SCIENTIFIC PROBLEMS
OF MACHINES OPERATION
AND MAINTENANCE
2 (154) 2008

MICHAŁ CZYŃSKI*

Energy efficiency of hydrostatic transmission. Comparing results of laboratory and simulation tests

Key words

Hydrostatic transmission, energy losses, energy efficiency, laboratory tests, simulation tests.

Słowa kluczowe

Przekładnia hydrostatyczna, straty energetyczne, sprawność energetyczna, badania laboratoryjne, badania symulacyjne.

Summary

The paper presents the results of laboratory and simulation tests of energy efficiency of hydrostatic transmission with volumetric control. The tests were carried out at the test bed built on the basis of PTO type displacement machines and allowed the verification of mathematical models of the loss and energy efficiency of elements and hydrostatic transmission proposed by Z. Paszota [2].

1. Introduction

The growing costs of energy and hydrostatic driving powers force design engineers and producers of elements and systems to limit the amount of energy loss. The work concentrates mainly on improving machines, valves, and control

* Politechnika Szczecińska, Wydział Techniki Morskiej, Al. Piastów 41, 71-065 Szczecin, tel. (91) 449-48-10, e-mail: michal.czynski@ps.pl

methods [6, 7, 8]. Designing energy-saving systems should not be limited to the selection of elements characterised by high efficiency at their nominal parameters of work. The designing process should not lack analyses optimising the selection in regard to the system's work conditions (pressure, the delivery of the pump, oil viscosity) especially the ones that occur the most often or last the longest. Therefore, it is necessary to carry out simulation calculations of energy behaviour in a system. Applications that make it possible must be based on precise and laboratory tested mathematical loss models revealing all factors influencing the amount of loss. The models should be built in such a way as to make calculations with any work parameters as precisely as possible.

The majority of known energy loss models in hydraulic elements was presented in the work [1]. The models describe mainly physical complexity of the phenomenon without referring to the work conditions of the system. The designers of the models also examine mechanical and pressure losses together. Building upon this base, these characteristics produced a precise model describing the energy behaviour of the system is actually impossible. The loss models presented by Z. Paszota [2] are free from these faults. The efficiency model of the hydraulic system built on their basis allows the simulation calculations of efficiency and takes into account factors such as the following:

- The load and speed coefficient of hydraulic motor,
- The amount of energy loss in the system's elements,
- The structure of velocity control system of hydraulic motor,
- The characteristics of engine propelling the pump,
- The characteristics of control elements, and
- The kinematic viscosity of working medium.

The concept of applying dimensionless k_i energy loss coefficients and introducing $\overline{\omega}_M$ and $\overline{M}_M$ reduced variables determining the speed and load of the hydraulic motor made the generalisation of system's efficiency model possible. The k_i loss coefficients provide information about the amount and ratio of particular kinds of losses in the elements in reference to the nominal work parameters of the system (mainly nominal pressure).

Selected results of laboratory tests on losses and energy efficiency in hydrostatic transmission with a variable displacement pump and working in open circulation are presented below. The tests allowed us to check if the calculations of system's energy efficiency according to the model proposed by Z. Paszota are precise and to verify the assumed hypotheses. Because of the accepted scope of work including tests on transmission with PTOZ2-25 pump and PTO2-16 motor, the obtained results are quantitative regarding this specific system's solution. However, research aspects of the tests enable us to treat the obtained results as qualitative relations.

2. Energy efficiency model of hydrostatic transmission with volumetric control

Efficiency η is the quality criterion of an element or a system enabling us to determine their ability to convert or transport energy. The efficiency is a dimensionless indicator changing within the range (0-1) and reflecting energy losses in the system's elements (volumetric, pressure and mechanical losses in displacement machines and pressure losses in pipes).

Total efficiency of the transmission is defined as a product of the efficiency of energy elements the transmission is built of the following:

$$\eta = \eta_P \cdot \eta_M \cdot \eta_C = \eta_{Pv} \cdot \eta_{Pp} \cdot \eta_{Pm} \cdot \eta_{Mv} \cdot \eta_{Mp} \cdot \eta_{Mm} \cdot \eta_C ,$$

where: η_P – total efficiency of the pump,
η_M – total efficiency of the hydraulic motor,
η_C – total efficiency of system's pipes,
η_{Pv} – volumetric efficiency of the pump,
η_{Pp} – pressure efficiency of the pump,
η_{Pm} – mechanical efficiency of the pump,
η_{Mv} – volumetric efficiency of the hydraulic motor,
η_{Mp} – pressure efficiency of the hydraulic motor,
η_{Mm} – mechanical efficiency of the hydraulic motor.

By using relations describing the energy efficiencies of hydraulic system's elements, the following statements describing the total efficiency of the transmission may be formulated [2] as follows:

a) The speed of an engine propelling the pump is constant (does not depend on the load) *($k_2 = 0$)*:

$$\eta = \frac{\left\{\overline{Q}_M - k_9\left[k_{7.1} + \left(1 + k_{7.2}\right)\overline{M}_M\right]\right\}\overline{M}_M}{k_{4.1} + \left[\left(1 + k_{4.2}\right)\overline{p}_{P2} + k_3 \cdot \overline{Q}_M^{\,2}\right]\left(\overline{Q}_M + k_1 \cdot \overline{p}_{P2}\right)},$$

b) The speed of an engine propelling the pump depends on the pump's load *($k_2 \neq 0$)*:

$$\eta = \frac{\left(X - k_1 \cdot \overline{p}_{P2}\right)Y}{\left\{k_{4.1} + \left[\left(1 + k_{4.2}\right)\overline{p}_{P2} + k_3 \cdot \overline{Q}_M^{\,2}\right]X\right\}\overline{Q}_M},$$

in which:

$$X = \frac{1}{2 \cdot k_2}\frac{1}{\overline{p}_{P2}} - \sqrt{\left(\frac{1}{2 \cdot k_2}\frac{1}{\overline{p}_{P2}}\right)^2 - \frac{k_1}{k_2} - \frac{1}{k_2}\frac{1}{\overline{p}_{P2}}\overline{Q}_M} ,$$

$$Y = \left\{\overline{Q}_M - k_9\left[k_{7.1} + \left(1 + k_{7.2}\right)\overline{M}_M\right]\right\}\overline{M}_M ,$$

$$\overline{p}_{P2} = k_{7.1} + (1 + k_{7.2})\overline{M}_M + (k_{5.1} + k_{5.2})\overline{Q}_M + k_8 \cdot \overline{Q}_M^{\,2},$$
$$\overline{Q}_M = \overline{\omega}_M + k_9\left[k_{7.1} + (1 + k_{7.2})\overline{M}_M\right],$$

where: k_1 – coefficient of relative volumetric losses in pump,
k_2 – coefficient of relative decrease in pump rotational speed,
k_3 – coefficient of relative pressure drop in internal pump ducts,
$k_{4.1}$ – coefficient of relative mechanical losses in pump,
$k_{4.2}$ – coefficient of relative increase of mechanical pump losses,
$k_{5.1}$ – coefficient of relative pressure drop in the pumping pipe,
$k_{5.2}$ – coefficient of relative pressure drop in the outflow pipe of the hydraulic motor,
$k_{7.1}$ – coefficient of relative mechanical losses in unloaded hydraulic motor,
$k_{7.2}$ – coefficient of relative increase of mechanical loss in hydraulic motor,
k_8 – coefficient of relative pressure drop in internal ducts in the hydraulic motor,
k_9 – coefficient of relative volumetric losses in the hydraulic motor,
$\overline{M}_M$ – hydraulic motor load coefficient,
$\overline{\omega}_M$ – hydraulic motor speed coefficient.

3. Test BED

The schematic diagram of the test bed is presented in Figure 1. Thanks to the application of a sensitive measuring apparatus at the test bed, it was possible to carry out precise tests of losses occurring in hydrostatic elements and systems. The applied system of putting the transmission under load made it possible to smoothly change the output torque of the hydraulic motor as well as to stabilise it on a given level irrespective of the motor rotational speed.

The tested hydrostatic transmission *I* was built based on machines that have similar construction, i.e. axial piston variable displacement pump PTOZ2-25 *(1)* and fixed displacement motor PTO2-16 *(2)*. The pressure was measured with four P3MB-type pressure transducers *(3,4,5)* placed in the inflow and outflow pipes of machines. There were T5 torque *(7)* and encoders *(8)* measuring rotational speed installed on the shafts of both units. The delivery of the pump was measured with PT3S flowmeter *(6)* placed between the pump and the motor.

The measuring system of the test bed was based on high quality elements produced mainly by the Hottinger Company. The basic elements of the measuring system were two multi-channel electronic measurement units Spider 8 and a computer with Catman 4.5 measuring software.

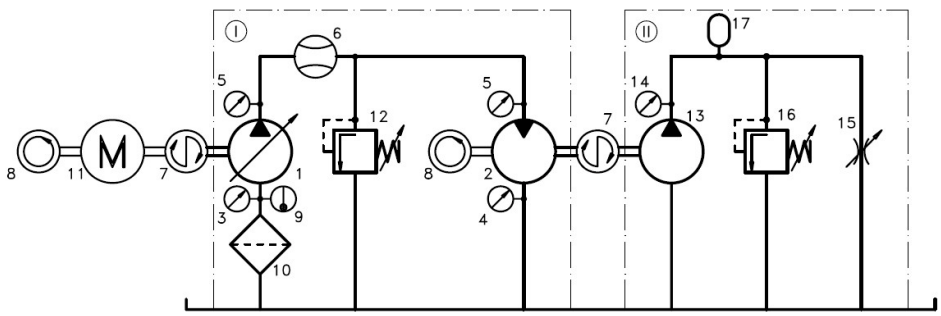

Fig. 1 The diagram of the test bed measuring the losses in hydrostatic transmission with a variable displacement pump: I. tested transmission, II. load system, 1. PTOZ2-25 pump, 2. PTO2-16 hydraulic motor, 3. P3MB/1bar pressure transducer, 4. P3MB/5bar pressure transducer, 5. P3MB/500bar pressure transducer , 6. PT3S piston meter, 7. T5/100Nm torque transducer, 8. MOL305 rotational speed transducer, 9. TP-371Pt100 temperature sensor, 10. oil filter, 11. Sg160M-4 11kW propelling engine, 12. pressure relief valve, 13. PTO2-40 pump, 14. pressure gauge, 15. throttle valve, 16. pressure relief valve, 17. hydraulic accumulator

Rys. 1. Schemat stanowiska umożliwiającego pomiar strat w przekładni hydrostatycznej z pompą o zmiennej wydajności: I. badana przekładnia, II. układ obciążający, 1. pompa PTOZ2-25, 2. silnik hydrauliczny PTO2-16, 3. przetwornik ciśnienia P3MB/1bar, 4. przetwornik ciśnienia P3MB/5bar, 5. przetworniki ciśnienia P3MB/500bar, 6. przepływomierz tłokowy PT3S, 7. przetworniki momentu obrotowego T5/100Nm, 8. przetworniki obrotowo-impulsowe typu MOL305, 9. czujnik temperatury TP-371Pt100, 10. filtr oleju, 11. silnik napędowy Sg160M-4 11kW, 12. zawór bezpieczeństwa, 13. pompa PTO2-40, 14. manometr sprężynowy, 15. zawór dławiący, 16. zawór przelewowy, 17. akumulator hydrauliczny

The methods of measuring volumetric, pressure, and mechanical losses in displacement machines were presented in the works [3, 4, 5].

4. Results of measuring energy losses in the elements of transmission

Selected results of tests on energy losses in the elements of the hydraulic system are presented below. All tests were performed with one kinematic oil viscosity of $v=35\ mm^2/s$ that was constant during the test.

The test results presented above allowed us to verify the assumed hypotheses while creating mathematical models of energy losses in displacement machines, i.e. in the variable displacement pump and in the fixed displacement motor.

While creating loss models, Z. Paszota assumed that the flow of the liquid in the channels and distributor is turbulent. The tests, however, showed that in the case of PTOZ2-25 pump (Figure 2a) and PTO2-16 motor (Figure 3a) the flow is turbulent and not fully developed. Nevertheless, it must be stated that the character of liquid's flow through the unit is influenced mainly by local resistance (disturbances of direction and stream section area), depending on the construction and the size of the machine. By building the model of pressure

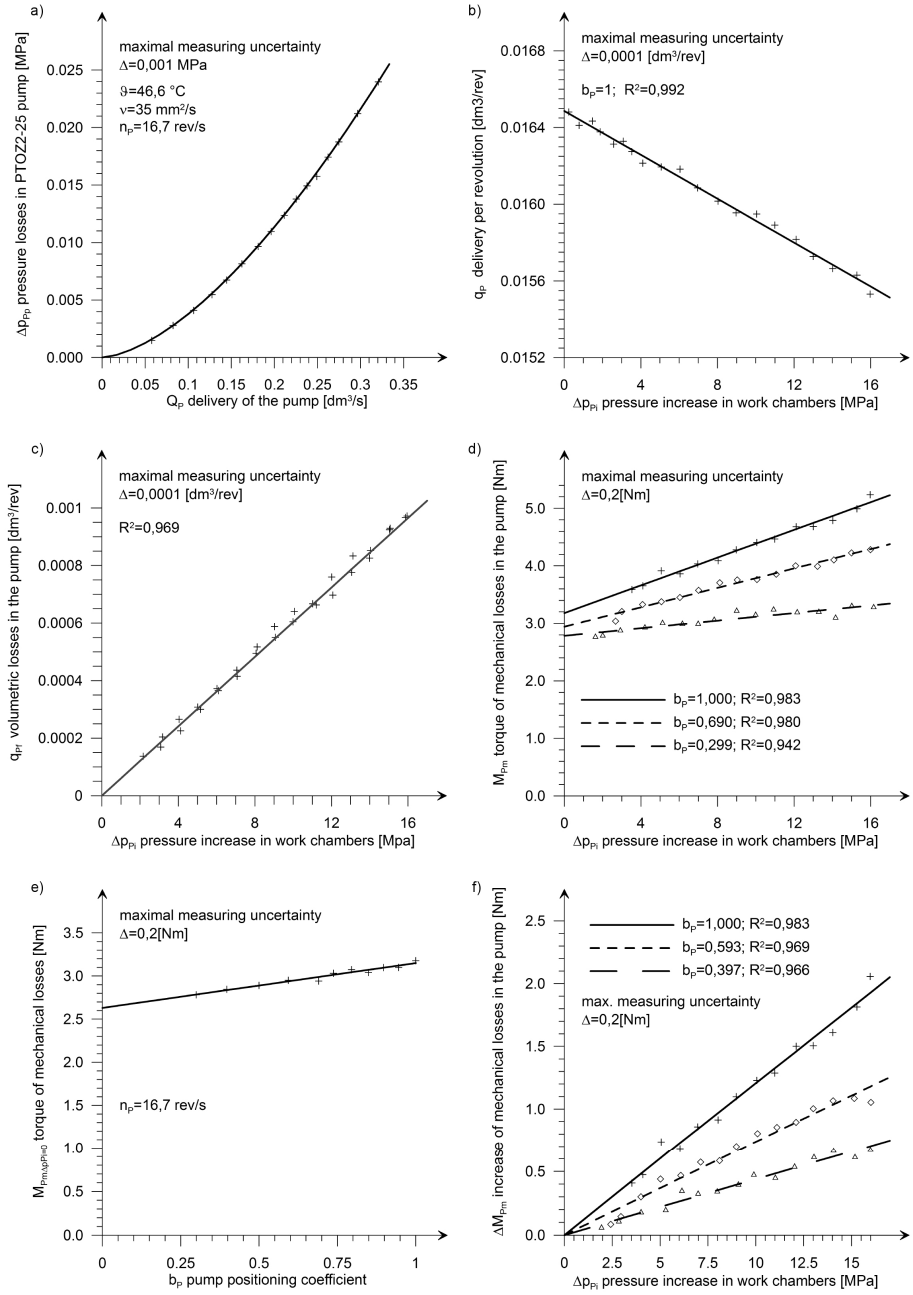

Fig. 2: Test results of PTOZ2-25 pump; a) pressure losses, b) theoretical working cubic capacity, c) volumetric losses per one shaft's revolution, d) mechanical losses, e) mechanical losses in unloaded pump, f) torque increase of mechanical losses

Rys. 2. Wyniki badań pompy PTOZ2-25; a) straty ciśnieniowe, b) teoretyczna objętość robocza, c) straty objętościowe na jeden obrót wału, d) straty mechaniczne, e) straty mechaniczne w nieobciążonej pompie, f) przyrost momentu strat mechanicznych

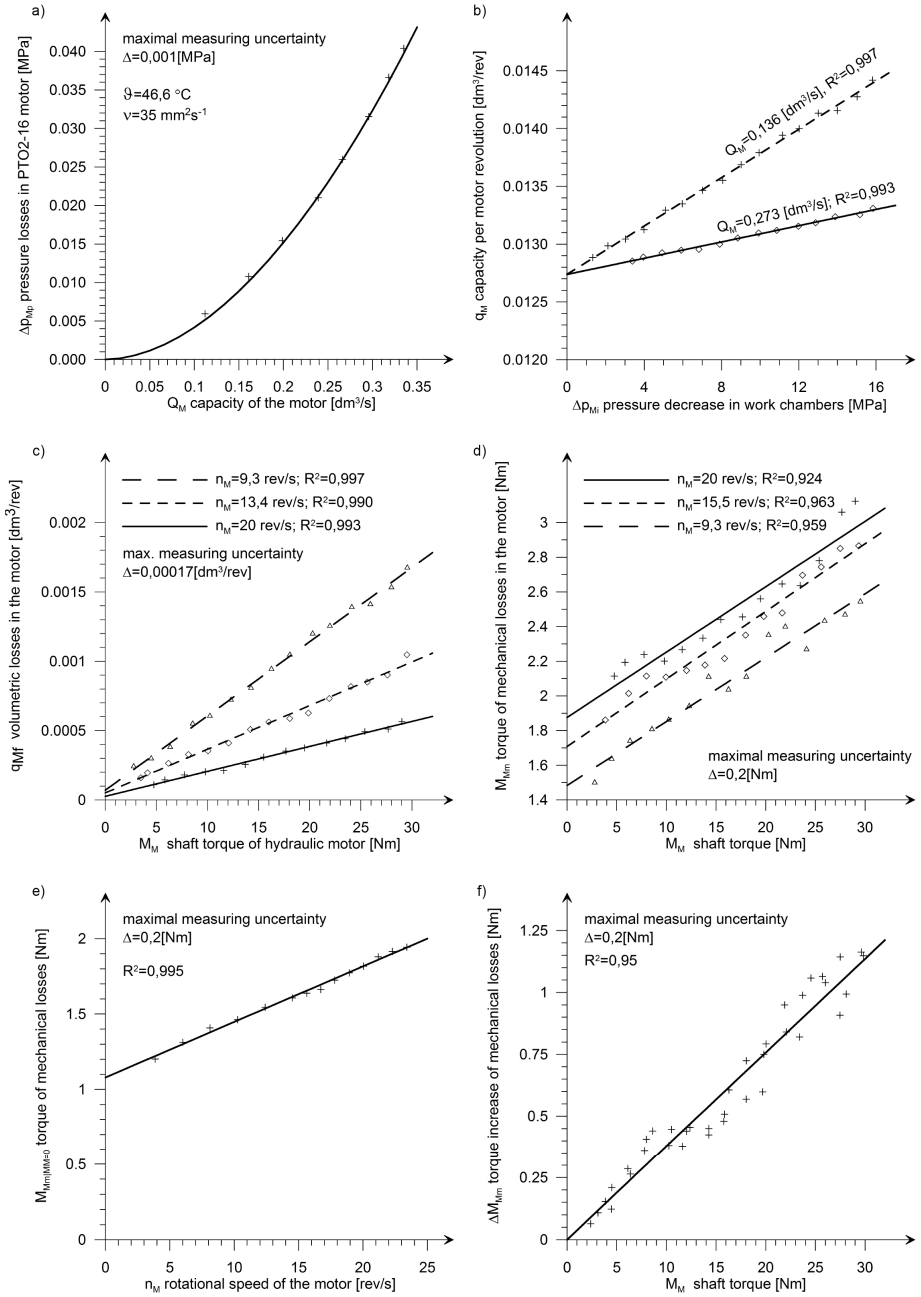

Fig. 3. Test results of PTO2-16 hydraulic motor; a) pressure losses, b) theoretical working cubic capacity, c) volumetric losses per one shaft's revolution, d) mechanical losses, e) mechanical losses in unloaded motor, f) torque increase of mechanical losses

Rys. 3. Wyniki badań silnika hydraulicznego PTO2-16; a) straty ciśnieniowe, b) Teoretyczna objętość robocza, c) straty objętościowe na jeden obrót wału, d) straty mechaniczne, e) straty mechaniczne w nieobciążonym silniku, f) przyrost momentu strat mechanicznych

losses [2], one may, to limited extent, ”adjust” it to a particular machine without the necessity of interfering in the structure of the formula. The limitations result from the fact that, in the case of streamline flow, the pressure losses also depend on the oil viscosity, which was not considered in the model.

The results of measuring volumetric losses in the pump (Figure 2c) were analysed and proved the assumption that the losses do not depend on q_{pgv} instantaneous geometric working volume but are proportional to q_{Pt} pump's theoretical working volume (pump's size) and load (Δp_{Pi} decrease of pressure in working chambers) to be right. In the case of hydraulic motors, it was assumed that volumetric losses do not depend on the rotational speed of the motor. The tests on PTO2-16 motor (Figure 3c), however, did not confirm this assumption.

Moreover, the tests allowed us to verify hypotheses for mechanical losses in displacement machines. Z. Paszota assumed that in unloaded displacement machines their size (setting and rotational speed) influenced the magnitude of loss torque. This assumption, however, was not confirmed by the tests, because they proved that both the pump's setting (Figures 2d, e) and rotational speed of the motor (Figures 3d, e) influenced the amount of mechanical losses in unloaded units. The tests confirmed the hypothesis that the torque increase of mechanical losses is proportional to the load and the instantaneous working volume of the units (Figures 2f and 3f).

The results of measuring pressure losses in the system's pipes (Figure 4a) are the basis for the statement that the flow is laminar (flow resistance is in direct proportion to flow rate). It proves that the hypothesis, assuming that character of the working factor's flow is right.

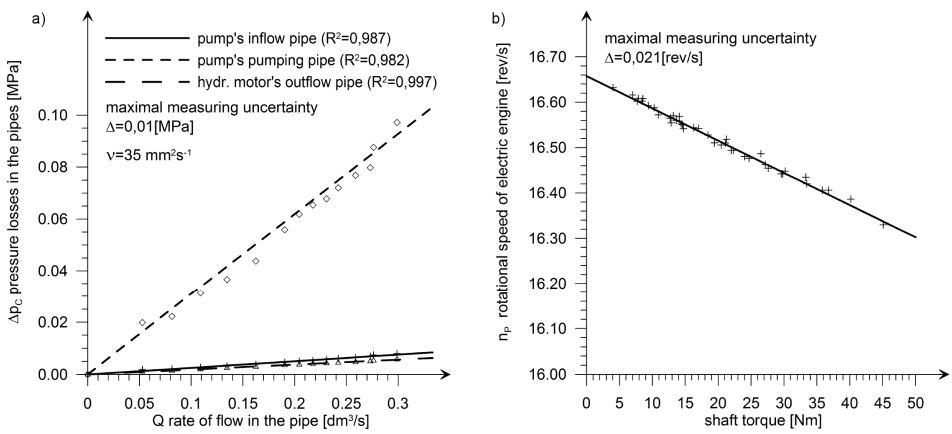

Fig. 4. a) The results of measuring pressure losses in the system's pipes, b) fragment of characteristic Sg160M-4 electric engine propelling the pump

Rys. 4. Wyniki badań: a) strat ciśnieniowych w przewodach, b) fragment charakterystyki silnika elektrycznego Sg160M-4 napędzającego pompę

The above mentioned divergences between the assumptions made by Z. Paszota and the results of tests on the system's elements forced modifications in the models of pressure and mechanical losses in the pump, volumetric, pressure and mechanical losses in the hydraulic motor, as well as in the efficiency model of hydrostatic transmission.

5. Proposition to modify mathematical efficiency model of hydrostatic transmission with volumetric control

After introducing the above mentioned changes in the models of losses, the expressions describing the energy efficiency of the transmission are the following:

a) The speed of an engine propelling the pump is constant (does not depend on the load) *($k_2 = 0$)*:

$$\eta = \frac{\left\{\overline{Q}_M - k_9\left[k_{7.1} - k_{7.1.1}\left(1-\overline{\omega}_M\right) + \left(1+k_{7.2}\right)\overline{M}_M\right]\right\}\overline{M}_M}{k_{4.1} - k_{4.1.1}\left[1-\left(\overline{Q}_M + k_1\cdot\overline{p}_{P2}\right)\right] + \left[\left(1+k_{4.2}\right)\overline{p}_{P2} + k_3\cdot\overline{Q}_M^{\,1,6}\right]\left(\overline{Q}_M + k_1\cdot\overline{p}_{P2}\right)},$$

b) The speed of an engine propelling the pump depends on the pump's load *($k_2 \neq 0$)*:

$$\eta = \frac{\left(X - k_1\cdot\overline{p}_{P2}\right)Y}{\left\{k_{4.1} - k_{4.1.1}\left(1-X\right) + \left[\left(1+k_{4.2}\right)\overline{p}_{P2} + k_3\cdot\overline{Q}_M^{\,1,6}\right]X\right\}\overline{Q}_M},$$

in which:

$$X = \frac{1}{2\cdot k_2}\frac{1}{\overline{p}_{P2}} - \sqrt{\left(\frac{1}{2\cdot k_2}\frac{1}{\overline{p}_{P2}}\right)^2 - \frac{k_1}{k_2} - \frac{1}{k_2}\frac{1}{\overline{p}_{P2}}\overline{Q}_M},$$

$$Y = \left\{\overline{Q}_M - k_9\left[k_{7.1} - k_{7.1.1}\left(1-\overline{\omega}_M\right) + \left(1+k_{7.2}\right)\overline{M}_M\right]\right\}\overline{M}_M,$$

$$\overline{p}_{P2} = k_{7.1} - k_{7.1.1}\left(1-\overline{\omega}_M\right) + \left(1+k_{7.2}\right)\overline{M}_M + \left(k_{5.1}+k_{5.2}\right)\overline{Q}_M + k_8\cdot\overline{Q}_M^{\,1,87},$$

$$\overline{Q}_M = \overline{\omega}_M + k_9\left[k_{7.1} - k_{7.1.1}\left(1-\overline{\omega}_M\right) + \left(1+k_{7.2}\right)\overline{M}_M\right],$$

where: $k_{4.1.1}$ – coefficient of decrease of mechanical losses in unloaded pump related to decreasing q_{Pgv} geometric working volume of the pump,

$k_{7.1.1}$ – coefficient of the decrease of mechanical losses in unloaded hydraulic motor related to its decreasing n_M rotational speed.

6. Simulation tests on energy efficiency of transmission

Based on laboratory tests, the values of k_i energy loss coefficients in the system's elements were determined. The values are the following:

- For the pump: $k_1=0.056$, $k_3=0.001$, $k_{4.1}=0.076$, $k_{4.1.1}=0.013$, $k_{4.2}=0.046$,
- For the hydraulic motor: $k_{7.1}=0.058$, $k_{7.1.1}=0.024$, $k_{7.2}=0.038$, $k_8=0.001$, $k_9=0.044$,
- For the pipes: $k_{5.1}=0.0055$, $k_{5.2}=0.0004$, and
- For an engine propelling the pump: $k_2=0.02$.

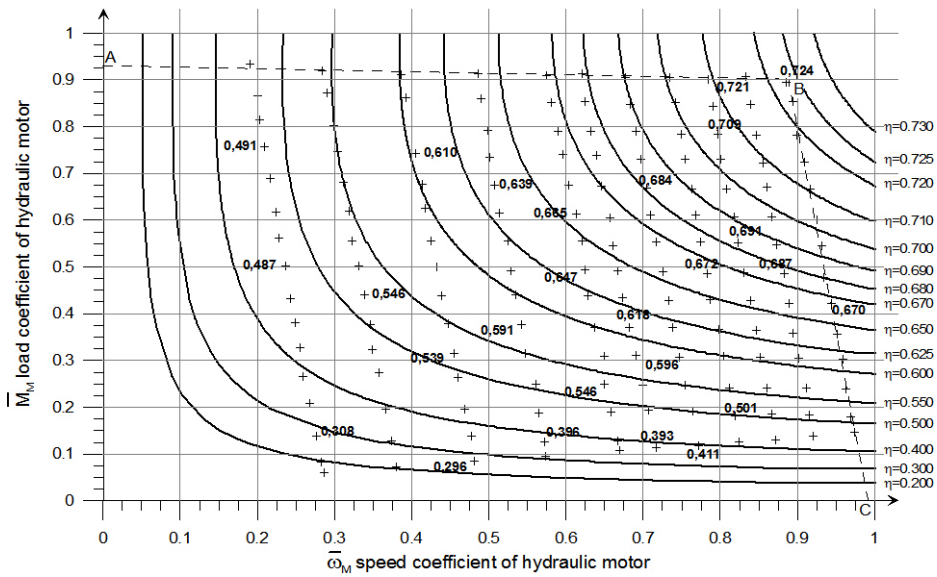

Fig. 5. Results of laboratory tests on total efficiency of transmission (points) and results of simulation calculations (lines) according to the model proposed by Z. Paszota [2]. Maximal relative error of the simulation is 18,5%

Rys. 5. Wyniki badań laboratoryjnych sprawności całkowitej przekładni (punkty) oraz obliczeń symulacyjnych (linie) wg modelu zaproponowanego przez Z. Paszotę [2]. Maksymalny względny błąd symulacji 18,5%

Figures 5 and 6 present the results of laboratory tests (points) and calculations of hydrostatic transmission's efficiency (curves). The diagrams make it possible to assess the influence that the load and speed of the hydraulic motor $\left(\overline{M}_M, \overline{\omega}_M\right)$ have on the energy efficiency of the transmission and to assess the accuracy of simulation calculations. Figure 6 shows how the accuracy of simulation calculations increases at low values of $\overline{M}_M$ load and $\overline{\omega}_M$ speed coefficients of hydraulic motor. It proves that the changes introduced in the model were right.

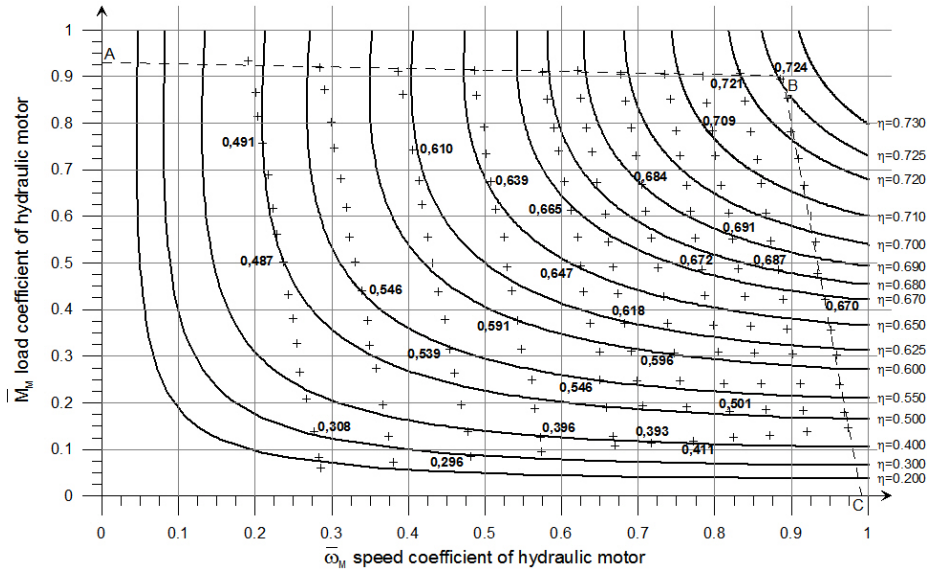

Fig. 6. Results of laboratory tests on total efficiency of transmission (points) and results of simulation calculations (lines) according to the model proposed by autor. Maximal relative error of the simulation is 3,5%

Rys. 6. Wyniki badań laboratoryjnych sprawności całkowitej przekładni (punkty) oraz obliczeń symulacyjnych (linie) wg modelu zaproponowanego przez autora. Maksymalny względny błąd symulacji 3,5%

7. Conclusion

The subject of the presented tests was a classic hydrostatic transmission with a variable displacement pump and fixed displacement motor working in open circulation. The system was built based on axial piston machines such as PTOZ2-25 pump and PTO2-16 hydraulic motor. The tests were concentrated on capturing the influence of the speed and load of hydraulic motor on the amount of energy loss. The obtained results of laboratory tests made it possible to verify the assumed hypotheses and to modify the mathematical model proposed by Z. Paszota [2].

The tests showed that mathematical models proposed by Z. Paszota are useful for simulation determination of energy efficiency of PTO type displacement machines and transmissions that are built of the machines. It was proved that the proposed changes of the model significantly improved the accuracy of simulation calculations, especially at low values of $\overline{M}_M$ load and $\overline{\omega}_M$ speed coefficients of hydraulic motor.

There is a need to continue the work aiming at capturing the influence the changes that hydraulic oil viscosity has on the amount of energy loss. There is also a need to perform laboratory tests on hydrostatic transmissions, including

machines of different construction, and to develop the efficiency model in such a way to make possible the calculations of hydrostatic transmissions working in closed circulation.

References

[1] Balawender A.: Analiza energetyczna i metodyka badań silników hydraulicznych wolnoobrotowych. Wydawnictwo Politechniki Gdańskiej, Gdańsk 1988.
[2] Paszota Z.: Aspects énergétiques des transmissions hydrostatiques. Wydawnictwo Politechniki Gdańskiej, Gdańsk 2002.
[3] Czyński M.: Model sprawności energetycznej przekładni hydrostatycznej - laboratoryjne metody pomiaru strat. Materiały XIII ogólnopolskiej konferencji CYLINDER 2003. KOMAG, Gliwice 2003.
[4] Czyński M.: Badania laboratoryjne modelu sprawności energetycznej przekładni hydrostatycznej z pompą o zmiennej wydajności. Materiałach X jubileuszowego seminarium Napędy i Sterowanie 2004 towarzyszącego Targom Producentów, Kooperantów i Sprzedawców Zespołów Napędowych i Układów Sterowania. Politechnika Gdańska, Międzynarodowe Targi Gdańskie S.A., Gdańsk 2004.
[5] Czyński M.: Badania laboratoryjne strat ciśnieniowych w pompie PTOZ2-16 - weryfikacja matematycznego modelu strat. Materiały VII Konferencji „Okrętownictwo i Oceanotechnika - Transport Zintegrowany". Politechnika Szczecińska, Wydział Techniki Morskiej, Szczecin 2004.
[6] Osiecki L.: Wpływ przestrzeni martwej na straty energetyczne w pompach wielotłoczkowych. Hydraulika i Pneumatyka - 2007 nr 3.
[7] Manring N.D., Mehta V.S., Raab F.J., Graf K.J.: The shaft torque of a tandem axial-piston pump. ASME Journal of Dynamic Systems, Measurement, and Control - 2007 nr3.
[8] Śliwiński P.: Power balance in hydraulic satellite motors supplied with oil and HFA-E emulsion. W: 5th FPNI PhD Symposium Krakow 2008 an initiative of Fluid Power Net International: proceedings, Krakow, Poland, July 1-5, 2008

Manuscript received by Editorial Board, April 21, 2008

Sprawność energetyczna przekładni hydrostatycznej.
Porównanie badań laboratoryjnych i symulacyjnych

S t r e s z c z e n i e

W artykule zamieszczono wyniki badań laboratoryjnych i symulacyjnych sprawności energetycznej przekładni hydrostatycznej ze sterowaniem objętościowym. Badania zostały przeprowadzone na stanowisku badawczym zbudowanym w oparciu o maszyny wyporowe typu PTO i umożliwiły częściową weryfikację matematycznych modeli strat i sprawności energetycznej elementów i przekładni hydrostatycznej zaproponowanych przez Z. Paszotę [2].