumferential gap between the casing and the teeth tips. In order to determine the instantaneous parameters of the pump, the authors use original HYGESim software, which consists of a number of modules: a lumped parameter model, a module for identifying forces acting on the gears and the torque on the pump shaft, and a geometric model identifying the values required for the other modules to work properly. HYGESim allows for micro-movements of the gears due to load, as well as for the wear of the pump casing due to the contact between the casing and the gears (Vacca, Guidetti 2011). The authors observe that in certain conditions (low viscosity liquids and a sufficiently large circumferential gap) the assumption about the laminar flow through the gap will cause the simulation results to significantly deviate. In exchange they propose that the flow through the gap is turbulent for low viscosity liquids, e.g., for aircraft fuel. The presented simulations, confirmed by experiments, demonstrate that an assumed laminar flow in such conditions results in increased leaks through the gap, which cause the volumetric efficiency of the pump to be underrated.

Ghazanfarian and Ghanbari (2015) shows a double external gear pump for fuel delivery systems. The pump comprises three gears, the central gear being the drive gear and the neighbouring two – the driven gears. The configuration of the connections allows two simultaneous streams of liquid to be delivered. The authors present a two-dimensional CFD model with a dynamic mesh, which allows the identification of instantaneous pump operating parameters, velocity values and distributions as well as pump pressures inside the pump. The model also allows the identification of both the areas in which cavitation is possible and the paths of the internal leaks. The authors also present the results of simulations performed for a decreased circumferential gap. With the gap reduced by 9 μm, the flow rate increased by 30%.

In hydrostatic drive mechanisms, gear pumps work with more viscous liquids, and the working pressures reach 32 MPa. It is a significant factor, which negatively influences the scale of leakage through internal gaps. An increase in the discharge pressure causes the leaks through the gaps to increase and the volumetric efficiency to decrease.

Borghi *et al.* (2009) discusses the influence of the rotational speed of the pump and of the discharge pressures on the volumetric efficiency. In the lumped parameter model, there is assumed a control volume reflecting the volume of the liquid inside the pump in which the pressure is constant and equal to the discharge pressure. The control volume is in communication with the other areas in the pump. The model assumes leaks through both the circumferential and the axial gaps. The presented simulation results indicate that the growth of the discharge pressure entails a growth in the flow through the axial gap, as this depends mainly on the pressure difference. The highest flow through the circumferential gap is observed at low rotational speeds and at low pressure difference, which fact is the result of low pressing force of the gears against the surface of the casing. The flow through the gaps decreases together with an increase in the rotational speed. The flows through individual gaps have a variable share in the volumetric losses, depending on the pressure and the rotational speeds. The efficiency grows together with the growth of pump rotational speed, while an increase in the discharge pressure causes the efficiency to decrease. The authors also observe the negative influence of the position of bearing blocks with respect to the gears on the scale of leaks through the axial gap. An inclination of the face of the bearing block with respect to the face of the gear results in increased leakage and reduced volumetric efficiency. The presented model proves adequate for discharge pressures lower than 65% of maximum pressure. The authors suggest that the differences observed for higher pressures can be potentially explained by the actual inclination of the bearing blocks, which may be greater than the expected inclination and which may cause increased leakage through the axial gap. To ensure correct operation of the axial compensation the proper methodology of designing axial compensation must be implemented Sliwinski (2019).

Guo *et al.* (2020) investigate the influence of wear on volumetric efficiency. The CFD model with dynamic mesh allows the identification of the instantaneous parameters of the pump. The model allows for the volumetric losses due to the flow through both the circumferential gap and the axial gap. The authors demonstrate that during pump operation, its casing is subjected to wear. On the intake side, there is a wear of material due to friction between the teeth tips of the gear and the pump casing. This contact is the result of the discontinuation of the oil film between the two elements. The second area in which moveable elements engage in friction is between the face of the gear and the side surfaces of the bearing blocks. As a result, the above surfaces become increasingly worn. The simulations were performed for discharge pressures within the range from 10 to 25 MPa and for various losses in the above areas. The wear in the casing varied from 20 to 120 μm, and the wear on the face of bearing blocks was from 50 to 500 μm. The authors observe a decrease in the volumetric efficiency of the pump due to wear, and also note that this decrease is influenced more by the wear in the pump casing than on the bearing blocks.

2. Influence of circumferential gap size on volumetric efficiency

The flow through the gaps inside the pump has a direct effect on the volumetric efficiency, hydraulic-mechanical efficiency and on the total efficiency of the gear pump. Volumetric efficiency is defined as a quotient of the actual flow generated through the pump and of the ideal flow:

$$Q_{ideal} = q \cdot n; \tag{1}$$

$$\eta_v = \frac{Q}{Q_{ideal}}. \tag{2}$$

The flow generated through the pump defined as ideal flow minus leakage:

$$Q = Q_{ideal} - Q_{leak}. \quad (3)$$

The flow due to leaks has two main components: axial gap flow and circumferential gap flow. Axial gap compensation has been a solution used in gear pumps for years. It minimizes the gap height and as a result – limits the leaks. The principle behind this solution is as follows: Liquid is delivered from the discharge side to the precisely defined surface of the bearing blocks. In result, the generated force moves the bearing blocks towards the gears. According to Borghi *et al.* (2009), owing to such a solution, the leaks through the gap depend mainly on the pressure difference. This compensation is commonly used in gear pumps and proves effective at limiting the axial gap leaks. Therefore, this research focuses on the circumferential gap.

Circumferential gap flow is identified with the use of various models. The first modelling method is to use an equation for a sharp-edged orifice. Such an approach is presented by Borghi *et al.* (2009) and Wahab (2010). Assuming that the gap is narrow, its edges are sharp, and the flow is turbulent, the flow rate is described with the following relationship:

$$Q_c = C \cdot A \cdot \sqrt{\frac{2 \cdot \Delta p}{\rho}}. \quad (4)$$

Schiffer *et al.* (2013) offer another approach to the problem. In order to simplify the analysis, both the gear and the casing walls were unfolded on a plane, which procedure is justified by the limited width of the teeth and the small height of the gap relative to the height of the teeth. The circumferential gap is in such case treated as a gap with one moving wall. The total flow through the gap is then determined as the sum of the flows due to pressure gradient (Poiseuille) and due to the rotation of the gears (Couette). The flow is described with equation:

$$Q_c = \frac{b \cdot h}{2} \cdot \left(\frac{\Delta p}{6 \cdot \nu \cdot \rho \cdot r_s \cdot \pi} \cdot h^2 - v \right). \quad (5)$$

Rituraj *et al.* (2018) also define the circumferential gap flow as the sum of flows described by the Poiseuille and Couette equations. They also observe that the share of individual flow components changes depending on the rotational speed of the pump and on the pressure difference. In the case of pumps operating at high pressures and in a medium range of rotational speeds, the Poiseuille flow is the dominant component, while in the case of low pumping pressures and high rotational speeds (pumps in fuel injection systems), the components are comparable or the Couette flow component dominates. The authors indicate that the precise modelling of pump instantaneous parameters requires consideration for the fact that the flow through the gap is turbulent (low viscosity liquids and a sufficiently large circumferential gap). In such case, the pressure drop at the gap entry becomes a significant phenomenon, as it accounts for up to 95% of the total pressure drop along the entire length of the gap.

Pump wear is an important aspect addressed by Guo *et al.* (2020). The flow rate is variable in time, as the number of displacement elements is finite. This non-uniform delivery results in pressure pulsation and impedance of the hydraulic system. In effect, the forces due to pressure acting on the pump elements are also variable. Deformations of pump elements (e.g., of gear shafts) cause the lubricant film between the tooth tip and the casing to be discontinued – the casing is thus subjected to wear and the circumferential gap increases.

Regardless of the assumed circumferential gap flow model, the key factors responsible for the flow rate seem to be: pump rotational speed, pressure difference between the discharge port and the intake port, the viscosity and density of the liquid, the geometric dimensions of the gap and the wear degree of the pump. Of all the above-listed parameters, particular attention should be paid to the gap height and to the pressure difference between the pump ports. These two factors have a significant influence on the extent of the internal leaks. Even a limited increase in the pressure difference or in the gap height causes a significant decrease in the flow rate generated by the gear pump (Borghi *et al.* 2009; Guo *et al.* 2020; Szwemin, Fiebig 2021). Therefore, a precise model of this flow rate and of the volumetric efficiency of the pump necessarily depends on the precise knowledge of the height of the circumferential gap.

This height can be determined following a number of methods. What seems to be currently the most accurate method is presented by (Rituraj *et al.* 2018; Vacca, Guidetti 2011). The HYGESim tool developed by the authors identifies instantaneous forces acting on the gears and uses this information to determine the position of the gears with respect to the pump casing. An important aspect motivating the presented research is the fact that the above articles do not address the deformation of the pump elements as a factor in determining the size of the circumferential gap. Some of the authors assume in their analyses that pump elements are rigid and do not deform, and some of the authors do not mention this issue at all.

In this article the authors present the use of FEM, which is frequently used in numerical strength calculations (Cieślicki *et al.* 2018, 2019; Ghionea *et al.* 2012; Kollek *et al.* 2017; Osiński, Warzyńska 2022), as a method to find the height of the circumferential gap in an external gear pump with consideration the deformation of pump elements. The aim of the work is to show that the deformation of pump elements significantly affects the size of the circumferential gap. For this purpose, a geometric and a discrete model was prepared of a 3PZ4 gear pump manufactured by WPH, which is popular in machine building and is offered by many manufacturers under different trade names with similar operating and design parameters. The calculations were performed for the discharge pressures of 8, 16, 24, and 32 MPa. The model was validated in strain measurements of pump casing for discharge pressures of 24 and 32 MPa.

The model here presented is a successive stage of research on numerical modelling of gear pumps. The model has been significantly modified with respect to the version presented previously (Cieślicki *et al.* 2018, 2019). The modifications are described below.

3. The research object

The geometric and discrete models were constructed for an external gear pump type 3PZ4, which is a typical design with pressure axial compensation. The most important parameters of the pump are shown in Table 1. The investigated gear pump (Figure 1) consists of two gears set in plain bearings. The floating bearing blocks are located in the pump casing. The pump is assembled by bolting the front plate and the cover plate. The model does not involve the sealings.

Table 2 shows the material properties required in the calculations. The gears are made of alloy steel. The casing and the bearing blocks are made of aluminum alloy. The plate and the cover are made of gray iron. The pump has composite plain bearings comprising a steel sleeve, on which there is sintered a porous thin layer of tin or bronze. During the rolling process, the pores are filled with PTFE. As the steel sleeve forms a large part of the bearing, the model is based on an assumption that the entire bearing is made of steel. In the previous research articles, the bearing was not included, and instead the diameter of the shaft necks was increased.

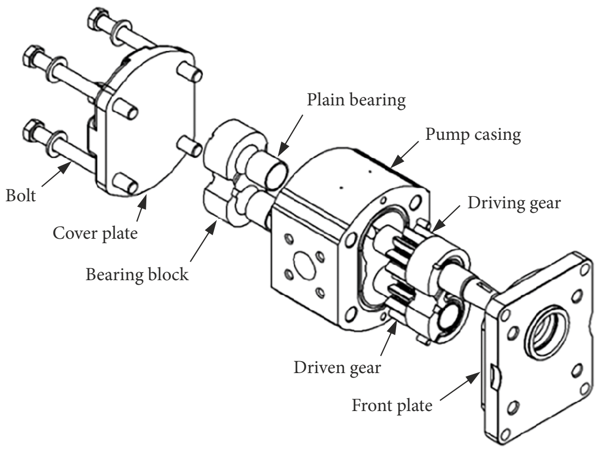

Figure 1. External gear pump type 3PZ4 manufactured by WPH (exploded view)

Table 1. Parameters of the tested gear pump

Pump parameters	Value
Number of teeth [–]	10
Tip diameter [mm]	52
Tooth width [mm]	1
Pressure angle [degree]	20
Maximum continuous discharge pressure [MPa]	28
Maximum instantaneous discharge pressure [MPa]	32

4. Numerical model of the gear pump

The numerical model (Figure 2) was prepared with the use of the ABAQUS software (*https://www.3ds.com/products-services/simulia/products/abaqus*). The discrete model was constructed by dividing the geometric model into solid finite elements. Both the casing and the gears are made of tetrahedral elements. As compared to the model presented in the previous articles, this model has a significantly increased mesh density on the inner side of the casing and on the teeth tips, thus generating a large amount of data required when calculating the height of the circumferential gap. This type of finite element was chosen because gear pumps have a complex geometry containing many curves. In this case, the use of a mesh composed of tetrahedral elements and its local densification allowed to obtain a good reflection of geometry with a relatively low number of distortions. The casing consists of 2251695 elements and 423543 nodes, and each gear has 138508 elements and 27874 nodes. The model comprises a total of 3408874 elements and 727518 nodes.

The pump was fixed to the plate with bolt connections loaded in tension. The plate was fixed with the use of a boundary condition, preventing movement in three directions. The bolts connecting the pump elements were loaded with the tensile force resulting from the assembly

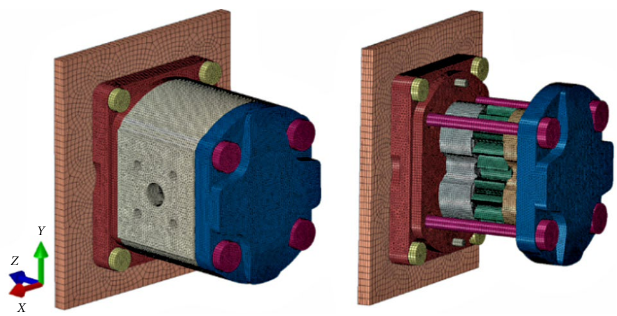

Figure 2. Numerical model of the gear pump

Table 2. Properties of the materials used in the pump elements

Material	E [GPa]	ν [–]	$R_{m\,\min}$ [MPa]	$R_{0.2\,\min}$ [MPa]
Steel	210	0.30	1000	700
Aluminum alloy	72	0.32	470	400
Aluminum alloy	72	0.32	80	120
Gray iron	110	0.33	250	–

torque provided in the technical specification of the pump. The following relationship was used to calculate the tensile force (Bijak-Żochowski *et al.* 2017):

$$F = \frac{2 \cdot M}{d_s \cdot \tan\left(\gamma + \rho'\right) + d_m \cdot \mu}. \tag{6}$$

Surface-to-surface contact points were modelled between the pump elements. The driven gear was prevented from rotating around axis Z with the use of a MPC.

The conditions assumed in the model correspond to the operation of the pump when pumping liquid at a pressure of p_t. The simulations were performed for four discharge pressure values: 8, 16, 24, 32 MPa. The pump elements were loaded with pressure. When loading the casing, a linear pressure increase was assumed as a function of angle ϕ on the intake side and a constant pressure was assumed on the discharge side (Figure 3a). The intake and the discharge sides are divided with the use of a plane defined by the rotation axes of the gears.

The modelling method of the gears is another important difference with respect to the earlier research. Previously, the gears were modelled in the form of cylinders having a diameter equal to the tip diameter of the gears. In the current version, the gears are fully modelled. As the specific delivery q of the pump varies with the rotation of the gears, the torque on the pump shaft also varies, as described in the following equation:

$$M_p = \frac{q(\phi) \cdot \Delta p}{2 \cdot \pi}. \tag{7}$$

For this reason, an assumption was made in the model that the gears are in a position in which the torque has an average value. The gear loading method was also modified. Originally, the gears were loaded with forces calculated analytically on the basis of linear pressure distribution on the intake side and of constant pressure on the discharge side. Currently, the surfaces of the gears are loaded with pressures in accordance with the following relationship:

$$p_i = \frac{i}{3} \cdot p_t. \tag{8}$$

The i takes a value from 1 to 3. All of the teeth located on the discharge side are loaded with a discharge pressure. The suction pressure at the inlet port was neglected due to its insignificant value in relation to the discharge pressure. The model does not take into account the effects of the running-in process of the gear pump.

5. Simulation results

Load acting on the pump causes its elements to deform. Figure 4 shows the deformation of the pump casing for the pressure of 32 MPa (grey colour) with respect to the undeformed casing (black colour). The deformation scale factor is 10. The deformation is significantly greater on the discharge side than on the intake side. The deformation of the casing on the discharge side is the direct result of the pressure acting on the casing walls. The intake side

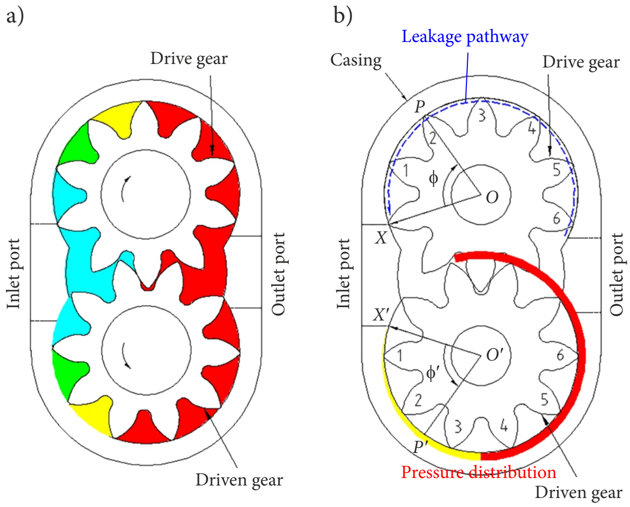

Figure 3. Pressure distribution on the pump casing (a) and on the gears (b), and the coordinate systems used in the presentation of the simulation results

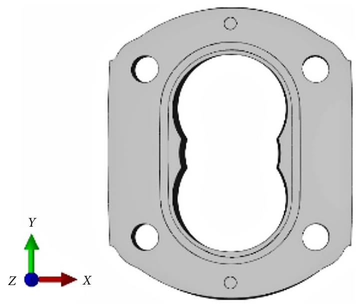

Figure 4. Casing deformation for discharge pressure of 32 MPa (deformation scale factor: 10)

deforms as the bearing blocks and gears press against the inner surface of the casing. As a consequence, the already discussed circumferential gap forms inside the pump, negatively influencing the delivery and the efficiency of the gear pump.

The simulation allowed the identification of the size of the gap around the circumference of the casing. Six teeth were selected on each of the gears between the intake and the discharge ports (Figure 3). The distance from the casing surface to each node on the tooth tip was calculated. The total number of nodes on the tip of each tooth was 133.

Figures 5 and 6 show the change of mean gap height as a function of the discharge pressure. For teeth from 1 to 3, the gap height decreases with the growth of the pressure. For the first tooth past the intake port, the gap height was observed to have a minimum value (p = 8 MPa) or to be absent (p > 8 MPa). As can be clearly seen, the gears become displaced towards the intake side under the influence of the forces due to the pressure distribution inside the pump. In effect, an increase in the discharge pressure causes the gap height to decrease on the intake side of the pump. For teeth from 4 to 6 on the discharge side, the gap height increases with the growth of the pressure. For 8 MPa, the maximum gap height is 0.149 mm (tooth 6) and increases until it reaches its maximum height of 0.343 mm for the driven gear, at the discharge pressure of 32 MPa. The differences between the driving gear and the driven gear are insignificant. In the case of the driving gear, the maximum gap height is 0.338 mm.

The influence of the pressure on the gap height can be thus clearly observed. Increasingly higher discharge pressures cause the gap to decrease on the intake side and to increase on the discharge side. This fact is due to two phenomena. Firstly, the pressure causes the gears to press against the casing on the intake side. Secondly, the pump elements deform. In order to better illustrate these phenomena, Figures 7 and 8 show the average distances of the nodes forming the teeth tips (e.g., distance *PO* in Figure 3) and of the nodes forming the inner casing surface (e.g., distance *P′O′* in Figure 3) from the centre of the gear in its initial position (unloaded) as a function of angle of rotation of pump shaft.

When the load is not applied, the distance *r* is 26 mm for the casing, and for tooth tips this value is reduced by the tolerances. At 8 MPa, the deformation of the casing is limited and only slightly above 0.05 mm. For low discharge pressures, the gap height depends equally on the displacement of the gears and on the deformation of the casing. On the other hand, for higher pressures, the deformation is greater and has a decisive influence on the gap height. The deformation of the casing can be also observed to be much greater on the discharge side than on the intake side, which seems natural in the view of the pressure distribution inside the pump.

Figures 9 and 10 show the change of average gap height as a function of ϕ. The data presented in this form allow an observation that the division into the intake side and the discharge side takes place on the first tooth passing the intake port, and that tightness is preserved as the

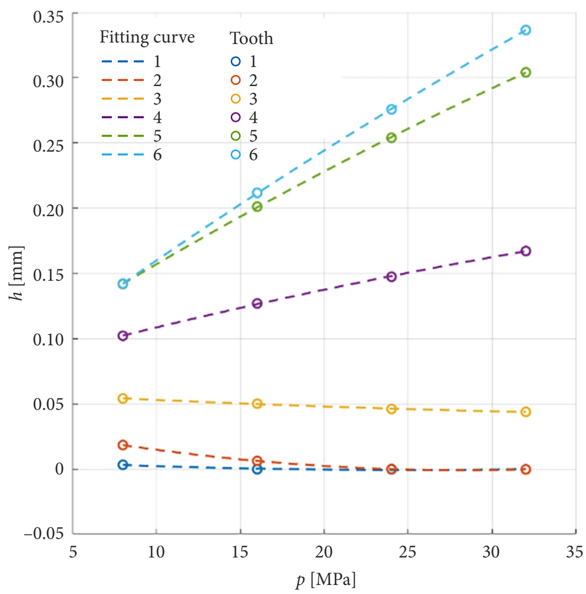

Figure 5. Change of mean gap height as a function of the pressure for the driving gear together with fitting curves

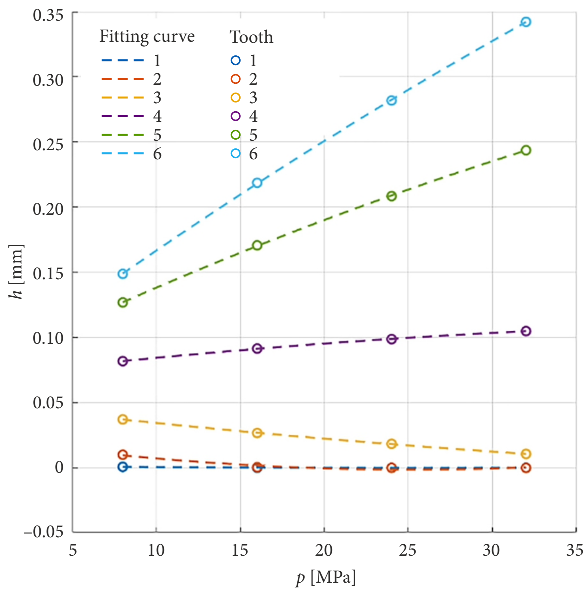

Figure 6. Change of mean gap height as a function of the pressure for the driven gear together with fitting curves

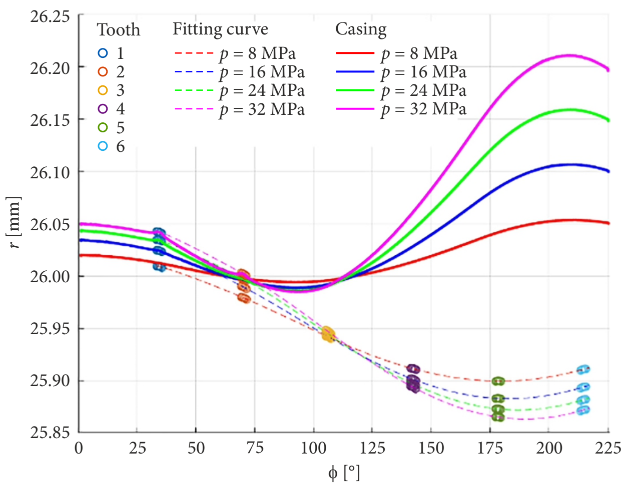

Figure 7. Distances of the tips in the driving wheel and the inner casing surface from the axis of rotation of the gear (unloaded)

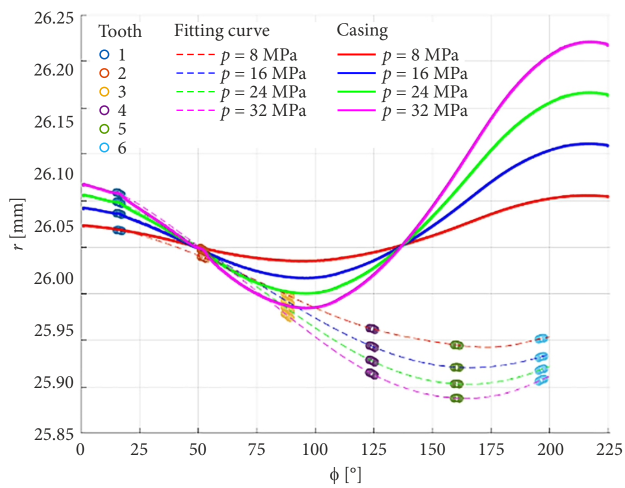

Figure 8. Distances of the tips in the driven wheel and the inner casing surface the axis of rotation of the gear (unloaded)

gears press against the pump casing due to pressure difference between the intake side and the discharge side. An increase of the circumferential gap height is observed towards the discharge port.

As already mentioned, the size of the circumferential gap has a significant influence on the level of flow through the gaps inside the pump. This in turn translates into the decrease of the generated flow rate and as a result – into a decrease in the volumetric efficiency of the pump. The deformation of the casing has a significant influence on the gap size and cannot be ignored in the modelling of the flow generated by pumps operating at high pressures. This deformation should also be considered in the case of pumps made of materials having low Young's modulus, as they would deform more at a similar load.

6. Validation of the model

The model was validated by measuring pump casing deformations in operating conditions and for discharge pressures of 24 and 32 MPa. The measurements were performed with the use of seven self-compensated strain gauges and the HBM QuantumX MX1615 amplifier. Figure 11a shows the locations of the measurement points.

A hydraulic system (Figure 11b) ensures that the pressure is constantly regulated on both the intake and the discharge sides. The tested pump 1 is driven with an electric motor 2. On the intake side, the pressure is regulated by a system comprising a feed pump 3 and adjustable throttle valves 9 and 10. The load on the pump is applied through a throttle valve 6. A safety valve 7 protects the tested pump against extensive discharge pressure.

The suction pressure is controlled with a manuvacuometer 5, and the discharge pressure – with a manometer 4. The torque was measured with a torque meter 8, which also allows measurements of the rotational speed of the shaft. A flowmeter 11 measured the flow rate generated by the tested pump. The oil temperature in the tank was controlled with a temperature sensor. Temperature changes were insignificant due to the short measurement time and the high heat capacity of the system. Prior to the

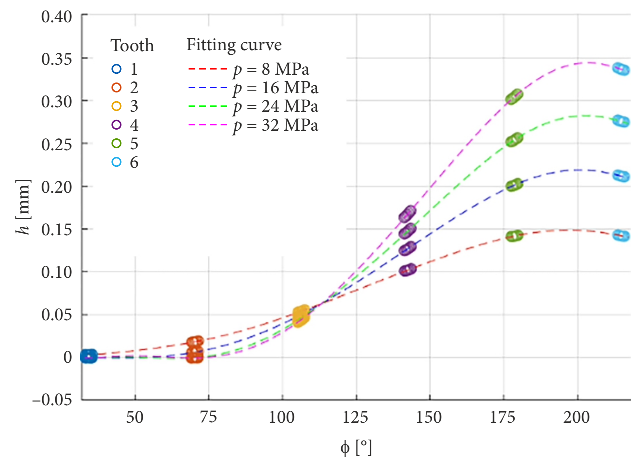

Figure 9. Height of the circumferential gap as a function of an angle of rotation of pump shaft for the driving gear together with the fitting curves

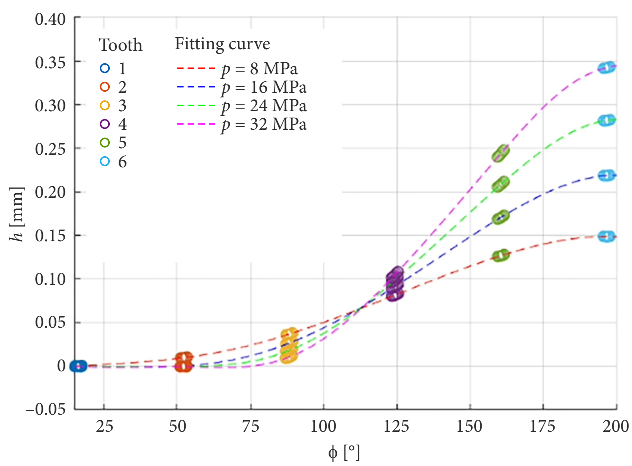

Figure 10. Height of the circumferential gap as a function of an angle of rotation of pump shaft for the driven gear together with the fitting curves

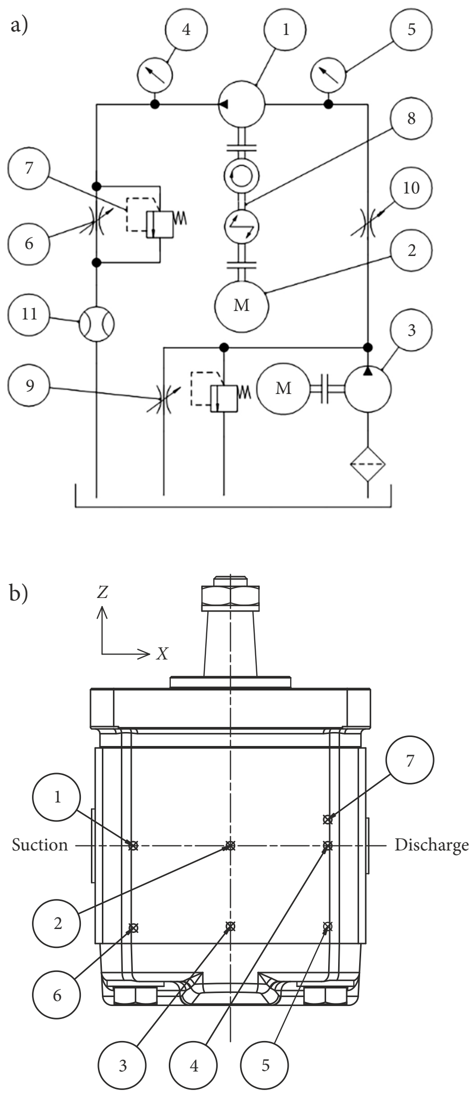

Figure 11. Schematic diagram of a hydraulic system for generating loads on the tested pump (a); locations of the strain gauges on the tested pump (b)

tests, the pump was subjected to a break-in cycle in accordance with the recommendations of the manufacturer.

The measurement of the casing deformations was started with the disabled test stand and with loosened bolts connecting the casing elements. Subsequently, the bolts were tightened with a torque spanner, while measuring the deformation of the casing due to bolt tension. The test stand was then activated while the measurements were continued. In the next step, the discharge pressure was gradually increased up to 32 MPa.

Figure 12 shows the measured deformations (Cieślicki *et al.* 2019) and the results of the FEM calculations. Letters *X* and *Z* indicate directions in accordance with the applied coordinate system. Observations demonstrated very good similarity of the numerical model with the results of the experiment.

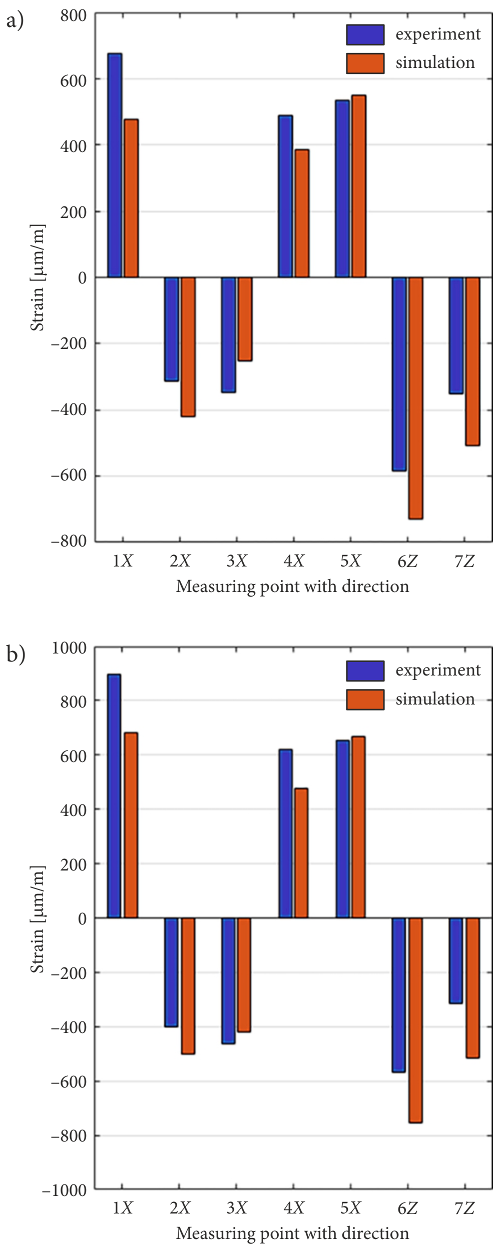

Figure 12. Pump casing deformations under a load of: a – 24 MPa; b – 32 MPa

Conclusions

This article presents the results of FEM numerical calculations. The method is typically used to determine the effort of a structure. The article presents a numerical model of an external gear pump, which allows the identification of the circumferential gap height during the pumping process. The size of the circumferential gap is shown for the discharge pressures of 8, 16, 24 and 32 MPa.

The gap height varies along the circumference of the gear. The pump is sealed on the first tooth, which passes the intake port of the pump. Subsequently, moving along the circumference in accordance with the rotation direction of the gears, the gap height increases and reaches its maximum height in the vicinity of the pump discharge port.

The simulation results demonstrated that the size of the circumferential gap depends not only on the displacement of the gears and bearing blocks, but also on the deformation of the pump elements under increasing load. A change of load pressure from 8 to 32 MPa causes the maximum gap height to increase almost twice. The differences between the driving gear and the driven gear are insignificant.

The simulation results indicate that the circumferential gap is zero for the first tooth for both driving and driven gear. It results from the adopted simplification. The model does not take into account the effects of the running-in process of a gear pump. However, the error resulting from this simplification is small because a gap greater than the height of the oil film on the first tooth passed the inlet port would cause a discharge to suction leakage, which would ultimately prevent pressurized fluid from being pumped.

The simulations of the flow rate generated by the pump and of the volumetric efficiency are based on the ability to predict the extent of internal leaks. The use of the FEM allows a more precise identification of the gaps in the pump. Regardless of the method applied to model the flow rate through the circumferential gap, its height remains an important parameter. In the case of high discharge pressures, deformations of the pump casing significantly influence the size of the gap. Ignoring pump deformation when modelling the flow may cause the flow rate through the gaps to be underrated. As a consequence, the delivery and the efficiency of the gear pump may be overrated.

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Author contributions

Rafał Cieślicki:
- »» created the concept of research.
- »» was responsible for the design and development of the data analysis.
- »» was responsible for data collection and analysis.
- »» was responsible for data interpretation.
- »» visualized the research results.
- »» wrote the first draft of the article.

Mykola Karpenko:
- »» reviewed and edited the article.
- »» supervised the research.

Disclosure statement

The authors declare no conflict of interest.

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