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Simulation and experimental research of hydraulic losses of an energy converting flow machine with rotating piston

Krzysztof Nozdrzykowski[1]✉, Adam Komorowski[2]

Maritime University of Szczecin, Institute of Basic Technical Sciences
[1] 2-4 Willowa St., 71-650 Szczecin, Poland, e-mail: k.nozdrzykowski@am.szczecin.pl
[2] 1-2 Wały Chrobrego St., 70-500 Szczecin, Poland, e-mail: a.komorowski@am.szczecin.pl
✉ corresponding author

Key words: energy-converting machine, rotating pistons, simulation of hydraulic losses, volumetric efficiency, comparative analysis, correlation and integral calculi

Abstract

This article describes the construction and principles of operation of a prototype energy-converting machine – a flow machine with pistons rotating in toroidal spaces. A comparative analysis of the results of simulation and experimental tests of fluid flow resistance in the presented prototype machine has been performed with the use of correlation and integral calculi. The article also presents a comparative analysis of the volumetric efficiency of the examined pump and of a Vogelsang pump with similar design and size. The present state of the research makes it possible draw conclusions about the high exploitation value of the proposed pump construction. For this displacement pump, performance increases along with rotational speed. However, it is different from other pumps by virtue of is simple and compact construction as well as greater capacity in relation to its size.

Introduction

A hydraulic engine is an energy converting flow machine that converts the pressure energy of a fluid stream into mechanical energy (e.g. the rotary motion of an output shaft). The principle of operation of the hydraulic engine is the reverse of that of the positive displacement pump. In theory, therefore, each positive displacement pump can operate as a hydraulic engine and vice versa. These two machines typically have identical, or only slightly different, designs. A specific example of such an energy-converting machine is the positive displacement pump with rotary pistons. Research associates at the Institute of Basic Technical Sciences of the Maritime University of Szczecin have designed and built a prototype of a new-generation rotary pump with the working designation of M–05. The solution is described in detail in the patent application (Kuźniewski, 1982). One of the stages of verification of its operation consisted in the performance of simulation and experimental tests of fluid flow resistance in the pump's working space. The test results are presented in this article. The tests have been performed with the aim of optimizing the shape of the pump's working space to reduce flow loss.

Design of the energy-converting machine

The novelty of the technical solution (Figure 1) lies in the body of the pump (1), consisting of inner toroidal working spaces (5) in which rotary pistons (2), in the shape of segments of a rectangular cross-section toroid, are fitted. The toroidal working spaces merge, while the moving pistons do not touch each other. The working fluid is supplied and discharged through the openings (3) and (6). The pistons, connected to rollers (7) fitted in bearings in the pump's body, rotate in the directions indicated by arrows (8), while their motion is synchronized by means of a transmission gear. The manner in which the pistons are connected to the rollers or the

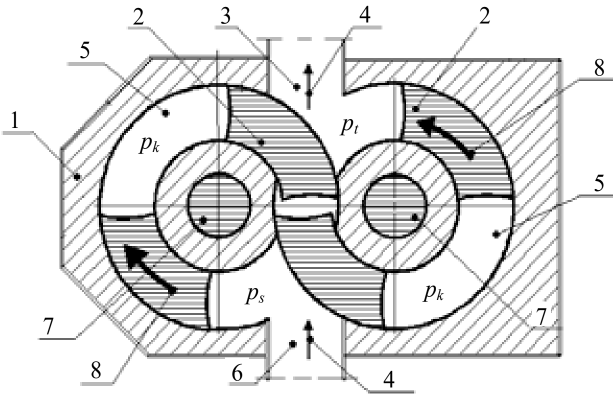

Figure 1. Positive displacement pump with rotary pistons

transmission gear is not shown on the illustration. In theory, such a machine should be characterized by an even mass of fluid flow and a high performance in relation to its size.

The machine can operate as either a high-performance pump or a hydraulic engine. The prototype was altered and improved during testing. As mentioned above, one of the testing stages was a comparative analysis between experimental and modeling results of the working fluid flow in the machine's working spaces.

Simulation and experimental tests

Theoretical values of the fluid flow resistance in various locations of the pump's working space were obtained through computation of the fluid flow resistance for a toroidal segment with rectangular cross-section (which is a component of the prototype) and for a rectilinear segment equal cross-section. The computation was carried out using the TORUS software, which calculates flow resistance depending on the geometric parameters of the toroidal and rectilinear segments. The geometric configurations for both segments are presented in Figure 2.

Figure 3 presents the algorithm used for the flow loss computation for both segment shapes.

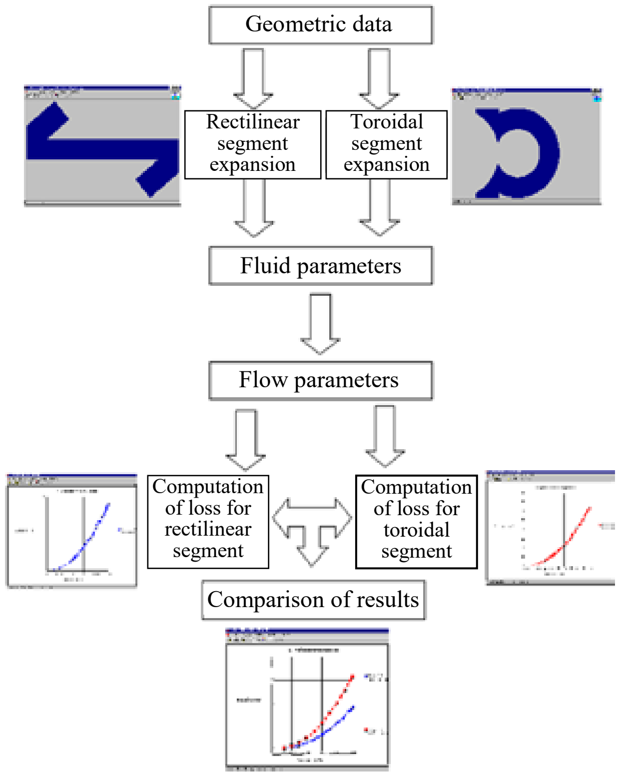

Figure 3. Algorithm for calculations of flow loss for rectilinear and toroidal segments

The computation makes use of relationships for flow loss coefficients that also depend on the geometric parameters of the segment (Bukowski, 1975; Crowe & Roberson, 1975; Puzyrewski, 1987). The coefficients have been approximated by means of functions accounting for changes of the geometric parameters, within the scope possible for the selected configurations.

The function of flow loss can be described as:

$$h_{Total} =$$
$$= \sum_{i=1}^{n} h_{SM}(\zeta_i, x_i, \varepsilon_i, v_i) + \sum_{j=1}^{m} h_{STR}(\zeta_j, y_j, \varepsilon_j, v) \quad (1)$$

where:

h_{SM} – losses on local obstacles; functions of the following values:

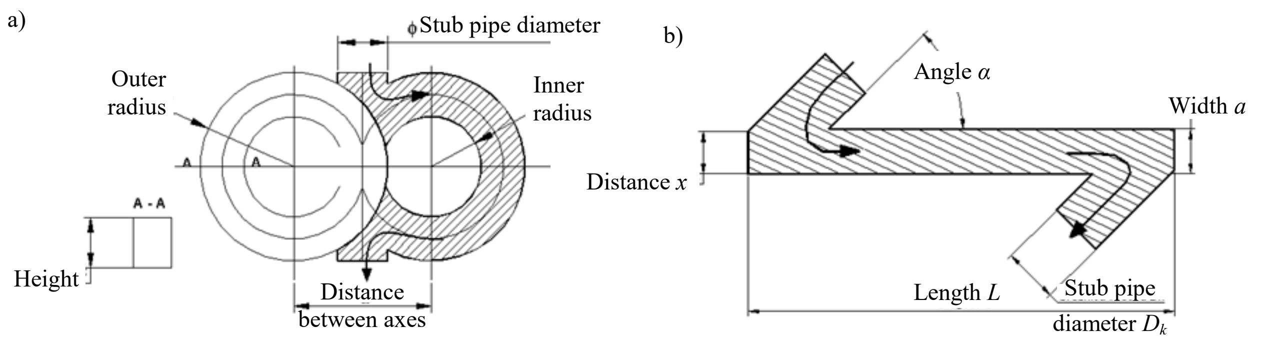

Figure 2. Geometric configuration: a) toroidal segment, b) rectilinear segment

ζ_i – flow resistance coefficient for the obstacle;
x_i – geometric parameters of the obstacle;
ε_i – fluid physical parameters;
v_i – flow velocity for the obstacle;
h_{STR} – losses caused by friction on segments of the same cross-section parameters; functions of the following values:
ζ_j – flow resistance coefficient;
y_j – geometric parameters of the segment cross-section and length;
ε_j – fluid physical parameters;
v – mean flow velocity throughout the total segment length for the same cross-section parameters.

Based on the above function, a numerical algorithm was developed and applied in TORUS. The algorithm accommodates and verifies the type of flow (laminar or turbulent) and, depending on the type of flow, uses appropriate functional relationships. To enable proper calculation of the aggregate local losses in the cases under review, the obstacles were classified and the corresponding loss coefficients, ζ, and flow velocities were calculated for specific types of obstacles. For particular cases, flow velocities were calculated using the mean initial velocity (based on the assumed performance (Puzyrewski & Sawicki, 1997)) and by applying the continuity equation where the cross section changed.

The assessment of the actual flow resistance was performed using a dedicated test bed, adjusted for toroidal (1) (Figure 4) or rectilinear (1) (Figure 5) flow tests. The test bed was equipped with a flow meter (2), pressure sensors (3) and a feed pump (5) that forced the flow of the working fluid (contained in a tank) (4) in the closed-loop system. The transverse dimensions and the geometric parameters of the inlet and outlet were identical for the rectilinear and toroidal segments. The length of the rectilinear segment was equal to the length of the toroidal section's perimeter.

The calculations and measurements were carried out for performance values ranging between 0.001 and 0.01 m^3/s, taking ten points at regular intervals. The setup was characterized by the following parameters:
- critical Reynolds number = 2000,
- coarseness of the segment's surface = 0.03 mm;

geometric parameters of the pump:
- outer radius = 75 mm,
- inner radius = 40 mm,
- distance between axes = 110 mm,

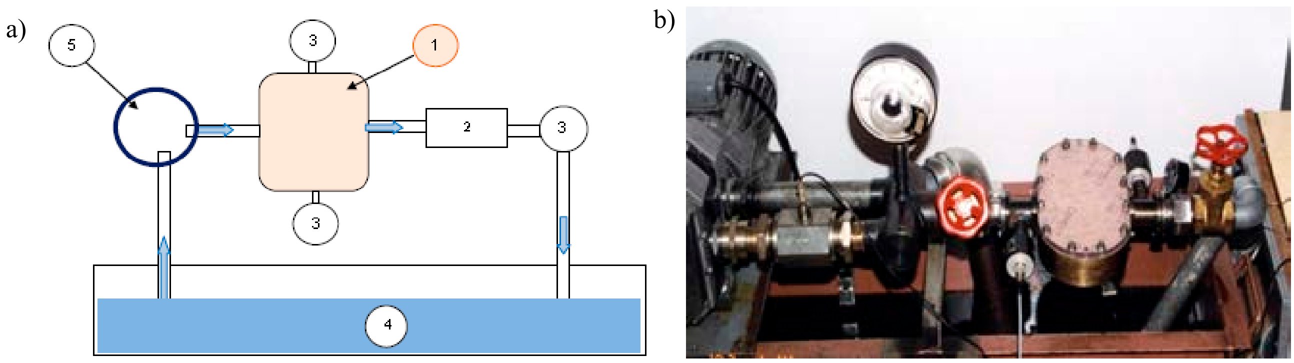

Figure 4. Test bed diagram a), test bed adjusted for toroidal flow resistance tests b)

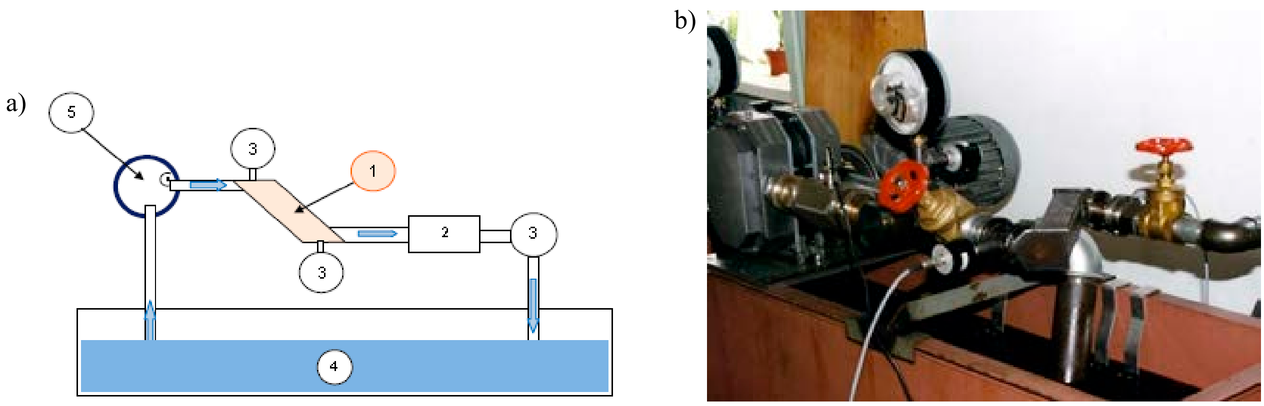

Figure 5. Test bed diagram a), test bed adjusted for rectilinear flow resistance tests b)

– height = 50 mm,
– stub pipe diameter = 42 mm;

geometric parameters of the working fluid:

– density = 961 kg/m^3,
– dynamic viscosity = 0.001 Ns/m^2.

The results of calculations and measurements are presented in Table 1 and in Figures 6 and 7.

Table 1. Values of pressure losses determined theoretically and experimentally

Perfor-mance, m^3/s	Pressure losses determined theoretically, kPa		Pressure losses determined experimentally, kPa	
	Rectilinear section	Toroidal section	Rectilinear section	Toroidal section
0.001	0.1798	0.3167	0.189	0.339
0.002	0.7192	1.267	0.75516	1.35569
0.003	1.618	2.851	1.6989	3.05057
0.004	2.877	5.068	3.02085	5.42276
0.005	4.495	7.919	4.71975	8.47333
0.006	6.473	11.40	6.79665	12.198
0.007	8.810	15.52	9.2505	16.6064
0.008	11.50	20.27	12.075	21.6889
0.009	14.56	25.66	15.288	27.4562
0.010	17.98	31.67	18.879	33.8869

Comparative analysis of results

A comparative analysis of the results obtained from the theoretical calculations and experimental tests was performed through correlation calculus. The Pearson correlation coefficient was used to determine the degree of correlation between the results of calculations and tests, yielding a value of 0.9998 when applied to the flow loss for toroidal and rectilinear segments. This result confirms both the correctness of the mathematical system adopted and a positive verification of the developed computation program, as well as the fact that the program is apt to be used for optimizing the pump's design details.

The degree of differences between the study results was determined based on the following relationship:

$$D = \sqrt{\frac{1}{(x_2 - x_1)} \int_{x_1}^{x_2} \left[f_2(x) - f_1(x) \right]^2 \mathrm{d}\,x} \qquad (2)$$

where: $f_2(x)$, $f_1(x)$ – functions to be compared, x_2, x_1 – integration limits. This expression returns results in the form of polynomials and enables the evaluation of the percentage difference between the results

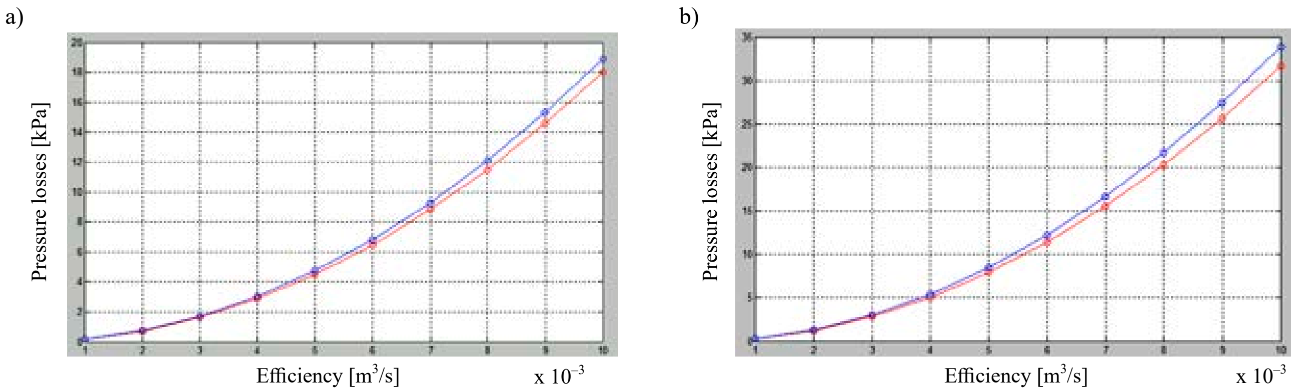

Figure 6. Results of calculations (red) and experiments (blue) of flow losses for a toroidal segment a), rectilinear segment b)

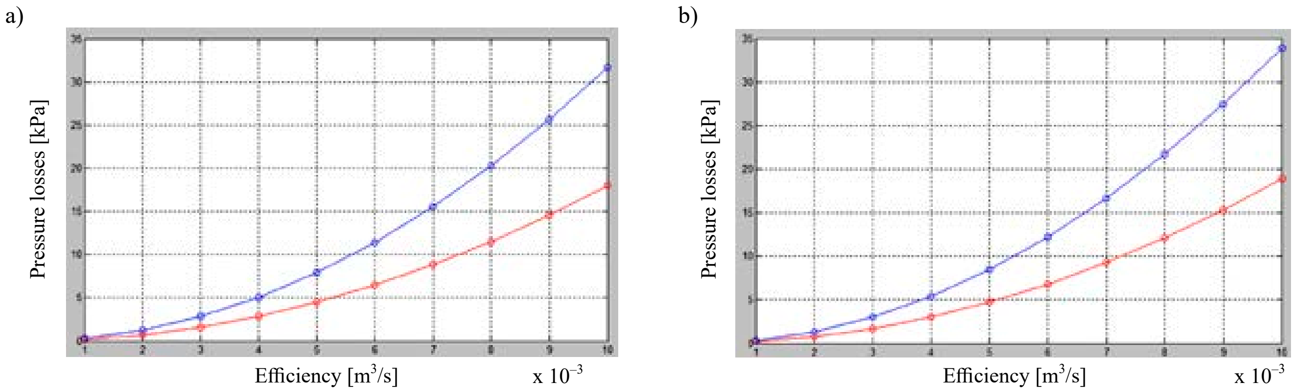

Figure 7. Flow losses for a toroidal segment (blue), and a rectilinear segment (red) obtained from theoretical calculations a) and experimental tests b)

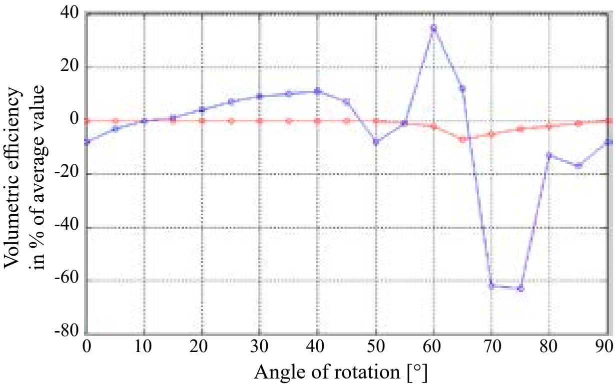

Figure 8. Theoretical performance of the M-05 pump (red) and the Vogelsang rotary pump (blue)

obtained from simulations and experiments, and for the toroidal and rectilinear segments.

The computations show a 5–7% difference and confirm a high degree of correlation between the results of theoretical calculations and experimental tests (Figure 6). They also show a 42.56% increase of the flow pressure loss determined through theoretical computation on the toroidal segment compared to the rectilinear segment, and a 43.96% increase of the same quantity determined through experimental measurements (Figure 7). This means that further optimization of the pump's working space shape, especially its inlet and outlet sections, is necessary; however, it should be stressed here that the aggregate pressure loss in the pump's working space has a small value relative to the pump's performance and operation parameters (Drozdowski & Komorowski, 2000; Stępniewski, 2005). The pump is characterized by steady performance, confirmed by way of a comparative analysis of the prototype and of a Vogelsang pump with similar design and size (Figure 8) (Górski, 2010).

Conclusions

High correlation of results of the comparative analysis confirms the correctness of the mathematical apparatus used and is a positive verification of the developed computation program.

In percentage values, losses at the pump's inlet and outlet have the biggest share of the aggregate flow loss. Therefore, the pump's design requires further optimization, especially aimed at improving the shape of its inlet and outlet sections.

The results show a considerable flow loss in the toroidal segment compared to that in the rectilinear segment (42%). However, the loss value relative to the pump's performance is small. In relation to the assumed range of study parameters, it does not exceed a value of 3%. It should be mentioned here that when the pump is in operation, the working fluid is in contact with the pump's body and pistons only on the toroidal segment. Taking into consideration the fact that the distance from the upstream to the downstream section of the pump is short, these losses have little effect on the pump's overall performance.

On the basis of the current state of the tests, a conclusion may be drawn that the proposed pump design solution has high operational values. It is a positive displacement pump, so its performance increases proportionally to the rotation velocity; however, unlike other pumps, it has a simple and compact design while its performance relative to the size is higher than that of other pumps.

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