such loads determine the safety of operation of the undercarriage. But the magnitudes of the forces have not been precisely identified yet. Therefore further experimental re-search is needed to make the design of such systems more accurate.

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DEVELOPMENT OF THE FLUID POWER GEAR MACHINES

ABSTRACT

Clarification of the fluid power gear machines is presented in the article. The machine are divided into four groups, ie. The external gearing involute fluid power machines, the internal gearing involute fluid power machines, the internal gearing cycloidal machines, and the machines featuring a number of gears. Findings of the research carried out by the FPRG from the Technical University of Wroclaw concerning the development of all types of the machines mentioned above have also been presented in the paper. Furthermore, the new types

of the fluid power gear machines as well as the results of the experimental research on the machines have been presented.

KEYWORDS
Fluid power machines, internal gearing, external gearing, involute gearing, cycloidal gearing, pulsation

I INTRODUCTION

Fluid power (hydraulic) gear machines are well known and commonly used in the design of hydraulic motor systems of machines and appliances.

Figure 1 provides the illustration of analysis of the present state and development tendencies of the gear machines. The machines can be divided into four groups, with the assumption of the following division criteria: the kind of gear, the tooth profile, and the collaboration principle of the gears in the machine.

The first group (figure 1a) includes external involute gearing machines, in which the gears work at the fixed axis. These machines function as pumps or high-speed hydraulic motors.

The second group (figure 1b) includes internal involute gearing machines, in which the gears have also determined axis. The machines work as pumps or high-speed hydraulic motors. These units, however, are of a more compact design, lower weight, and also lower displacement pulsation, as well as lower noise level while at operation.

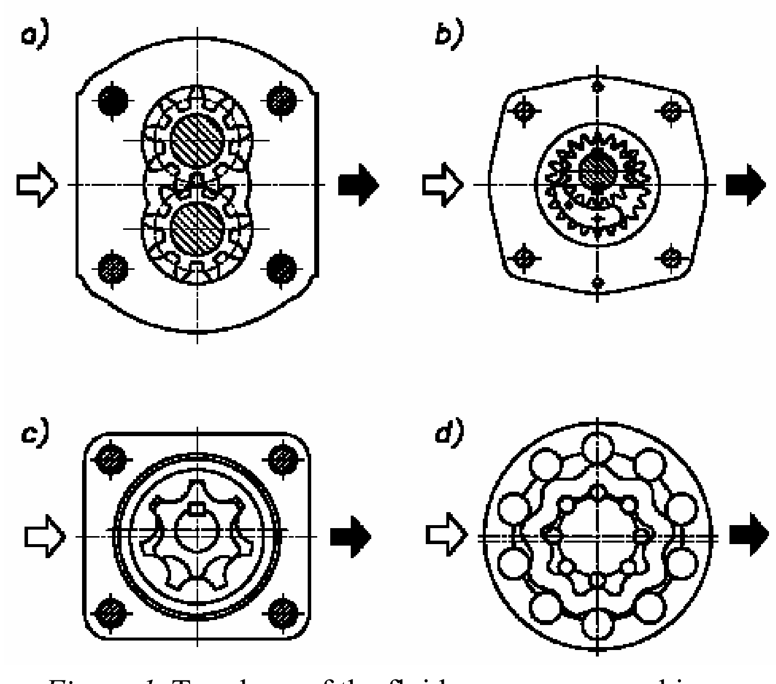

Figure 1. Typology of the fluid power gear machines:
a) machines of the 1st group featuring external involute gearing,
b) machines of the 2nd group featuring internal involute gearing,
c) machines of the 3rd group featuring internal cycloidal gearing,
d) machines of the 4th group featuring numerous gears

The third group (figure 1c) consists of internal cycloidal gearing machines, the gears of which can work both at the fixed as well as at the moving axis (orbital motion). These machines include pumps and high-speed gerotor motors.

These units are of even more compact design than the machines of the second group mentioned above, and of lower weight, lower pulsation, and lower operation noise level. Apart from that, applying the internal gearing of the tooth difference z_2-z_1=1 in the machines enables developing of the following design solutions: low-speed high-torque orbital motors and control systems.

Finally, the fourth group of the machines (figure 1d) is special for the multi gear systems both of the internal and of the external gearing used in the design, which can work at the fixed as well as at the moving axis. This group includes low-speed high-torque orbital motors featuring extremely high displacements and high torques, and also multifunctional fluid power gear machines.

Since the 70s of the 20th century, in the Institute of the Machine Design and Operation at the Technical University of Wroclaw, a group of engineers called the Fluid Power Research Group (FPRG) (Stryczek, 2005), under the leadership of this article's author, have been working on developing the already existing hydraulic gear machines, and also creating and improving the novel designs.

II MACHINES OF THE FIRST GROUP

Machines of the first group stand for the classic design that originates in the Keepbie's pump from 1604 (Abel, 1971).

b) efficiency characteristics of PZ3–10 pump

Figure 2a depicts a design of the modern external involute gearing pump which is a representative of the PZ3 series, displacements of q=1-100 cm^3/rev and working pressure p_n=20MPa. The pump series was made as a result of cooperation between PZL-Hydral Company, Wroclaw and the Fluid Power Research Group (FPRG) from the Technical University of Wroclaw. The fundamental development work on the design was focused on designing a special external involute gearing (Kollek and Stryczek, 1978) that would fulfill the following requirements: compact design, low displacement pulsation, little volume of the closed space. Experiments concerning optimization of the axial clearance compensation in the

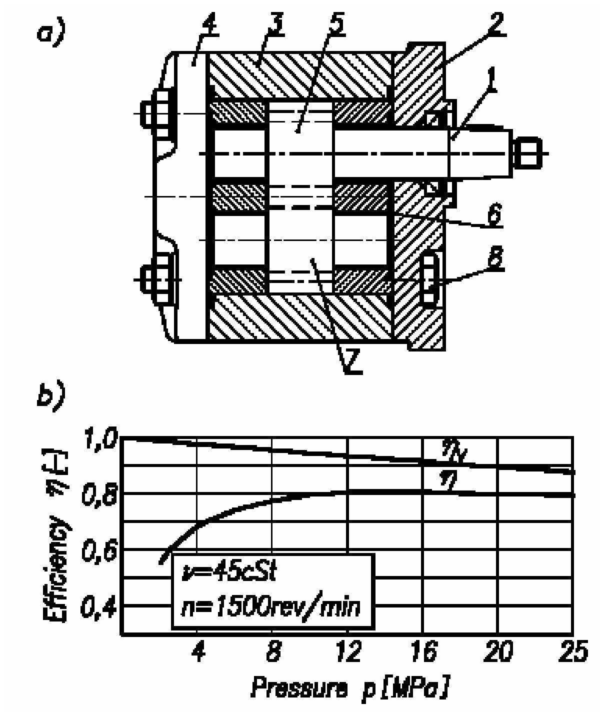

Figure 2. Gear pump PZ3 featuring external involute gearing:
a) view of the pump (1-shaft; 2,3,4 – elements of the body; 5 – driving gear; 6 – slide bearing; 7 – driven gear; 8 – screw),

gear machine were also carried out. As a result, the pumps could work continuously at pressure p=20MPa and the maximum pressure p_{max}=25MPa, keeping the volumetric efficiency η_v=0.9, and the total efficiency η=0.8 (figure 2b). The units can also be used in hydraulic systems as hydraulic motors.

III MACHINES OF THE SECOND GROUP

One of the basic evaluation criteria of the machine is the coefficient of the displacement irregularity (inlet flow rate) δ. Using (Stryczek, 2007) and having made all the necessary calculations, values of the coefficient for the machines of the first and the second group were compared (figure 3).

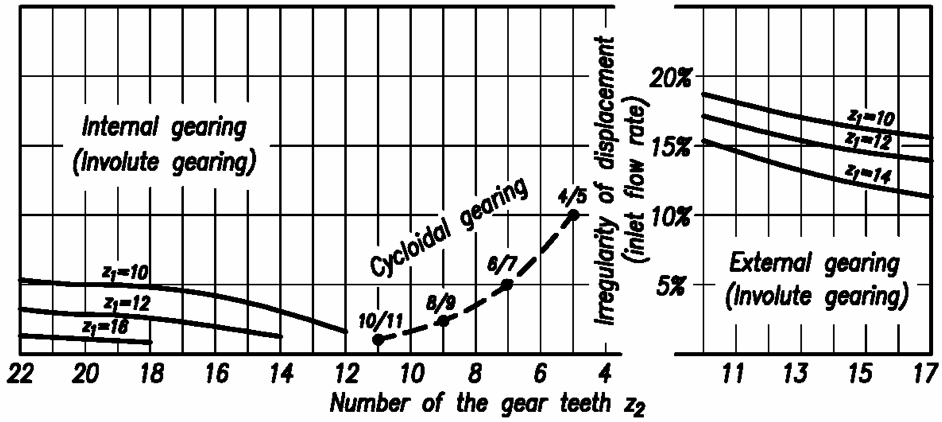

Figure 3. Displacement (inlet flow rate)irregularity coefficient of the fluid power machines featuring internal and external gearing

The figure shows that the pulsation of the displacement (of the inlet flow rate) of the second group machines with the internal tooth system is lower than the one of the first group machines with the external tooth system, which might result in quieter work. In relation to this, FPRG have developed a special computer program for designing the second group machines, named Fluid Power Involute Gears (Greczanik and Stryczek, 2006). The program enables to make geometrical, kinetic, and hydraulic calculations as well as simulation of the gear manufacture. By means of the computer program, an internal gearing pump ZW-4 has been designed. The pump's displacement $q=4cm^3/rev$ and the working pressure p_n=14MPa (figure 4a). In order to additionally lower the pump's displacement pulsation, a large number of the driven gear teeth (z_1=16) has been assumed in the design, which, as shown in figure 3, ensures achieving the pulsation coefficient δ=1-2%. The ZW-4 pump has been produced by the Hydraulic Pump Manufacturer in Wroclaw, Poland (Wytwornia Pomp Hydraulicznych we Wrocławiu). The tests of the machine have proven it to show the assumed displacement $q=4cm^3/rev$ and the working pressure p_n=14 MPa.

The most important, though, is that, as shown in figure 4b, the noise level of ZW-4 is low and clear t_A=62-64 dB(A). It is 4-6 dB(A) lower than the noise level of the external tooth system gear pump featuring equal displacement $q=4cm^3/rev$ and belonging to PZ3 pump series (figure 2a). Thus, the low displacement pulsation of ZW-4 pump has been confirmed. At present, work on the development of the internal involute gear pump series within the scope of high displacements q=40, 50, 60 cm^3/rev is being conducted.

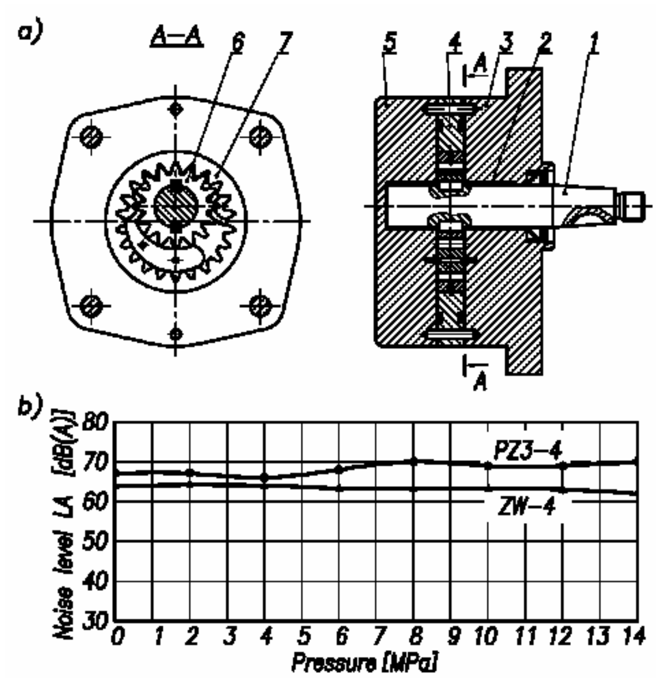

Figure 4. Gear pump ZW-4 featuring internal involute gearing:
a)view of the pump (1-shaft; 2-bearing; 3,4,5-elements of the body; 6,7-gears),
b)noise characteristics of ZW-4 pump featuring internal gearing juxtaposed with the characteristics of PZ4 featuring external gearing

IV MACHINES OF THE THIRD GROUP

Further development of the internal gearing machines is possible, provided that the cycloidal profiles are applied in the process of designing. The theoretical background for designing of the internal cycloidal gears is presented in (Stryczek, 2003). It shows that in the hydraulic machines it is possible to apply the cycloidal gears with a small number of teeth of the driving gear equal $z_1=6\div10$ and the teeth difference $z_2-z_1=1$. These numbers are smaller than the ones used in the internal involute gears where $z_1\geq10$ and $z_2-z_1\geq4$. Using the internal cycloidal gear eliminates the necessity of applying the sickle-shaped insert that is normally present in the involute gears, which lowers the overall dimensions and weight of the machine. At the same, what is achieved is lowering of the value of the displacement (inlet flow rate) irregularity coefficient, which is depicted in figure 3 with a dashed line for selected numbers of teeth z_1/z_2=6/7, 8/9, 10/11. In order to achieve high volumetric efficiency of the machine, rules of designing axial clearances in relation to the internal cycloidal gearing have been developed (Stryczek, 2005 and Bednarczyk and Stryczek, 2004). As a result, the FPRG have developed a special computer program for designing cycloidal gear machines named the ‘Fluid Power Cycloidal Gears’ (Antoniak, Stryczek, 2006).

By means of that computer program a series of gerotor pumps (the symbol of the series PGK) featuring the axial clearance compensation has been made. The series includes q=10, 20, 40 cm^3/rev units working at the nominal pressure p_n=20 MPa.

The patent pending design of the pump (Bednarczyk, Pietrus and Stryczek, 2004) is presented in figure 5a. The experimental research on the pumps (figure 5b) has proven the cycloidal gear and the compensation system to work properly, and the unit was working continuously at pressure p=20MPa with volumetric efficiency η_v=0.8 and total efficiency η=0.7.

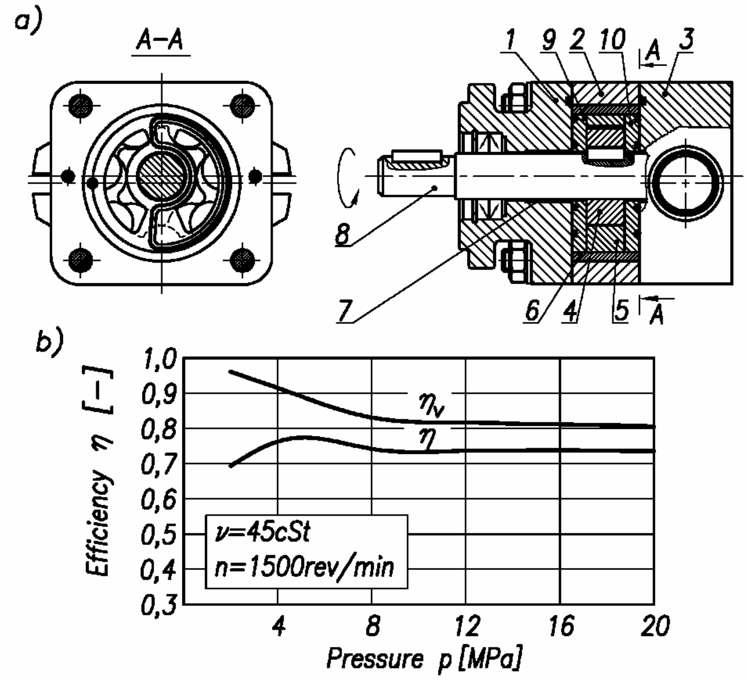

Figure 5. Gerotor pump PGK with the axial clearance compensation:
a) pump design (1,2,3 – bodies; 4,5 – cycloidal gears; 6,7 – bearings; 8 – driving shaft; 9,10 – compensation plates),
b) characteristics of the pump's efficiency

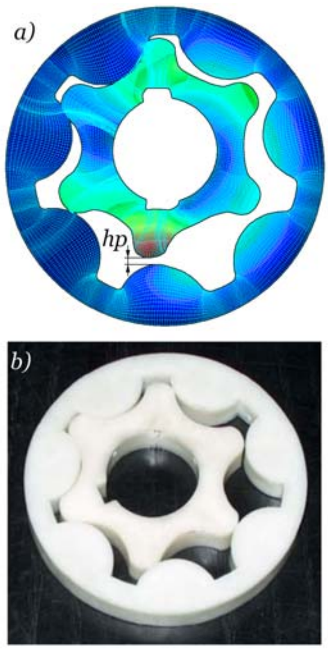

Figure 6. Internal cycloidal plastic gear system
a) deformation of the gears and the radial clearance hp between the gear teeth,
b) view of the polyoxymethylene (POM) gears

A serious problem connected with the manufacture of the gerotor pumps is the technology of the cycloidal gearing. So far, the gears have been made of steel, employing the classic machining methods (milling, pull broaching, grinding) (Stryczek, 1994), which is costly and labor consuming. Taking all the facts into consideration, the FPRG took up the task of developing the design and technology of the plastic cycloidal gears. From the design viewpoint, it is crucial to study the deformations of the plastic gears caused by the loads resulting from the pump operation. For the research, polyoxymethylene (POM) was used, and the deformations were determined by means of the computer simulation using the finite elements method (MES) (Biernacki and Stryczek, 2007). The results are presented in figure 6a.

The figure shows that torque M and pressure p working on the gear set cause the deformation of the external tooth gear (asterisk) in the direction of the suction zone of the pump, and the teeth of the gear are pressed to the driven gear (ring). In the consequence of the process, radial clearance hp=0.05÷0.1mm causing internal leakages and lowering the efficiency of the pump. In relation to the problem, the research on the selection of a plastic that would provide durability and keeping the original profile of the gear. The technological work included preparation of the form and making the gears by means of injection moulding. The view of the gears made of POM is presented in figure 6b. The gears were mounted in the gerotor pump where they were working properly within the low pressure range of p=1÷2MPa. Using the plastics is aimed at simplifying the technological process of the gear production, lowering the noise level as well as enabling pumping of water and water-oil emulsions.

Introduction of the internal cycloidal gears of the teeth difference z_2-z_1=1 made it possible to realize the orbital gear motion that became the operational basis for the new types of machines of the third group, that is of the high-torque low-speed orbital motors and of control systems.

With the use of the above mentioned computer program, the Fluid Power Research Group have designed internal tooth cycloidal gear systems for ORBIT motors produced by PILMET Company in Wroclaw.

V MACHINES OF THE FOURTH GROUP

The above discussed fluid power gear machines are equipped with the set of two gears. The FPRG have taken up an attempt of constructing a machine featuring a larger number of gears collaborating with each other according to the orbital motion principle.

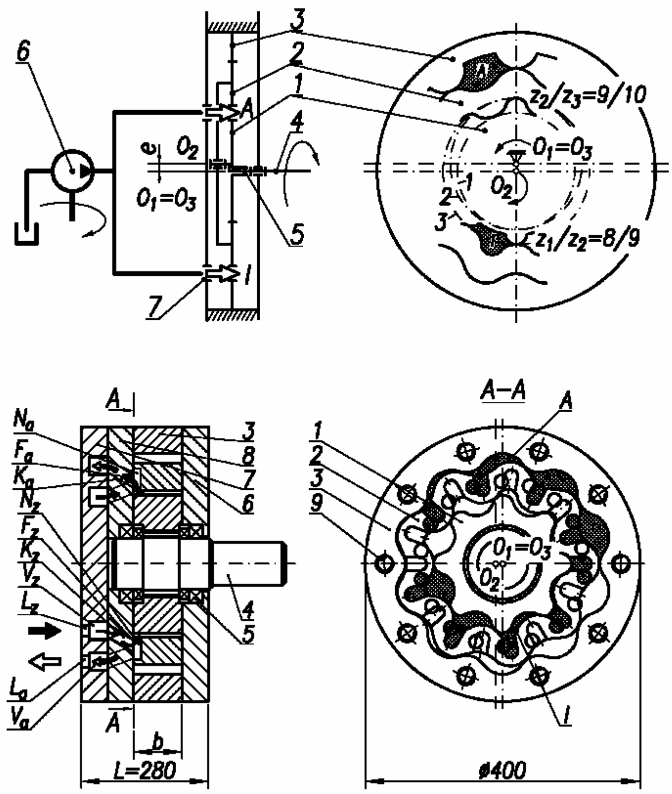

Figure 7. Orbital motor featuring the double cycloidal gear system:
a) conceptual draft of the motor (1,2,3 – gears; 4 – shaft; 5 – cage; 6 – pump; 7 – channel; A, I – displacement chambers),b) design and operation principle of the motor (1,2,3 – gears; 4 – shaft; 5 – bearing; 6,7,8 – body elements; 9 – screws)

Realizing the assumptions, the group, according to (Balawender ans Stryczek, 1999) have developed the concept of the orbital motor featuring the double tooth system cycloidal gear, presented in figure 7a. The motor consists of three gears: central gear (1), externally toothed with O_1 centre, gear ring (2) externally and internally toothed with O_2 centre, as well as solar gear (3) internally toothed with $O_3=O_1$ centre. Between the gears there are two gearings – one between the central gear (1) and the internal tooth system of the gear ring (2), and the other between the external tooth system of the gear ring (2) and the solar gear (3). All teeth in both tooth systems are in continuous contact which enables the mutual driving of the gears and the cage (5) can be eliminated. In figure 7a the dashed line indicates it as imaginable. The teeth in both gears make each other closed displacement chambers which in reference to the first tooth system is labeled as I, whereas to the second, as A. The chambers are charged with oil at a high pressure from the pump (6) through a delivery channel system (7). The pressure of the oil initiates the orbital motion of the gear ring (2) within the eccentricity $O_1O_2=e$. The consequence of the motion is the rotary motion of the gear (1) and the collaborating driving shaft (4). A solution corresponding with the concept is presented in figure 7b.

The motor features a front plate (6), a central plate(3), a back plate (8), and a cover (7). In the central plate (3) the internal tooth system of the solar gear z_3 has been shaped, and then inside of the gear, the ring (2) with the double external and internal gearing z_2 and the central gear (1) with the external tooth system z_1 have been placed. The design of the cycloidal gears was made by means of the computer program ('Fluid Power Cycloidal Gears').

The cycloidal gear set makes two gearings with the number of teeth $z_1/z_2=8/9$ and $z_2/z_3=9/10$. It passes the torque on to the driving shaft (4) located in bearings (5). Owing to the orbital motion of the gears it is necessary to design a special timing, namely the system of charging the chambers between the teeth with oil, and then discharging it to the tank. The system consists of:

- delivery port L_z and discharging port L_a,
- delivery collector V_z and discharging collector V_a,
- delivery channel K_z and discharging channel K_a,
- delivery orifice F_z and discharging orifice K_a,
- delivery channels N_z and discharging channels N_a of the directional control valve

The working fluid is delivered to the motor through the delivery port L_z, and then it flows in the delivery collector V_z to delivery channels K_z, and further on through the delivery orifice F_z as well as through the channels of the directional control valve N_z it gets into the displacement chambers I and A between the gear teeth. As shown in figure 7a, only half of I and A chambers is charged. The other half is at the same time connected to the discharge that is realized on the way (N_a), (F_a), (K_a), (V_a) and (L_a).

Findings of the experimental research on the motor have been presented in table 1. It is necessary to emphasize the fact that at a relatively small size (see figure 7b – diameter D_z and length L_z), the motor achieves the displacement (inlet flow rate) q=9000 cm^3/rev. This displacement (inlet flow rate) can be increased twice or even three times through enlarging the width of the gears b. After improving the production technology of the motor, it is possible to increase its working pressure up to the value of 10-12 MPa. As a result, it is possible to build low-speed motors of very high torques and relatively small size, which can be used in mining machines, building machines, cranes, and ship machines.

Table 1.

Findings of the experimental research on the orbital motor featuring the double cycloidal gearing

Reference number	Parameter	Symbol of the parameter	Value of the parameter resulting from the research
1.	Specific inlet flow rate	q_s	9000cm^3/rev
2.	Working pressure (pressure difference at the inlet and outlet of the hydraulic motor $\Delta p_s = p_{we} - p_{wy}$)	Δp_s	5.9 MPa
3.	Torque on the shaft of the motor	M_s	6624 Nm
4.	Rotational speed range of the motor operation	n_{min}-n_{max}	1-40 rev/min
5.	Viscosity range of the working fluid	v_{min}-v_{max}	30-100 cSt

One of the directions in the development of fluid power machines is making the so called multifunctional fluid power machines. This name stands for machines of a complex structure, combining a few basic machines, realizing various functions, eg. the function of a pump, a motor, transmission, a speed changer, a coupling, a brake, or enabling operation at various stages, i.e. in motion, at the stable operation, and at braking.

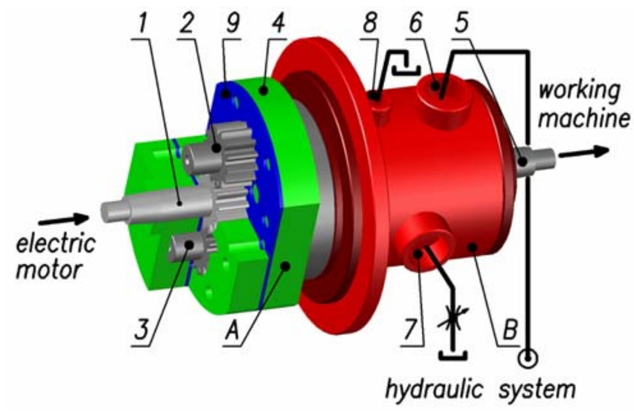

Figure 8. Design and operation principle of the multifunctional fluid power machine:A – gear unit, B – rotating distributor, 1,2,3 – gears; 4 – body of the machine, 5 – neck of the distributor's shaft, 6,7,8 – connection ports, 9 – compensation pad

According to the presented assumptions, a design of the machine has been developed, which is shown in figure 8. The machine consists of two basic machines, i.e. gear unit (A) and rotating distributor (B). There are three gears working in the gear unit: the solar gear (1) and two satellite gears (2). The solar gear shaft (1) is at the same time the input shaft of the machine. One part of the machine's body (4) is connected to the distribution shaft (5) of the rotating distributor (B). The distribution shaft neck (5) is at the same time the output shaft of the machine. The working fluid is delivered and received from the gear unit through the ports (6), (7), whereas port (8) is used for discharging the leakages.

As shown in figure 8, the machine is included in the driving system between the driving motor and the working machine, as well as simultaneously collaborates with the hydraulic system. Thus, it can perform three basic functions: of a pump, motor, speed changer or of a coupling.

If the working machine gets disconnected and the output shaft (5) gets blocked, the motor drives the multifunctional fluid power machine which at the time is functioning as a hydraulic pump. The input shaft with the solar gear (1) drive gears (2) and (3). The system receives the working fluid through port (6) and discharges it through port (7), collaborating with the fluid power system.

If the driving motor gets disconnected and the input shaft (1) gets blocked, the fluid power machine can work as a hydraulic motor. From the hydraulic system the working fluid is delivered to the gear unit, which moves gears (2) and (3) rolling them on the standing still central gear (1). Together with the gears, the body of the unit (4) and the output shaft connected to it (5) rotate, and the shaft drives the working machine.

If the multifunctional fluid power machine is connected both to the driving motor and to the working machine, then it works as a speed changer, or as a coupling. In that case, the flow of the working fluid goes through port (6) to the three-gear unit (1), (2), (3), and then is discharged through port (7) to the hydraulic system. On the discharge channel, the stream of working fluid is so throttled that the rotation of the gears is gradually blocked with the option of their total stoppage. After stopping the gears it is possible to pass the torque from the driving motor to the gears, and from the gears to the body (4) and to the output shaft connected to it (5). The level of stoppage depends on the flow rate of the working fluid at the outlet of the gear unit, that is the flow rate through the port (7) and the throttle valve located in the hydraulic system. Together with increasing of the valve setting, the flow rate increases, and at the same time the difference between the rotational speed of the driving motor and the speed of the working machine. It is possible then to achieve a smooth change of the rotational speed of the output shaft of the multifunctional fluid power machine in a range N_{min}-N_{max}=0-N_{motor}, namely performing as a speed changer. At the total blockage of the outlet flow, the multifunctional fluid power machine functions as a coupling.

The multifunctional fluid power machine has been made by PZL 'Hydral' Company, Wroclaw, and has been tested in the process in which the machine was functioning as a pump, a motor, and a speed changer. At the pumping work (driving work), the machine achieved the displacement (the inlet flow rate) q=60cm3/rev and the working pressure p=12MPa. While working as a speed changer it enabled the changes of the output shaft speed within the range of h=0÷1500 rev/min.

SUMMARY

Fluid power machines are commonly used in the design of the hydraulic driving and systems of machines and appliances. Development prospects of the machines are still open.

The research carried out by the FPRG from the Technical University of Wroclaw have shown that the development results from fulfilling of the following tasks:

- optimization of the involute and cycloidal tooth systems,
- revolutionary motion,
- optimization of shape and size of the channels and inner clearances in the machine,
- application of the novel materials and production methods.

Introducing those tasks into the designing practice is computer enhanced thanks to the use of such programs as the 'Fluid Power Involute Gears' and the 'Fluid Power Cycloidal Gears' developed by the FPRG from the Technical University of Wroclaw. Conducting the development process according to the specified assumptions, the FPRG have created new types of fluid power machines:

- gear pump featuring the optimized internal gearing,
- gerotor pump with the axial clearance compensation,
- orbital motor with the double internal cycloidal gearing,
- multifunctional fluid power gear machine.

The machine are of a simple and compact design. In relation to their size and weight, they achieve high displacements (inlet flow rates) coming up to a few thousand cm3/rev. Together with the improvement of their inner channel and inner clearance systems, it is possible to achieve increasingly higher working pressure amounting to 20MPa as well as volumetric efficiency η=75% and the total efficiency η=80%. The machines operate with the lower displacement pulsation (inlet flow rate) as well as with the lower noise level than the one of the piston machines.

It seems that the position of the fluid power gear machines among other fluid power types and units is not threatened. Achievement of really high technical parameters by the machines, as well as appearance of novel and unconventional design solutions in this area seems to be only a question of time.

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ПОСТРОЕНИЕ ДИНАМИЧЕСКИХ МОДЕЛЕЙ ОБРАБОТКИ ЗАГОТОВОК НА ОЛПЕРАЦИЯХ ЧИСТОВОГО ШЛИФОВАНИЯ

У статті розглянуті питання побудови динамічних моделей обробки заготовок на операціях чистового шліфування.

In the article the questions of construction of dynamic models of treatment of semi's are considered on operations of the clean polishing.

При построении модели процесса шлифования целесообразно использование системного подхода и модульного принципа, что позволяет, как рассматривать отдельные явления, происходящие в процессе обработки заготовки, так и их взаимодействие. Использование модулей дает возможность заменять отдельные блоки без существенных изменений остальной системы по мере накопления и уточнения информации о происходящих частных процессах и явлениях [1, 2, 3].

Множественность одновременных или практически одновременных элементарных процессов взаимодействия абразивных зерен с заготовкой при существенных различиях их параметров делает нецелесообразным попытки детализированного описания индивидуальных параметров и требует построения обобщенных групповых характеристик посредством теории стохастических процессов. Средние устойчивые значения считаются детерминированной компонентой, а в качестве случайной – рассматриваются отклонения от детерминированной составляющей. Основные факторы, которые учитываются при построении описания, представляются в форме