DOI 10.12851/EESJ201412C06ART08

***Vadim B. Samorodov*,**
ScD (Doctor in Engineering), Professor;

***Sergey A. Shuba*,**
Senior lecturer;

***Oleg I. Derkach*,**
Head of the Laboratory;

***Vadim M. Shevtzov*,**
Lecturer;

***Nikolay A. Mittsel*,**
Postgraduate;
National Technical University "KhPI"

Experimental Appropriateness Verification of K. Gorodetsky's Mathematical Model for Losses Determinination in Hydrostatic Transmissions for Modern Hydrolic Machines

Key words: *transmission, hydrostatic transmission, hydraulic pump, hydraulic motor efficiency.*

Annotation: *There were experimentally obtained dependences of efficiency coefficients for two hydrostatic transmissions of various sizes and performance on the basis of axial-piston hydraulic machines under various loading conditions. The experimental data is compared with the calculated one obtained using the mathematical model of K. Gorodetsky. The object of study was the hydrostatic monoblock transmission manufactured in 1999 and the hydrostatic transmission made in 2012 with a spaced hydraulic pump and hydraulic motor.*

Introduction. Proposed in the 80s of the XXth century the Gorodetsky's method of calculating the volumetric and mechanical efficiency (1,2), based on the analysis of a large number of universal characteristics of predominantly axial-piston hydraulic machines of the time, is widely used in computational and theoretical studies nowadays. However, the improvement of structural materials, increase of manufacturing precision and the overall reliability of the axial-piston hydraulic machines obviously results in the efficiency increase by reducing both the mechanical and volume losses [bur]. An adequate mathematical model of hydrostatic transmission efficiency (HST) is in high demand and needed to solve circuit problems of two- and multi-range hydrostatic-mechanical transmissions (HSMT). The refined mathematical model of HST efficiency in the evaluation and prediction of the main technical and economic indicators of tractors equipped with double-flow HSMT is of particular relevance. Therein lies the relevance of the research conducted below.

Purpose and objective of the study. The purpose of research is to verify experimentally the accuracy of the applied mathematical model of HST on the basis of modern axial piston hydraulic machines (APHM) with a swash plate. The research problem lies in comparing the calculated and experimental efficiency as the main parameter characterizing the efficiency of HST, for HST with an adjustable hydraulic pump and non-adjustable hydraulic motor GTN Hydraulics 3K10 P090 in monoblock performance (qd = Qm = 33 cm3) and HST with spaced adjustable axial piston hydraulic pump NP112 5MHL/ D2BCDBY1 and non-adjustable axial piston hydraulic motor MP112 2 / D2B35Y1 made by JSC «HYDROSYLA» (qd = Qm = 112sm3).

The mathematical model and solution to the set problem. To solve this problem, there was carried out a series of experiments at the stand for HST testing at the Department of Automobile and Tractor Engineering of the National Technical University "Kharkiv Polytechnic Institute" (stand №1, monoblock GTN Hydraulics 3K10 P090), as well as at the stand for HST testing at "Kharkov Tractor Plant after S. Ordzhonikidze" (stand №2, adjustable axial piston hydraulic pump NP112 5MHL / D2BCDBY1i non-adjustable axial piston hydraulic motor MP112 2 / D2B35Y1 made by JSC «HYDROSYLA»). The developed test stands are readjusted and, in addition to research of HST, make it possible to simulate the operation of dual-flow HSMT with an "input" and "output" differential.

Figure 1 shows the kinematic scheme of stand №1, as well as the connection points of instrumentation; Fig. 2 shows the appearance of the stand. The presence of the reduction gear 10 is due to unification of shafts with strain gauges of the torque (3). Control of the hydraulic pump was carried out using a stepper motor drive. A detailed description of the control system is presented in work (4).

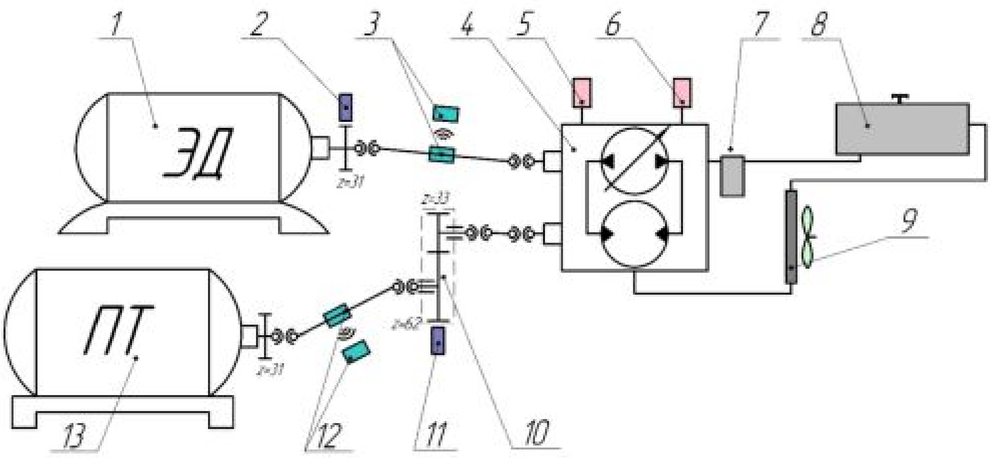

Fig. 1 – Kinematic diagram of stand №1for studying HST:
1 – asynchronous motor; 2, 11 – speed sensors; 3, 12 – strain gauges of the torque; 4 – hydrostatic transmission (HST); 5, 6 – excess pressure gauges; 7 – fine filter; 8 –tank; 9 – heat exchanger with a fan; 10 – parallel-shaft reduction gear unit; 13 electromagnetic powder load brake.

Fig. 2 – Appearance of stand №1 for studying HST

Figure 3 shows the kinematic scheme of stand №2 and connection points of instrumentation; Fig. 4 shows the appearance of the stand №2.

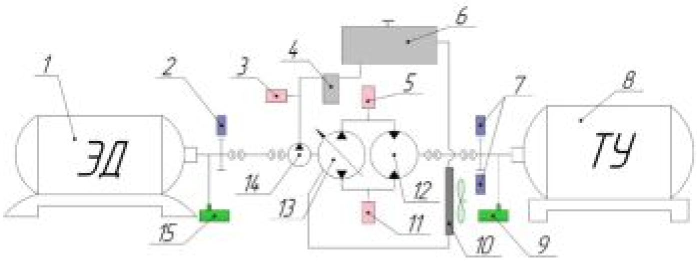

Fig. 3 – Kinematic diagram of stand №2 for studying HST:
1 – actuating motor; 2, 7 – speed sensors; 3 – sensor of recharge line overpressure; 4 – suction fine filter; 5, 11 – sensors of main power lines overpressure; 6 – tank; 8 – the braking system; 9, 15 – torque sensors; 10 – the heat exchanger with a fan; 12 – uncontrollable axial piston motor; 13 – controlled axial–piston pump; 14 – feed pump of gear type.


Fig. 4 – Appearance of stand №2 for studying HST

Fig. 5 shows a block diagram of HST stand №1.

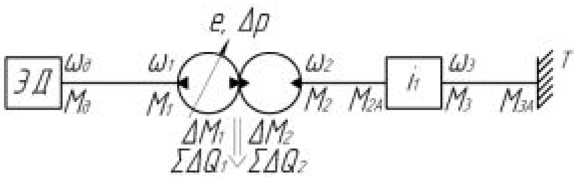

Fig. 5 – Block diagram of HST stand №1

According to the model of K. Gorodetsky volume losses in hydraulic machines 1 and 2 (internal and external leaks) $\Delta Q_{1,2}$ are determined from the expression

$$\Delta Q_{1,2} = (-\Theta)\cdot K_Y \frac{\Delta p}{\mu}\left(1 + C_Y\left|\omega_{1,2}\right|\right), \tag{1}$$

where K_Y and C_Y – coefficients of leakage; Δp – pressure fall, MPa; μ– dynamic viscosity of the fluid, Pa.

Loss of moments due to mechanical and hydraulic losses

$$\Delta M_{1,2} = (-\Theta)\cdot q\cdot\left[K_1\left|\omega_{1,2}\right|\left(1+K_2\bar{e}_{1,2}{}^2\right)+\frac{K_5\left(1+K_4\left|e_{1,2}\right|\right)}{\left(1+K_3\left|\omega_{1,2}\right|D_q\right)}\Delta p+\frac{K_8\left(1+K_7\left|e_{1,2}\right|\right)}{\left(1+K_6\left|\omega_{1,2}\right|D_q\right)}\right], \tag{2}$$

where D_q – characteristic size of the hydraulic machine $\left(D_q = \sqrt[3]{2\pi q}\right)$; $K_1, K_2, ..., K_8$ – coefficients of hydro–mechanical losses.

For calculations there were accepted the following factors

$K_Y = 0.0390\cdot 10^{-12}$ $м^3$; $C_Y = 1{,}44\cdot 10^{-2}$ sec; $K_1 = 2000\ Pa\cdot\text{sec}$;

$K_2 = 0{,}912$; $K_3 = 0{,}0955$ sec/ m; $K_4 = 0{,}653$; $K_8 = 0{,}825\cdot 10^6\ Pa.$

$K_5 = 0{,}0245$; $K_6 = 0{,}913$ sec/ m; $K_7 = 0{,}3375$;

The mechanical efficiency of the adjustable HM1 η_{1M} and non–adjustable HM2 η_{2M} can be represented as follows:

$$\eta_{1M} = 1 - \frac{\sum \Delta M_1}{M_1};\ \eta_{2M} = \frac{M_2}{M_2 + \sum \Delta M_2} \tag{3}$$

Here $\sum\limits_{i=1}^{h}\Delta M_1$, $\sum\limits_{i=1}^{h}\Delta M_2$ – the total mechanical losses of moments for HM1 and HM2 (h – number of pairs of hydro-mechanical friction, in which there occur fluid friction losses as well as dry Coulomb friction losses, and scrolling losses).

The expression for the volumetric efficiency of the HST has the form

$$\eta_0 = \eta_{10}\cdot\eta_{20} = \frac{Q_{1T} - \Delta Q_1}{Q_{1T}}\cdot\frac{Q_{2T} - \Delta Q_2}{Q_{2T}} = \frac{q\bar{e}\omega_1 - \Delta Q_1}{q\bar{e}\omega_1}\cdot\frac{q\omega_2}{q\omega_2 + \Delta Q_2}, \tag{4}$$

where Q_{1T}, Q_{1T}, q_1, q_2 – theoretical expenses and efficiency of adjustable HM1 and non-adjustable HM2.

The overall efficiency of the HST is calculated as the product of the volumetric and mechanical efficiency of the hydraulic pump and hydraulic motor

$$\eta_{ГОП} = \eta_{10}\cdot\eta_{20}\cdot\eta_{1M}\cdot\eta_{2M}\ . \tag{5}$$

The speed of the primary source rotation at stand №1 and №2 was $\omega_{Д} = \omega_{H} = 1500$ rev/min. In each case the tests were carried out for three different loading conditions: 1 – “low” loading; 2 – “average” loading; 3 – “high” loading. On the following charts of HST efficiency 1 – design curve, 2 – experimental curve. Figure 6 shows the results obtained on the stand №1, Figure 7 shows the results from the stand №2.

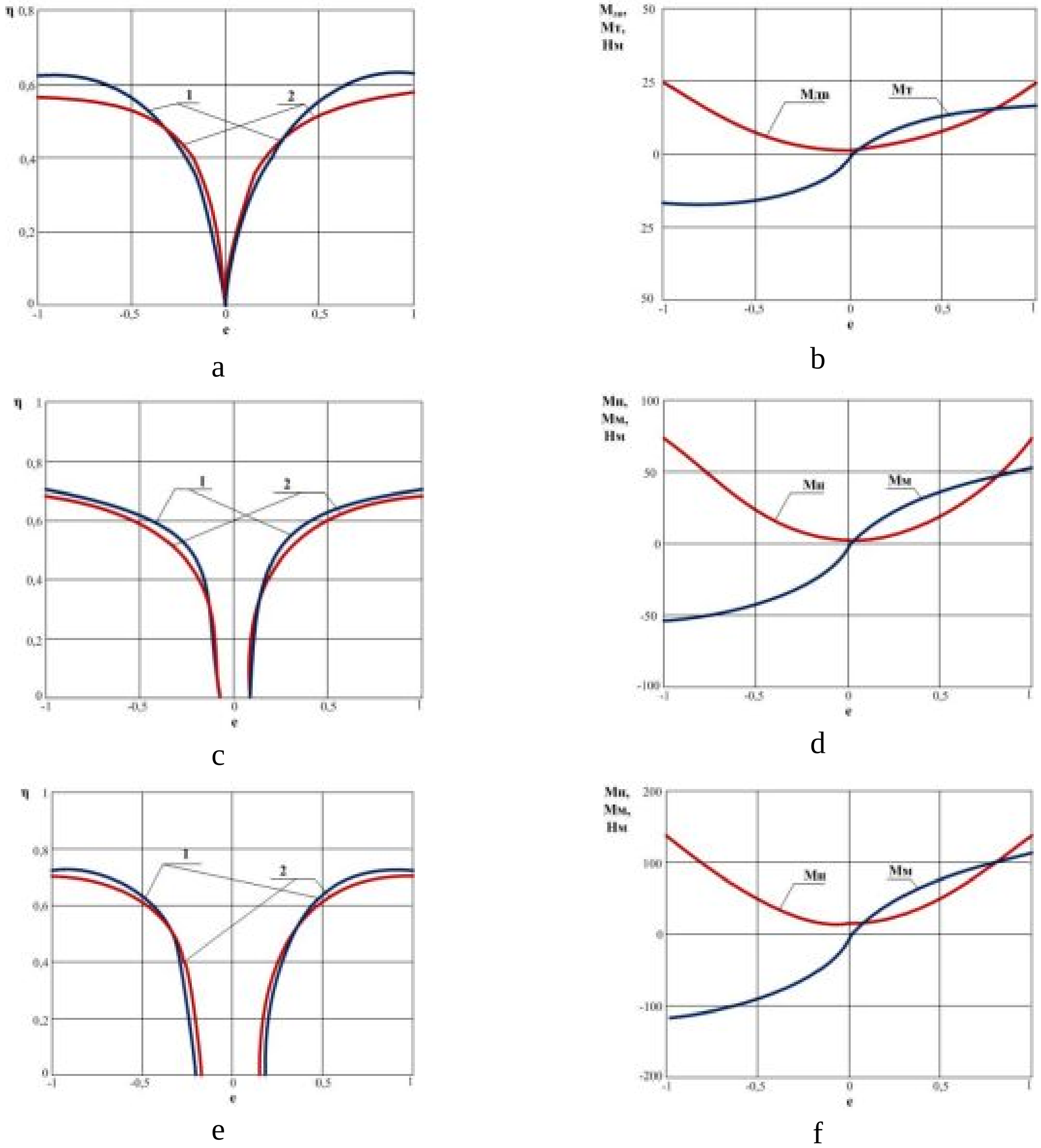

Fig. 6 – Results of the study of HST efficiency of GTN Hydraulics 3K10 P090:
a,b – “low loading”, c, d – “ average loading”; d, e – “ high loading”.

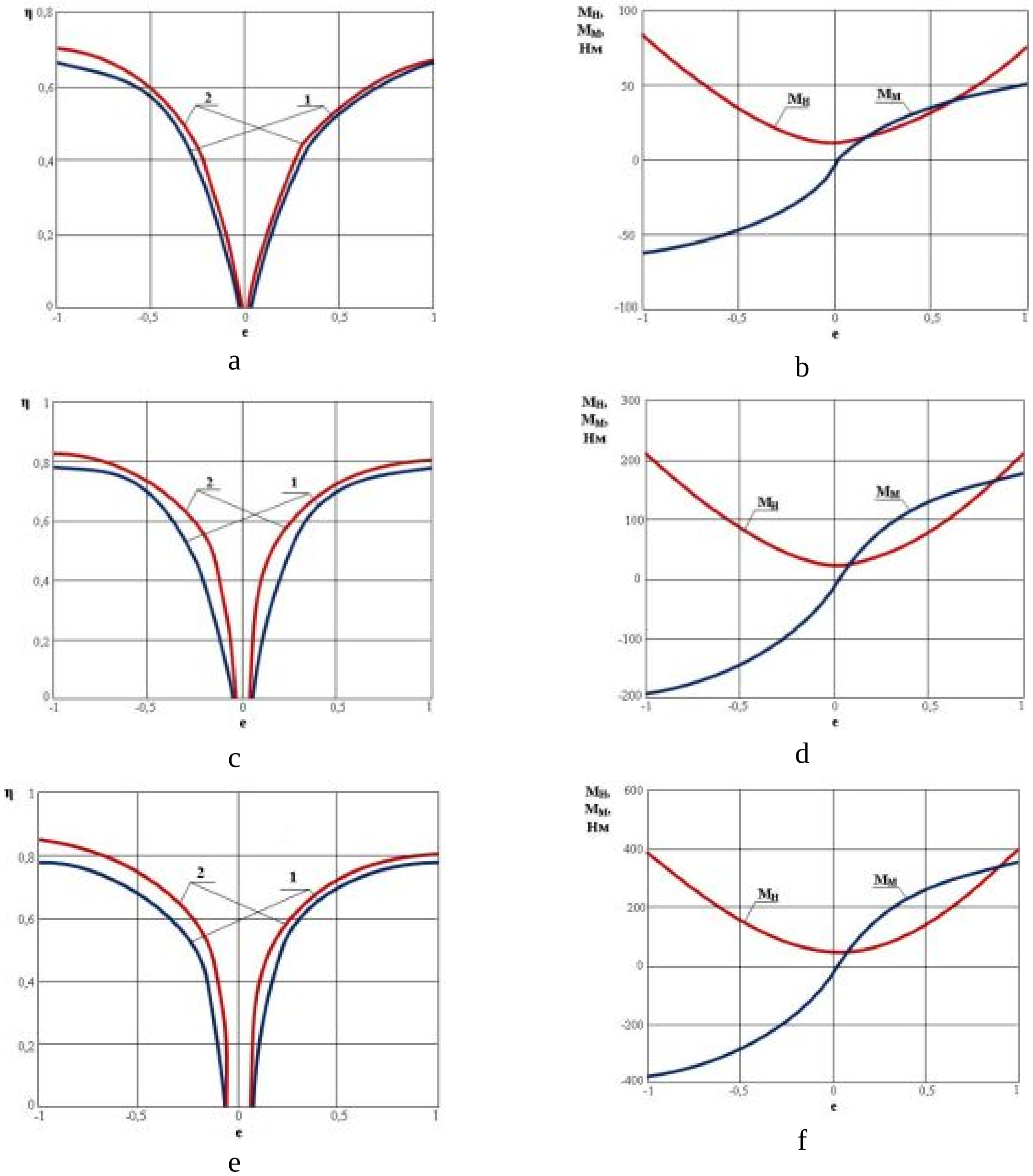

Fig. 7 – The results of study of HST efficiency NP112 5MHL / D2BCDBY1i MP112 2 / D2B35Y1:

a,b – “low loading”, c, d – “ average loading”; d, e – “ high loading”.

As it can be seen from Fig. 6–7, the experimentally obtained dependence of the HST efficiency at the stand №1 extends below the theoretical curve of efficiency for the values of the relative parameter of HST adjustment ***e***, greater than 0.2 ... 0.3 in absolute value, i.e., in the zone that constitutes 70–80% of the total area of HST adjustment. At stand №2 the experimentally obtained dependence of HST efficiency lies above the theoretical dependence of efficiency throughout the entire zone of HST adjustment.

Conclusions.

1. There was determined the computational and experimental efficiency of HST for two cases – with an adjustable hydraulic pump and non–adjustable hydraulic motor in

monoblock performance made in 1999 and HST with spaced hydraulic machines made in 2012.

2. When applying the calculated efficiency values of HST to the experimental one can clearly trace the trend towards the increase of the latter for subsequent hydraulic machines for all loading conditions.

3. Analysis of obtained dependences of HST efficiency shows the need to clarify the leakage factors and hydro–mechanical losses in the model of K. Gorodetsky for hydraulic machines of a new generation.

References:

1. *Gorodetsky K. Mechanical Efficiency of Volumetric Hydraulic Machines: Engineering Journal. 1977, №7; 19–23.*
2. *Gorodetsky K. Efficiency of Volumetric Hydraulic Transmissions: K. Gorodetsky, A. Mikhailin: Tractors and Agricultural Machinery. 1981, №9; 9–14.*
3. *Mittsel N. The Torque Sensor: Vestnik of NTU "KPI". 2014, №48; 3–9.*
4. *Samorodov V. Investigation of the Stepper Motor Drive Properties as a Control System of a Double–Split Hydrostatic Mechanical Transmission: V. Samorodov, N. Mittsel: East European Journal of Advanced Technologies. 2014, № 5/7 (70); 52–58.*