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CONCEPTUAL APPROACH TO SELECT PARAMETERS OF HYDROSTATIC AND MECHANICAL TRANSMISSIONS FOR WHEEL TRACTORS DESIGNED FOR AGRICULTURAL OPERATIONS

Igor Taran[1], Anatoliy Bondarenko[2]

[1] National Mining University, Faculty of Mechanical Engineering, Department of Transport Management, Dnipro, Ukraine

[2] National Technical University "Kharkiv Polytechnical Institute", Department of Automobile Construction and Tractor Construction, Kharkiv, Ukraine

[1]e-mail: taran_70@mail.ru

Abstract: *As a result of theoretical and experimental efforts, innovative scientifically grounded conceptual approach to select hydrostatic and mechanical transmissions (HMT) for wheel tractors designed for agricultural operations has been proposed. The approach is characterized by focusing on kinematic parameters, power parameters, and energy parameters of the transmission while performing technological operation "plowing"; it also takes into consideration braking features of a tractor in the context of various braking types. Application of the approach offers an opportunity to improve technical level of transmissions while updating current wheel tractors and designing new ones; moreover, that will better controllability as well as braking efficiency.*

The conceptual approach developed on the basis of proposed techniques and applied mathematical models makes possible to determine rational structure and basic design parameters of two-flow HMTs while designing; it also helps formulate recommendations concerning a technique for service braking and emergency braking under specific operational conditions.

Key words: *hydrostatic and mechanical transmission, wheel tractor, braking process, controllability, braking efficiency.*

1. Introduction

Agroindustrial complex is one of the most important sectors of the economies of many world countries. Efficient functioning of agroindustrial complex is impossible without the increase in competitiveness of agroindustrial machine-building products.

Need in stepless velocity control and tractive force as well as improvement of ergonomic properties while performing various technological operations have become main reason for the growth of worldwide production of wheel agroindustrial tractors with hydrostatic mechanical transmissions (HMT).

Increase in competitiveness of agroindustrial tractors with HMT is possible only at the expense of improvement of their technical level. Availability of both quantitative and qualitative regularities of changes in kinematic parameters, power parameters, and energy parameters of HMTs would allow not only reducing time for HMT development but also improving considerably transmission efficiency at the design stage.

The research has involved such modern methods as: comparing and finding analogues for structural analysis of HMTs of wheel tractor with various schematic representations both from technical and mathematical viewpoints; methods to solve differential equations of mathematical model concerning tractor braking process; methods of field testing in the process of experimental studies; statistic method to determine errors while comparing the results of both theoretical and experimental studies; method of morphological analysis for synthesis and analysis both foe current and prospective means to implement braking process of wheel tractors with various HMT; theory of optimization to determine optimal law of changing relative parameter of hydrostatic drive (HD) control while emergency braking of wheel tractors with HMT in terms of kinematic engine disconnection from driving wheels as well as to determine optimal point to locate adhesion within HMT.

2. Statement of the problem.

Currently, two-flow HMTs are the only type of stepless gear being mounted on agroindustrial tractors in series. Their application range widens both according to the number of tractor models and the power being transmitted. Share of wheel tractors with HMTs in terms of engine power range of 170-250 kW makes up almost 50%. Designs of HMTs are being developed towards the increase in power transmitted mechanically and decrease in the number of friction multidisk clutches as well as the decrease in the number of ranges (subranges) and complex mechanical parts respectively. Despite the popularity of HMT for tractor industry, modern HMT structures have considerable disadvantages. They are as follows: complicated structure and high production cost as a result; while subrange-to-subrange switching from within a contour housing hydraulic transmission, step-like changes in working fluid pressure are possible resulting in impact mode of HMT and its service life reduction; while braking from the velocity of 15 km/h and more and incorrect selecting of both braking mode and intensity of changes in HMT regulation parameters, there can be observed not only step-like changes in working fluid pressure within HMT but also sharp increase in values of HMT links angular velocities followed by overloading of both HMT and planetary gear set and adhesion.

Design and implementation of efficient and competitive HMTs for wheel agricultural tractors, meant not only for effective performance of basic technological operations but also for fast and safe load transportation, is possible only in terms of system approach availability (being considered in the paper) to determine main regularities of operating processes in such types of stepless transmissions. In addition it is required to detect and systematize the effect of the modes to implement service and emergency braking, laws of changes in HMT regulation parameters for kinematic parameters, power parameters, and energy parameters of HMT with various structures as well as on controllability and braking efficiency of wheel tractors.

3. Analysis of the results of recent studies and singling out the unsolved problem

The results of analyzing the tendencies and prospects to use stepless HMTs in terms of tractor industry as well as current state of the problem concerning structural analysis of HMTs and modeling of motion dynamics of wheel tractors with HMT have shown that authors of the majority of the considered papers, taking into account only personal designing experience and heuristic method, propose a configuration, basic structural parameters of two-flow HMTs and formulate recommendations concerning service braking and emergency braking of wheel tractor with HMT (Samorodov & Bondarenko, 2014; Samorodov & Pelipenko, 2016). Despite numerous research concerning development and analysis of HMTs for self-propelled machines (Pettersson, 2013; Popa & Buculei, 2013; Nielsen & Rozycki, 2012), too few attention is paid to the problem of structural analysis and control of braking of wheel tractors with stepless HMT.

Thus, modern designs of tractor transmissions of the given type require following improvements: refusal from the use of subrange-to-subrange switching to eliminate impact modes in HMT; prevention of power circulation within closed contour of HMT when tractor is performing its basic technological operations; increase in HMT efficiency; load reduction on both hydraulic components and mechanical components; that will make it possible to improve not only technical level of tractors but also the level of traffic safety while moving.

4. The objective of the paper

The objective of the paper is the development of conceptual approach to select HMT parameters for agricultural wheel tractors as well as the controllability and braking efficiency of wheel tractors with HMTs at the expense of systematic determination of basic regularities of operating processes in terms of stepless transmissions. In addition, the objective is to select reasonable way to implement braking process and the character of changes in HD control.

Statement of the main material. The proposed conceptual approach combines both theoretical and experimental research. Experimental research is carried out to verify the reasonability of the developed mathematical models use.

5. Theoretical studies

Determination of the structures and structural parameters of HMT for wheel tractors requires clear understanding of regularities of changes in kinematic parameters, power parameters, and energy parameters of HMTs of various designs. Moreover, it also helps find out and systematize key regularities of operating processes including power circulation in terms of such stepless transmissions.

Method of comparisons and analogies analogues for structural analysis of HMTs of wheel tractors with various schematic representations is the most convenient one to be used in this context both from technical and mathematical viewpoints.

The tendencies and regularities should be determined in terms of technological operation called “plowing” as it is the one ensuring the fullest tractor loading. In this context it is recommended to analyze HMT schemes both with input and output differential. Owing to the fact that HMT designs are developed towards the reduction of the number of friction multidisk clutches as well as the number of subranges and complex mechanical components (planetary gear sets in particular) it is not expedient to study HMTs with input differentials and output ones.

Integral structural analysis allows determining prospective schemes of HMTs covering the whole control range independently. Besides, the determined tendencies and regularities make it possible to reach the highest transmission efficiency when tractor is performing basic technological operations. That can be done by means of changes in transmitting relations and volumes of hydraulic machines. Moreover, it increases the transmission competitiveness significantly. In turn, traffic safety level improvement while operating wheel tractors with HMTs is possible owing to comprehensive analysis of braking process of such tractors.

Theoretical studies of braking process is impossible without developing generalized mathematical model of braking process in terms of wheel tractor with HMT which would take into account following description: characteristics of internal combustion engines and equation of crankshaft motion; operating processes within HMT; wheel-bearing surface interaction; braking system (description of operating processes of antilocking system, braking mechanism, and braking rigging was available); motion of both unsprung weight and sprung weight taking into consideration the effect of a suspension and tire stiffness which would also involve service braking and emergency braking, operation conditions, rules of pressing the brake pedal, and regularities concerning changes in control parameters of HMT.

Comparison of theoretical results with experimental ones obtained under laboratory conditions and application of field test methods helps test adequacy of HMT operating processes description while braking in terms of input-differential HMT scheme and output-differential HMT scheme. After that adequacy of the developed generalized mathematical model of tractor braking process in terms of Fendt 936 Vario wheel tractor braking and wheel tractor XT3-21021 with HMT-1C is verified.

Statistic method to determine error while comparing the results of theoretical and experimental studies allows assessing adequacy of the considered mathematical models.

Use of the methods to solve differential equations of mathematical models of tractor motion consisting of separate subsystems and components described with the help of theoretical and empiric dependencies as well as the method of morphological analysis. That makes it possible to synthesize and analyze both available and prospective means for service braking and emergency braking as well as rules of changes in control parameters of HMT hydraulic machines into kinematic parameters and power parameters of various-structure HMTs. Innovative technical solutions of wheel tractor braking to increase HMT productivity, controllability and braking efficiency of wheel tractors with HMTs are proposed.

A clutch is one of the most important HMT components; its location produces considerable effect on the working capacity of transmission. It is the theory of optimization that is used both to determine optimal clutch location (from the viewpoint of braking process dynamics) separately in terms of input- differential HMTs and those with output differential. Besides, optimal rule of changes in relative parameter of HMT control in the process of emergency braking of wheel tractors with stepless HMT in terms of kinematic drive disconnection from driving wheels is identified.

Let’s consider each stage in detail.

As HMT designs are developed towards the reduction of the number of friction multidisc clutches as well as the number of subranges and complex mechanical parts, it is proposed to analyze “input differential” and “output differential” schemes as those being the most popular and the constituent part of any HMT.

Following parameters are selected as the initial data required for the analysis: tractor mass is 9000 kg; maximum velocity being implemented within traction range in terms of coefficient of motion resistance being 0,5 (plowing) is 10 km/h.

In terms of HMTs operating according to "input differential", 6 variants are possible to connect mechanical branch and hydraulic branch with links of a planetary gear set being implemented in the form of 24 schemes of closed contours of HMTs. As an example, Fig.1 shows one of the studied HMT schemes with input differential; Fig. 2 shows HMT with output differential.

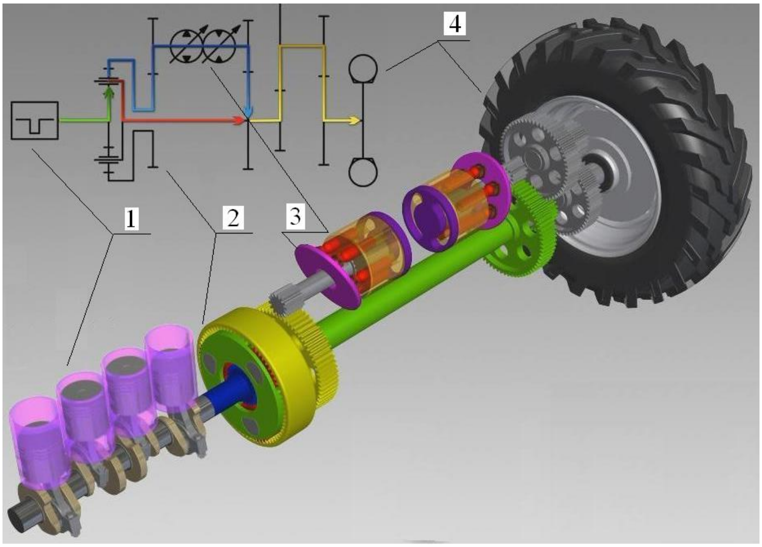

Fig. 1 Scheme of HMT with input differential: 1 is internal combustion engine; 2 is planetary gear set; 3 is HMT; 4 is a wheel.

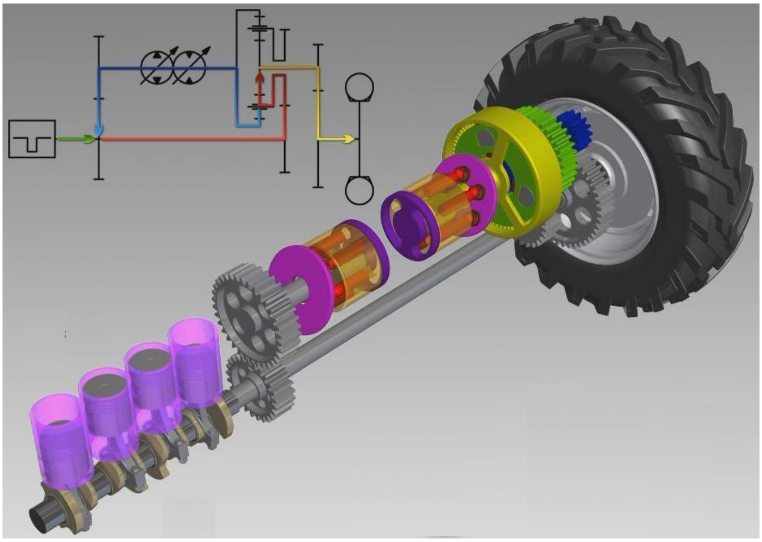

Fig. 2 – Scheme of HMT with output differential

Following stage after static analysis of all the possible HMT schemes is the study of power flows distribution within HMT. Values and directions of power flows transmitted by the links of two-flow transmission are determined according to the method adopted in classic planetary synthesis – by means of circular transmitting relation of the closed contour. Power flow directions within the closed HMT contour along parallel branches of two-flow transmission can be both the same (Fig.1, kinematic scheme, power flow direction is marked with the arrow) and different being transmitted in opposite directions, i.e. power circulation arises within the closed contour (Fig. 2).

According to the results of HMT complex static analysis, a series of kinematic prospective transmission schemes is formulated; their main structural parameters, standard sizes of HD hydraulic machines are identified; its kinematic parameters, power parameters, and energy parameters are determined; availability of power circulation is defined; and the availability of power circulation and velocities producing it are specified.

Since the velocity of modern tractors with HMT reaches 60 km/h, additional range is introduced into schemes of perspective HMTs. The transportation range switching to which allows implementing maximum velocity of 60 km/h in terms of coefficient of motion resistance being $f = 0,05$. Complete reversing is possible by means of introducing additional pair of pinion gears into HMT structure; in this context, values of kinematic parameters, power parameters, and energy HMT parameters while braking do not experience considerable changes.

Theoretical studies of braking process are impossible without generalized mathematical model of braking process of wheel tractor with HMT which would take into account following description: characteristics of internal combustion engines and equation of crankshaft motion, operating processes within HMT, wheel-bearing surface interaction, braking system (description of operating processes of antilocking system, braking mechanism, braking rigging), motion of both unsprung weight and sprung weight taking into consideration the effect of a suspension and tire stiffness as well as way of implementing service and emergency braking, operation conditions, rules of brake pedal pressing, and regularities of changes in HMT control parameters.

Reliability of braking modeling results depends immediately upon the acceleration dynamics (while accelerating transmission components are more loaded than during uniform motion; that is why the acceleration but not uniform motion is considered) as angular velocities at the moment of transfer from acceleration or uniform motion to braking mode are the initial conditions while braking to integrate angular accelerations of the links. It is the mathematical model of tractor acceleration that should be a part of conceptual approach from which the initial data are taken to model braking process.

Runge-Kutta method use to solve differential equation of the generalized mathematical model of tractor braking process helps show the effect of ways of service braking and emergency braking implementation, rules of changes in control parameters of HD hydraulic machines into kinematic parameters and power parameters of transmissions of various structures.

Clutch is one of the most important HMT components which location has effect on working capacity of transmission. Clutch can be located: right behind the engine (variant 1); within the mechanical branch of closed contour of HMT (variant 2); within the hydraulic branch of closed contour of HMT in front of HD (variant 3); and within the hydraulic branch of closed contour of HMT behind HD (variant 4). It is the theory of optimization (Hooke-Jeeves method) that is applied both to determine, from the viewpoint of braking process dynamics, optimal clutch area location separately for HMT schemes with input differential and HMT schemes with output differential and to define optimal law of changes in relative parameter of HMT control while emergency braking of wheel tractors with stepless HMTs in terms of kinematic drive disconnection from driving wheels.

Braking process of tractor with HMT is characterized by braking efficiency, stability, and controllability. Moreover, while HMT developing it is required to consider operating parameters of transmission components.

Trajectory stability (the highest deviation from the right line) as the assessment criterion is not considered in the paper. It is the process of maneuvering or turning that results in the highest deviations from the planned trajectory; this parameter is the trajectory controllability.

Trajectory controllability can be evaluated according to the value of tractor deviation from the specified trajectory. While determining trajectory controllability of a tractor, the curve along which tractor can move without transverse deviation in terms of specified initial velocity and constant setting angle of driven wheels is selected as a test trajectory.

To evaluate operating parameters of HMT it is convenient to use power parameters (operating pressure differential within HMT is $|\Delta P|_{\max}$) and kinematic parameters (angular velocity of satellites is $|\omega_s|_{\max}$, angular velocity of hydraulic pump shaft is $|\omega_1|_{\max}$ angular velocity of hydraulic drive is $|\omega_2|_{\max}$ as well as difference between values of angular velocities of driving and driven clutch shafts is $|\Delta\omega|_{\max}$). First of all, boundary values $|\Delta P|_{\max}$, $|\omega_1|_{\max}$, $|\omega_2|_{\max}$ depend on structural peculiarities of HMT; they are listed in specifications marked as P^*, ω_1^*, ω_2^*, P^* is for maximum possible pressure within delivery pipe of HD. Admissible value of angular velocity of satellites does not depend on HD parameters though it has its own limitation being 600 rad/s, i.e. $|\omega_s|_{\max} \le 600$ rad/s; it is marked as ω_s^*. Maximum admissible value of difference between angular velocities of driving and driven clutch shafts marked as $\Delta\omega^*$ depends on clutch type, its structural parameters etc.

Rule of change in $e(t)$ relative parameter of HD control is marked in the most significant way within the transmission parameters; in turn, the parameter has effect on braking efficiency and trajectory controllability.

Optimal laws $e(t)$ are applied for various HMT schemes, variants of clutch area locations and operating conditions.

6. Experimental research

By means of comparing theoretical results and experimental ones obtained under laboratory conditions applying field test methods it is possible to test adequacy of HMT mathematical model in braking mode in terms of input-differential HMT scheme with and output- differential one. After that adequacy of the generalized mathematical model of tractor braking in terms of Fendt 936 Vario wheel tractor braking and wheel tractor XT3-21021 with HMT-1C has been analyzed. Statistic method to determine error while comparing the results of theoretical and experimental studies allows assessing adequacy of the considered mathematical model.

Laboratory studies. Schemes of HMT with input and output differentials were used as the example to test adequacy of description of operating processes in HMT during braking mode under laboratory conditions.

The aim of the test was to determine the effect of the laws of changes in $e_1(t)$, $e_2(t)$ HD control parameters and laws of $M_g(t)$ braking torque change upon basic HMT parameters of various structures (two schemes were considered: with input and output differentials) while implementing braking process; the aim was also to test adequacy of the description of operating processes in HMT (approach to model development) used while modeling a braking process of a tractor.

The tasks of the tests were to determine angular velocities of ω_{e2*} hydromotor shafts, asynchronous electromotor ω_0 (assuming that ω_0 value is the basis to determine angular velocity of ω_{e1*} hydromotor shaft as well), ω_g powder loading electromagnetic brake (powder braking mechanism); P pressure in HD delivery pipe, P_p suction pressure being equal in value to the pressure generated by replenishment pump (value of operating pressure differential in HD is $\Delta P = P - P_p$); torque within M_0 asynchronous engine shaft and M_g powder braking mechanism shaft both while braking of powder braking mechanism shaft only at the expense of change in parameter of e_1 hydraulic pump control and while braking simultaneously at the expense of changing M_g braking torque and the parameter of e_1 hydraulic pump control in terms of preserving kinematic connection with electric motor in all the cases.

Object of the research was HMT with input and output differentials each consisting of planetary gear set, HD with maximum operating volume of hydraulic machine being 33 cm^3 each, reducing gears, and connecting shafts.

Two stands (Samorodov et al., 2015) were used while testing. Both stands consisted of planetary gear set; HD (controlled hydraulic pump, noncontrolled hydraulic motor); reducing gears (four reducing gears within HMT stand with output differential, three reducing gears within HMT stand with input differential); connecting shafts; powder braking mechanism (value of braking torque generated by powder braking mechanism depends immediately on the stress being set manually with the help of power source); asynchronous electric motor; two sensors of excessive pressure; stepping motor; inductive sensor of rotation frequency (three sensors within HMT stand with output differential, two sensors within HMT stand with input differential); two devices to measure torque; analogue-to-digital converter; laptop and other devices and apparatuses. As an example, Fig. 3 and 4 show scheme of stand with transmission with input differential.

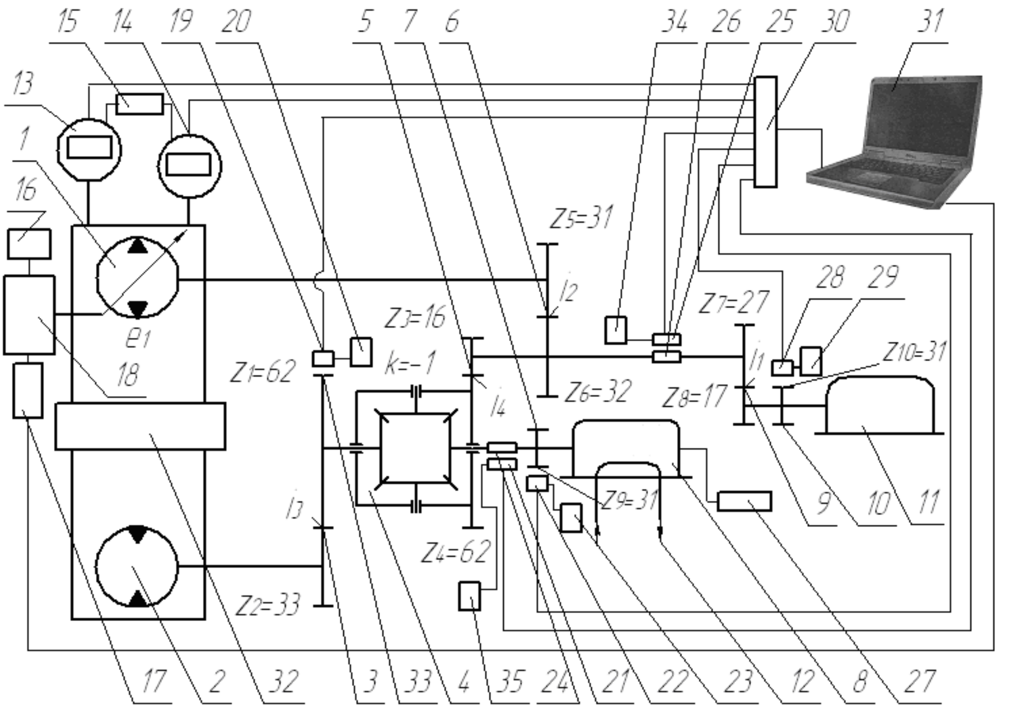

Fig. 3 – Scheme of HMT stand with input differential

Legend:

1 is controlled hydraulic pump; 2 is noncontrolled hydraulic motor; 3, 5, 6, 9 are reducing gear; 4 is planetary gear set; 7, 10, 33 are ring gear for inductive sensor of rotation frequency; 8 is powder braking mechanism; 11 is asynchronous electromotor; 12 is a system for powder braking mechanism cooling; 13, 14 are excessive pressure sensor; 15 is power source for excessive pressure sensor (36 V); 16 is power source for step motor (40 V); 17 is step motor driver; 18 is step motor; 19, 22, 28 are inductive sensor of rotation frequency; 20, 23 are power source for inductive sensor of rotation frequency (12 V); 21, 25 are unit to receive along radio channel, process and analogue-to-digital convert the signals from tensor bridge; 24, 26 are unit to amplify signals of tensor bridge, their analogue-to digital conversion and transmission along the radio channel; 27 is power source for powder braking mechanism; 29 is power source for inductive sensor of rotation frequency (5 V); 30 is analogue-to-digital converter; 31 is a laptop; 32 is a system for lubricant cooling within; 34, 35 are power source for unit to receive along radio channel, process and analogue-to-digital convert the signals from tensor bridge (5 V)

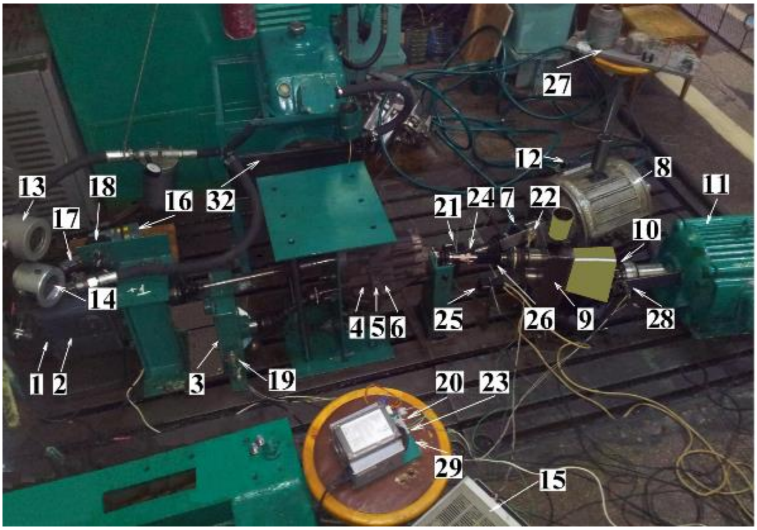

Fig. 4. General view of HMT stand with output differential
(legends are similar to the ones in Fig.3)

The regularity of changes in $e_1(t)$ hydraulic pump control parameters was specified using computer through FL86STH80-4208A step motor (torque is 4.5 Nm) with CNC 4.5A driver (Fig. 3–4, pos. 16–18).

Analogue-to-digital signal conversion was performed with the help of L-Card E14-140M 16-channel exterior (Fig.3 – 4, pos. 30).

The "Арктур-03" sensors (Fig. 3 – 4; sensor 13 records pressure in HD delivery pump and sensor 14 records pressure generated by replenishment pump) were used to measure excessive pressure; they are meant for continuous directly-proportional pressure conversion into direct-current signal. The sensor has been developed on the basis of "silicon-on-sapphire" tensor converter and microprocessor. To ensure long-term stability, tensor converter twisted into fitting is placed into heating and cooling chamber for two months where constant change of temperature modes eliminates all the stresses in the metal of membrane and tensor converter.

Angular velocities of shafts of hydromotor, asynchronous electromotor, and powder braking mechanism were determined with the help of inductive sensors of rotation frequency (pos. 19, 22 (Fig. 3 – 4) "ИДС" (the sensor is made according to GOST 15150-69 corresponding ТУУЗ.58-14310589-117-2001technical conditions), pos. 28 (Fig. 3 – 4) – sensor of angular velocity from antilocking system) characterized by reliability, simplicity, and relatively low cost.

During the studies torques were determined with the help of tensoresistors. In this context tensoresistors were glued on a shaft along the torsional stress line. The highest tortional stresses in terms of shaft torque transfer are observed within its transverse located at 45° angle to the generating line. Bridge circuits with four resistors were applied. Such circuits allow receiving powerful signal within bridge output and eliminating effect of shaft curve. Moreover, bridge circuits provide almost complete temperature compensation of tensoresistors.

To amplify and transmit the signal, set of devices developed at the Department of Automobile Construction and Tractor Construction Department was applied.

Stand tests took two stages:

- Stage one involved output-differential HMT assembling and braking process of powder braking mechanism shaft; first that was performed only at the expense of change in parameter of e_1 hydraulic pump control (in terms of different $e_1(t)$ laws), then simultaneously at the expense of change both in M_g braking torque parameter and e_1 hydraulic pump control parameter (in terms of different $M_g(t)$ and $e_1(t)$ laws) with the preservation of kinematic connections with electric motor in all the cases;

- Stage two involved stand readjusting to obtain input-differential HMT scheme; the same braking variants as for HMT with output differential are implemented for a new scheme.

Road studies. Adequacy of the generalized mathematical model of tractor braking process was tested in terms of Fendt 936 Vario wheel tractor braking process and XT3-21021 wheel tractor with HMT-1C being under study.

The aim of Fendt 936 Vario tractor testing was to carry out experiments of braking in motion with all and one driving axle within the road with different adhesion coefficients and varied drawbar force in terms of various laws of brake pedal pressing and joystick control (in the context of different deceleration rate – I, II, III, IV) as well as to verify the adequacy of the proposed mathematical model of braking process.

Straight-line motion in terms of $F_{kr} = 0$ kN drawbar force and curvilinear motion in terms of $F_{kr} > 0$ kN drawbar force were considered. Curvilinear motion means fixation of controlled wheels at the level of 5^0 right after the beginning of braking process. Drawbar force is generated at the expense of trailer transportation.

The tasks of the tests were to determine braking path S_g and maximum deviation of tractor from the specified trajectory $\Delta_{\max}$ while braking with all and one driving axle within traction and transportation transmission ranges on roads with different adhesion coefficient (dry asphalt, wet asphalt, snow) with $V_{\max}$ maximum possible velocity ($V_{\max} = 50$ km/h due to complex road conditions within the test area) under specified operating conditions with following $\Delta V = 10$ km/h step towards the decrease down to $V = 10$ km/h with drawbar force from $F_{kr} = 0$ kN with following $F_{kr} = 7{,}5$ kN step towards $F_{kr\max} = 15$ kN value ($F_{kr\max} = 15$ kN is the maximum value possible during experimental studies) in terms of different laws of brake pedal pressing.

Braking path S_g and maximum deviation from the specified trajectory $\Delta_{\max}$ were determined by means of double integration of longitudinal and lateral accelerations to be admissible owing to short period of braking process.

Fig. 5 shows equipment arrangement scheme within the tractor. The scheme consists of the following devices and facilities: video camera; 2 accelerometers; 2 laptops; electronic dynamometer.

To determine both longitudinal and lateral accelerations while Fendt 936 Vario tractor braking, mobile measuring device consisting of Freescale Semiconductor accelerometers of MMA7260QT model (pos. 5, 6, Fig.5), laptop 4 to process and store data, obtained during the experiment, was used. The measuring device has been developed by the Department of Technology of Machine-Building and Machine Maintenance of Kharkiv National Automobile and Road University.

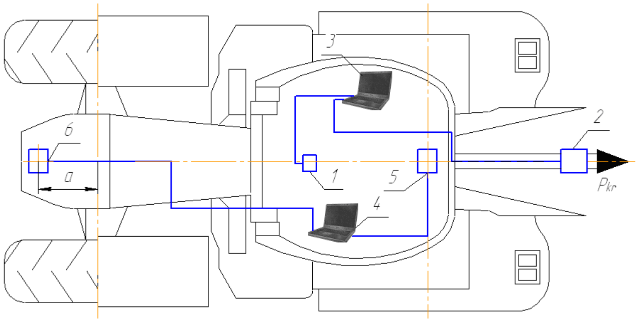

Fig. 5. Scheme of equipment arrangement within Fendt 936 Vario tractor (1 is video camera; 2 is electronic dynamometer; 3 is a laptop to save video camera and electronic dynamometer readings; 4 is a laptop to save accelerometers readings; 5, 6 are accelerometer; $a = 0{,}75$ m)

Use of accelerometers during experimental studies is stipulated by the possibility to determine values of longitudinal, lateral, and vertical accelerations of vehicles, evaluation of their aerodynamic, traction and velocity, braking properties, controllability and stability, motion smoothness etc. (Nagarkar et al., 2011; Andronic et al., 2014; Poongodi & Dineshkumar, 2013; Klets, 2013).

Final adequacy test of the mathematical model of tractor braking process was carried out in general in terms of XT3-21021 wheel tractor with HMT-1C braking.

The test was aimed at experimental study of service braking of XT3-21021 wheel tractor with HMT-1C being implemented with the help of change in e HD control relative parameter while moving with all driving axles on dry concrete road with metallic inclusions within various motion range; the aim was also to have final adequacy test of the proposed mathematical model (approach to model development) of braking process. Only one way of braking was considered for final confirmation of the adequacy of the proposed mathematical model.

Test tractor XT3-21021 with HMT-1C was the object of research. Straight-line motion in terms of $F_{kr} = 0$ kN drawbar force was considered.

The tasks of the test included determination of P pressure within HD delivery pipe, P_p suction pressure being equal to pressure generated by replenishment pump (further calculations and comparative analysis will contain value of operating pressure differential in HD – $\Delta P = P - P_p$ as it is ΔP to be determined during theoretical studies); ω_0 angular velocity of internal combustion engine crankshaft; V real velocity of tractor motion; law of change in relative control parameter $e(t)$ in terms of tractor braking having moved within I, II, III motion range.

Fig.6 demonstrates scheme of the tractor equipment arrangement. The equipment consists of following devices and facilities: 2 sensors of excessive pressure (to determine P pressure in HD delivery pipe and P_p suction pressure); inductive sensor of rotation frequency (to determine angular velocity of transfer gearbox shaft and specify V real velocity of tractor motion); Hall sensor 2SS52M (to determine n_0 rotations of internal combustion engine crankshaft); position sensors (to define law of change in relative control parameter $e(t)$) – 3590S-2-501 500R multiturn precision cable potentiometer; laptop; analogue-to-digital converter and other devices and facilities.

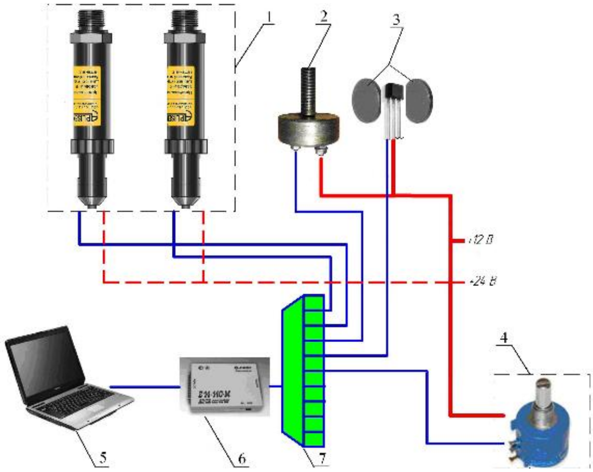

Fig. 6. Scheme of equipment arrangement within the test XT3-21021 tractor with HMT-1C (1is PC-28measuring pressure converter; 2 is inductive sensor of rotation frequency; 3 is 2SS52M Hall sensor; 4 is position sensor; 5 is a laptop; 6 is analogue-to-digital converter; 7 is BO-37F terminal block)

PC-28 measuring pressure converter operating within the range from 0.1 to 100 MPa is used as excessive pressure sensors.
Experimental results were processed involving Butterworth filter.
Comparison of theoretical results and experimental ones obtained under laboratory conditions was used to test adequacy of the description of operating process within HMT while operating in braking mode on the example of schemes with input and output differentials; It was defined that maximum theoretical and experimental results difference is no more than 6.0%. It was proved experimentally that it is possible to use the given HMT model either separately or as the constituent part of more complex – generalized mathematical model of braking process.
In general adequacy of the mathematical model of braking process is confirmed by experimental studies in terms of the considered XT3-21021 wheel tractor with HMT-1C and Fendt 936 Vario tractor. Maximum error while comparing theoretical and experimental results is mo more than 9.9% that allows performing qualitative complex theoretical research of braking process of wheel tractors with HMT in various ways on the basis of the developed generalized mathematical model.
Application expediency of the developed conceptual approach to the selection of HMT parameters for wheel tractors is proved in terms of tractors with 170 – 250 kW engine power for which series of prospective kinematic transmission schemes was determined according to the given methodology: 3 with input differential, 4 with output differentials (maximum values of coefficient of efficiency of the transmissions is within 0.826 – 0.862). Dynamics of acceleration process was studied on the example of technological operation “plowing” and braking of tractors.
The results of complex study of braking of wheel tractor with 170 – 250 kW engine power having HMT of various structures are the basis to determine that the most appropriate service braking is implemented at the expense of using change in relative parameter of HD control in terms of preserving kinematic connection with the engine. While using the braking mode, decrease in intensity of HD control parameters changing is accompanied by the decrease of operating pressure differential, reduction in angular velocities of transmission links as well as increase in braking path and decrease in tractor deviation from the specified trajectory. According to the results of analysis of emergency braking of wheel tractor with various HMT schemes that can be implemented only in terms of engine kinematic disconnection from driving wheels it is determined that there is no single optimal law of change in relative control parameter for all the transmissions. First of all it is connected with the fact that minimum value of operating pressure differential in HD, angular velocity of satellites, angular velocity of hydraulic pump and hydromotor shaft do not always correspond to minimum value of the difference in values of angular velocities of driving and driven clutch shafts. Being applied, the recommendations make it possible to increase technical level of transmissions while modernizing the available and designing new wheel tractors as well as to improve controllability and braking efficiency of tractors while braking.
In the process of emergency braking in terms of engine kinematic disconnection from driving wheels the transmissions operating capacity is preserved only in case of correct selection of the point of power flow break, i.e. the precise point of engine disconnection from driving wheels. As a result of solving optimization problem, following fact is determined: from the viewpoint of braking process dynamics and values of generalized criterion, HMT clutch with input differential is recommended to locate behind the engine or in hydraulic branch of closed contour in front of HD (each variant has no clear and obvious advantage); in terms of HMT with output differential it is preferable to locate the clutch within hydraulic branch of closed contour behind HD.
It has been defined that in case of emergency braking in terms of engine kinematic disconnection from driving wheels, change in HD control parameters to increase HMT operating capacity, controllability, and braking efficiency of a tractor (in terms of using antilocking system) should be performed automatically corresponding to the change in real velocity of tractor motion. Following the recommendations makes it possible for a driver to stop emergency braking at any stage without any negative effects for the transmission and continue tractor motion or acceleration to maneuver if necessary. It allows improving considerably traffic safety level.

7. Conclusions and recommendations as for the further use

Use of the developed conceptual approach to the determination of HMT rational structure and parameters for wheel tractors oriented not only to study of HMT kinematic, power and energy parameters when tractor performs technological operation "plowing" but also to peculiarities of changes in transmission kinematic and power parameters during various braking modes of a tractor, simplify significantly the development of competitive wheel tractors with HMT and improvement of traffic safety level.

The results of the study can be recommended to be used by enterprises involved into HMT design and manufacture as well as by organizations dealing with operating wheel tractors with HMT.

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