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RESEARCH OF ROBOTISED MANUAL TRANSMISSIONS FOR ALL-TERRAIN VEHICLES

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ABSTRACT

The object of study is transmissions of all-terrain vehicles (6x4, 8x8). The purpose of the research is the creation of multi-robotic mechanical transmissions for all-terrain vehicles. The work describes the main technical characteristics of the vehicles, as well as units that are subject to automated control: friction clutch, gearbox and distributor box with electro-pneumatic drives. It represents the basic requirements for robotic units and units with automatic control. The work represents laws for gear-shifting of manual transmissions vehicles and enlarged algorithms of gear -shifting when operating in difficult environmental conditions on the support surface with a high-coefficient of resistance to movement. We've determined forces of the resistance to movement, and the maximum values of decreasing the vehicle speed while shifting gears. We have studied the principle schemes of the robotized manual transmission control. The paper represents mathematical expressions, on the basis of which the basic parameters of the experimental samples of executive mechanisms for transmission units control are determined. It represents the stand for laboratory experimental studies of robotic mechanical transmissions and the results obtained on the stand during the work of the transmission units in the automatic mode. The analyzed parameters when working at the stand is the total time of gear-shifting, the clutch switching (on\off), time and magnitude of pressure increase in the transmission automatic control mechanisms and power transmission control cylinder, time of synchronization, dynamic loads during gear shifting, the overall power losses in the work of transmission units and other parameter. The article describes the results of experimental studies of the work of units and the control units of the robotized manual transmission as part of all-terrain vehicle (6x4) in different driving conditions, including deformable bearing surfaces. It also shows the comparison of calculated data and experimental results obtained on the stand and on the road. It represents conclusions and recommendations on creation of robotic mechanical of multi-speed transmission with automatic control for all-terrain vehicles.

Keywords: all-terrain vehicles, robotic mechanical transmission, multistage transmission, pneumatic actuator, synchronization process, automatic control.

INTRODUCTION

One of the main areas is the automation of mechanical gear transmission control, especially the control of a multi-stage gearbox and a clutch. The invention of microprocessors, capable of operating in harsh environments of all-terrain vehicles with high speed, allows not only to improve the accuracy of control, but also to increase the number of information parameters, thus realizing more flexible control logic. As a result, the automatic transmission control system leads to improved efficiency of the use of the engine power as well as traction and speed properties, it improves fuel efficiency and reduces the driver's fatigue

Improving the technical level of modern vehicles, including all-terrain ones, is highly correlated with the degree of saturation of their units and systems with automatic devices based on microprocessor control.

Microprocessor control provides improved performance of vehicles in many respects: technical, economical and ergonomic. Complexes with microprocessor control which is just a little more expensive than mechanical counterparts are, but by the aggregate measure price / quality is much superior to them.

One of the main areas is the automation of mechanical gear transmission, especially the multi-stage gearbox and the clutch. The invention of microprocessors, capable of operating in severe environments and all-terrain vehicles with high speed, can not only improve the accuracy of control, but also increase the number of information parameters, thus realizing a more flexible control logic. As a result, the automatic transmission control system leads to improved efficiency of use of engine power and traction and speed properties, improves fuel efficiency and reduces driver fatigue.

Characteristics of the research subjects

The subjects of study in this paper are the KAMAZ-65117 off-road vehicle (6x4) and the RUSAK-3993 all-terrain vehicle (8x8). Table-1 shows the main technical characteristics of KAMAZ-65117 (Figure-1).

Figure-1. KAMAZ 65117 (6x4).

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Table-1. Main specifications of KAMAZ-65117.

Manufacturer					KAMAZ, RUSSIA			
The weight parameters and load								
Carrying capacity of th vehicle, kg					14500			
The total weight of a vehicle, kg					24000			
The load on the rear axles, kg					18000			
The load on the front axle, kg					6000			
Engine								
Engine model					KAMAZ 740.62.280			
Max. torque Nm (kgcm)					1177 (120)			
at crankshaft rotational speed, rev / min					1300			
Maximum net power, kW (hp)					206 (280)			
at crankshaft rotational speed, rev / min					1900			
Displacement, l					11,76			
The location and number of cylinders					V-typed, 8			
Clutch								
Model					ZF SACHS MFZ 430			
Transmission								
Model					TM16-2000			
Transmission ratio								
U_1	U_2	U_3	U_4	U_5	U_6	U_7	U_8	U_9
14,2	11,4	9,98	8,03	6,75	5,45	4,63	3,7	3,1
U_{10}	U_{11}	U_{12}	U_{13}	U_{14}	U_{15}	U_{16}	R	
2,5	2,2	1,7	1,5	1,2	1,0	0,81	11,7	
Main gear								
Gear ratio					5.94			

Table-2 shows the specifications of the RUSAK-3993 all-terrain vehicle, developed by the specialists of the NNSTU and the KOM Group (Russia).

Figure-2. 3D model of the RUSAK-3993 terrain vehicle (8x8).

Table-2. Main technical characteristics of the RUSAK-3993.

Axle configuration	8x8 / all
Passengers capacity (with driver), persons.	7
Dimensions, mmL x W x H	6890 x 2520 x 3085
Gross vehicle weight, maximum, kg	6400
Payload, minimum, kg	2000
Engine (type)	Diesel
Transmission	Mechanical, robotic
Clutch	Frictional, dry, single disk
Gearbox	Manual, 6-speed, M6-700
The gear case (between 1, 2 axes and 3, 4 axes)	Mechanical, 2-speed, with differential lock
The main gear (make, type)	Double, spaced, includes a central wheel transmission
Tyres	Low pressure, AVTOROS 1300x700-21 "
Inflation of the tyres	Central tire-inflation system,
Water propulsion	YES

Requirements for robotic units and transmission units

Analysis of work on transmissions control systems vehicles will allow to separate the following basic requirements for the clutch control mechanisms, as well as selection and switching of gears:

a) The clutch scram (maximum 0.25 seconds) during gear shifting and fast, but gentle start after switching;
b) The clutch switching when the rotational speed of the engine crankshaft drops to the idle speed, so that in case of a car braking to a full stop the possibility of stopping of the engine is excluded;
c) The gear of torque in the opposite direction (from the drive wheels to the engine) with the engine running or not. It is necessary for the engine braking when driving the car and at the parking lot, as well as starting the engine by the car towing;
d) The prevention of the transmission parts from overload by the dynamic torque.
e) The design of the mechanism must exclude the possibility of the air leakage from the cavities of the accelerating mechanism and pneumatic cylinder of the clutch.
f) The pneumatic air supply of the control mechanisms should be implemented from the vehicle pneumatic system with a maximum pressure of 7.2 kg / cm2,
g) Solenoid valves must be integrated in the control mechanisms that will prevent the airway and thereby will improve the performance and reliability of the mechanism.
h) The gearshift control mechanism is performed as a separate unit and is mounted on the gearbox.
i) An average (neutral) position of the pneumatic cylinders of selection and switching of gears should be ensured by preliminary preloaded springs of double action.
j) The mechanism design must exclude the possibility of the air leakage from the pneumatic cavities into the transmission.
k) The gear-switching by mechanisms is carried out only after a full clutch release.
l) The electric signal for the clutch switching must be carried out by the mechanism only after a full switching of a gear or neutral.
m) The mechanism must have a mechanical lock, that excludes the simultaneous operation of 2-gears and a lock that prevents the possibility of movement of the stem forks under the action of inertial forces.
n) The mechanism should be designed to enable the possibility of mechanical gear-switching for the movement of the vehicle in the absence of pressure in the pneumatic system (emergency mode).

Let us consider the basic requirements for the controller:

a) The possibility of easy selection of gears in any sequence without distraction of the driver's attention from the road.
b) The possibility to easily set the handle of the controller in the "neutral" position.
c) Reverse gear locks to prevent it from accidental operation.
d) 4 For the purpose of convenience of selection of the gears the controller lever must be fixed on different strokes.
e) The controller shall provide, if necessary, the possibility of the forced clutch release.

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f) In order to ensure the requirements of the proper safety operation of the vehicle the controller must let the engine start only when the handle of the controller is in the "neutral" position.
g) The controller must be carried out on the "Double-H" - shaped pattern of the lever movement, which best meets the above mentioned requirements and does not require changes in the driver's skills.
h) The controller must ensure the possibility of the electric scheme operation in the standby mode.
i) The controller must ensure with minor modifications the possibility of universal application for the control the 6, 8-and 16-speed transmissions.

The equations of the vehicles movement in case of gear changes

Among the works devoted to the theory of a vehicle, it is known that the progression denominator *q* is determined by:

$$q = \frac{u_{ki}}{u_{ki+1}} = \frac{n''_e}{n'_e}, \tag{1}$$

where u_{ki}, u_{ki+1} – neighboring transmission gear ratios

At this interval $[n'_e, n''_e]$ it can be called a theoretical range of variation of the engine speed when shifting gears or theoretical kinematic rmp interval of the leading parts of the transmission. We can characterize this interval through dimensionless quantity q, which will be called an indicator of the kinematic rpm interval of leading parts of the transmission. In real operating conditions, the process of gear shifting occurs over time and is generally connected with a change of the vehicle speed, which actually results in a significant difference of the actual values of the rpm interval of the leading parts of the transmission from the kinematic interval caused only by the interval of stages of gear ratios.

The value of the indicator, which characterizes the actual rpm interval, in case of shifting to a higher gear can be determined:

$$\lambda_в = \frac{v_0 \cdot q}{v} \tag{2}$$

The shifting to a lower rate, which characterizes the actual rpm interval will be expressed by the following equation:

$$\lambda_н = \frac{v \cdot q}{v_0}. \tag{3}$$

The differential equation of the vehicle during gear shifting:

$$\frac{dv}{dt} = -\frac{3{,}6 \cdot g}{(1+\xi) \cdot G}\left(G \cdot \Psi + \frac{k \cdot Fm \cdot v^2}{3{,}6^2} \right) \tag{4}$$

Trying to solve the equation (4), we obtain expressions for determining the index characterizing the actual rpm interval, in the case of shifting to a higher gear and a lower gear:

$$\lambda_в = \frac{q}{\dfrac{3{,}6}{v_0}\sqrt{\dfrac{G \cdot \Psi}{k \cdot Fm}} \cdot tg\left[arctg\, \dfrac{v_0}{3{,}6 \cdot \sqrt{\dfrac{G \cdot \Psi}{k \cdot Fm}}} - \dfrac{g \cdot t \cdot \sqrt{\dfrac{\Psi \cdot k \cdot Fm}{G}}}{1+\xi} \right]} \tag{5}$$

$$\lambda_н = \frac{3{,}6 \cdot q}{v_0} \cdot \sqrt{\frac{G \cdot \Psi}{k \cdot Fm}} \cdot tg\left[arctg\, \frac{v_0}{3{,}6 \cdot \sqrt{\dfrac{G \cdot \Psi}{k \cdot Fm}}} - \frac{g \cdot t \cdot \sqrt{\dfrac{\Psi \cdot k \cdot Fm}{G}}}{1+\xi} \right] \tag{6}$$

In the study of equations (5) and (6) the most clear and expedient is to analyze the index change, which characterizes the actual speed and rpm interval in the factor function $t{\cdot}\Psi$, because arguments t and Ψ mutually complement each other, fully expressing the physical essence of the phenomena. Figure-3 shows a graphical representation of the functional dependence of the index value, which characterizes the actual rpm interval of a gear change from the time of gear shifting in case of shifting from a lower to a higher hear and from a higher to a lower gear when $\Psi \geq 0$.

When driving on deep snow with greater resistance to motion when shifting to a higher gear the value of the actual rpm interval increases greatly, which in its turn leads to an increase in the synchronization operation, so designing the electro-pneumatic control system of the transmission and clutch drive parameters should be chosen so that they have a maximum synchronizing torque; allowing to improve the dynamic quality of the experimental sample of the all-terrain vehicle and simultaneously reduce the amount of slippage work and therefore rate of wear of the synchronizers.

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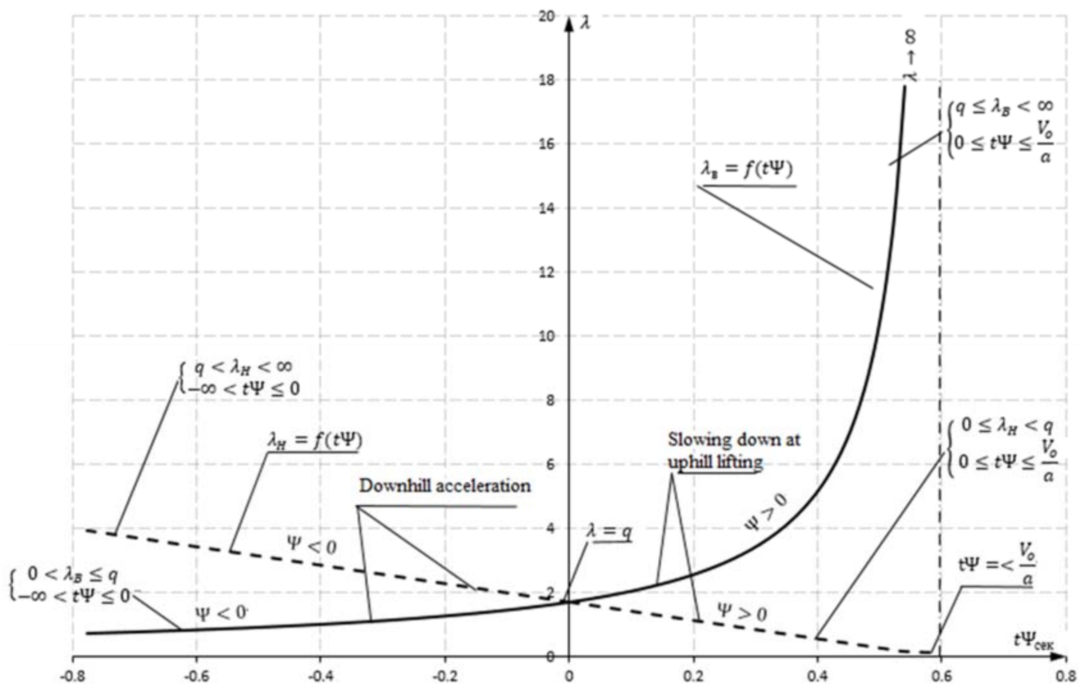

Figure-3. Functional dependence of the index value, which characterizes the actual rpm interval of a gear change from the factor function t·Ψ.

The calculations revealed that for large values of the total resistance of the road (when driving on deep snow) the value of the gearshift time has significantly greater impact on the value of the actual rpm interval than the actual value of the kinematic stage interval. Therefore, the transmission control system actuators should ensure the quickest gear-shifting (maximum 0.8 - 1.0.)
From Figure-3 it follows:

a) when $t{\cdot}\Psi=0$, and $\Psi\neq0$, then $v=v_0$, $\lambda_h= \lambda_l =q$, i.e. if the time of shifting is equal to zero, the actual rpm interval is equal to the kinematic one;

b) when the vehicle loses speed during gear shifting, which can take values in the interval $0<v<v_0$, the factor $t{\cdot}\Psi$ takes value in the interval $0 < t \cdot \Psi < \frac{v_0 \cdot (1+\xi)}{g}$, in this case the value $\lambda_h >> q$, and the value $\lambda_l << q$.

c) if the vehicle has lost speed to zero during gearshifting, then $\lambda_h=\infty$, a$\lambda_l=0$.In this case we have correlation $t \cdot \Psi = \frac{v_0 \cdot (1+\xi)}{g}$.

From this relation it is always possible to determine the maximum time during which gear shifting should be carried out without stopping the vehicle.

It should also be noted that the value of the actual rpm interval allows you to determine the value of engine power after the process of gear shifting.

The general scheme of the transmission program control

The program multistage transmission control in automatic mode consists of several modules. The transmission control unit (main modules) provides communication between them. Functional diagram of the interaction between the main modules with each other, as well as with the engine control unit is shown in Figure-4.

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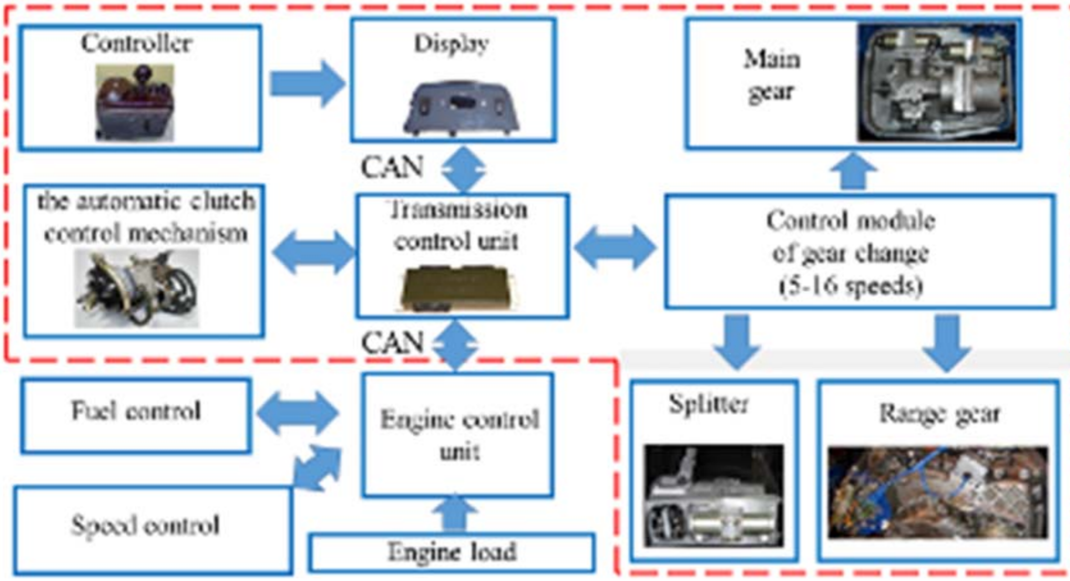

Figure-4. Functional diagram of the interaction of the main modules of the program.

The general scheme of the multistage transmissions control program is made in such a way that when the program starts the main module is used for entering of the main variables that will be used in the modules of the gear-switching control and clutch control, then there is a call of standard libraries designed for preparation of the modules of the gear-switching control and the clutch control, then these modules are initialized and activated. The control module of the gear-switching is performed continuously until the completion of the program, and the clutch control module is performed is interrupted with a period of once per millisecond maximum.

The clutch control module (Figure-4) in real-time mode there is monitoring and control of the distance between the clutch plates, that provides smooth and precise on / off switching of the clutch depending on the position of the gas pedal and the engine rotation speed changes when starting the vehicle and shifting gears.

The algorithms for automatic on / off gear switching of the main transmission gearbox, the divider and the splitter are implemented in the control module (Figure-4) in accordance with the Law on automatic transmission. Through a constant exchange of information with the main module in the real-time mode based on the analysis of information on engine rotation speed, load, fuel supply and data from the controller (on which the automatic control mode is selected) we make a decision a decision on the possibility and necessity of gear shifting.

The module of work with the display unit (Figure-4) with the help of the LEDs and indication of certain characters allows the operator (driver) to have visual information as input signals to the switching of a particular gear, and directly on switched positions in each of the 3 gears (the main gearbox, the divider and the splitter) and the clutch control mechanism. The module of work with the display unit has its own microcontroller that provides a permanent data exchange with the main module on the CAN bus

The controller (Figure-4), a setter of a gear for motion in this mode is set to the position corresponding to automatic shifting.

The main module synchronizes the data transfer between the modules of the program and controls the operation of the main control unit of the multi-stage transmission.

To connect the program with other software programs (the engine control unit and the indicator with their microcontrollers) the protocol SAE 1939 is used on the CAN- bus.

Mathematical model of the actuators

Figure-5 shows the designed model of the electro-pneumatic transmission control drive (Blokhin *et al*., 2013 a, 2013 b, 2014 a, 2014 b).

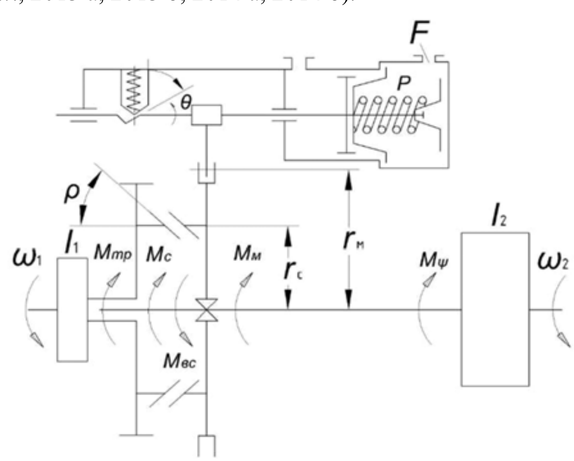

Figure-5. The designed model of the electro-pneumatic transmission control drive.

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The main parameters of the electro-pneumatic of the transmission control drive: the cross section of the feeding channels, the diameters of the active elements, the rigidity of the system, the allowed values of the friction components were determined based on the equation:

$$m\frac{d^2x}{dt^2}+c\cdot x=a\cdot t-b \tag{7}$$

In equation (7) the *a* coefficient is connected with a variation of pressure in the system, which is defined by the expressions:

$$\frac{dp}{dt}=\frac{RT}{V}\mu F\sqrt{\frac{2g}{RT}p_p^2\frac{k}{k+1}\left(\frac{2}{k+1}\right)^{\frac{2}{k-1}}} \tag{8}$$

or

$$\frac{dp}{dt}=\frac{RT}{V}\mu Fp_p\sqrt{\frac{2gk}{RT(k-1)}\left[\left(\frac{p}{p_p}\right)^{\frac{2}{k}}-\left(\frac{p}{p_p}\right)^{\frac{k+1}{k}}\right]} \tag{9}$$

Expression (8) is used for the air outflow critical zone and (9) is use for the subcritical zone.

Figure-5 shows behavior of the pressure in the power cylinder (p) and displacement of the piston (x).

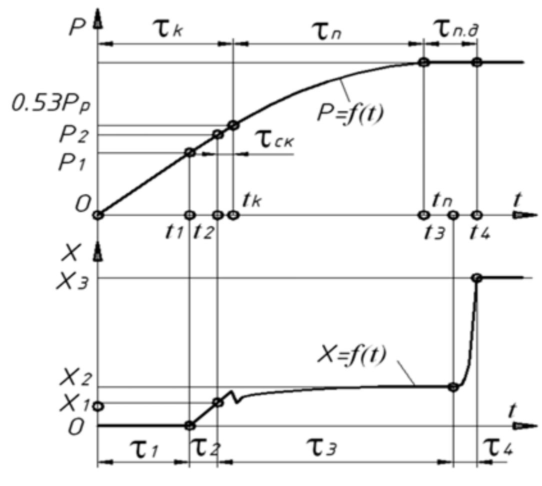

Figure-6. Behavior of the pressure in the power cylinder (p) and displacement of the piston (x).

Dynamics of change of rotational speed ω_1 и ω_2, and the input and output of the shafts is determined by the dependencies:

- when switching to high gear:

$$\begin{aligned} J_1 d\omega_1 &= (M_c + M_{TP} - M_{B.C})dt, \\ J_2 d\omega_2 &= (M_c - M_M - M_\psi)dt; \end{aligned} \tag{10}$$

- when switching to low gear:

$$\begin{aligned} J_1 d\omega_1 &= (M_c - M_{TP} - M_{B.C})dt, \\ J_2 d\omega_2 &= (M_c + M_M + M_\psi)dt. \end{aligned} \tag{11}$$

Solving the resulting differential Equation (8) and (9) together with the differential equation of the piston motion Equation (7), we determine the synchronization time.

The main parameters of the electro-pneumatic transmission control drive was chosen taking into account vehicle's hard operating conditions (for example, movement of the 8x8 vehicle in a deep snow). In such condition, the actual rpm interval must not differ by more than twice from kinematic rpm interval.

RESULTS

The results of experimental studies

The NNSTU named after R. E. Alekseev (in city Nizhny Novgorod, Russia) carried out experimental research on a specialized stand, RF patents №154871, №154102 (Figure-7) for testing of transmissions with command control and automatic control (Blokhin *et al*., 2015 a, 2015 b, 2015 d, 2015 f). The stand allows us to carry out the study of efficiency of the units of mechanical transmissions (the clutch, the divider, the main gear, the splitter, determining the efficiency of the units, the study of the synchronization process, etc.) and mechatronic control systems for automatic or semi-automatic modes. At this stand, we worked out the clutch control algorithms for multi-stage mechanical gearboxes.

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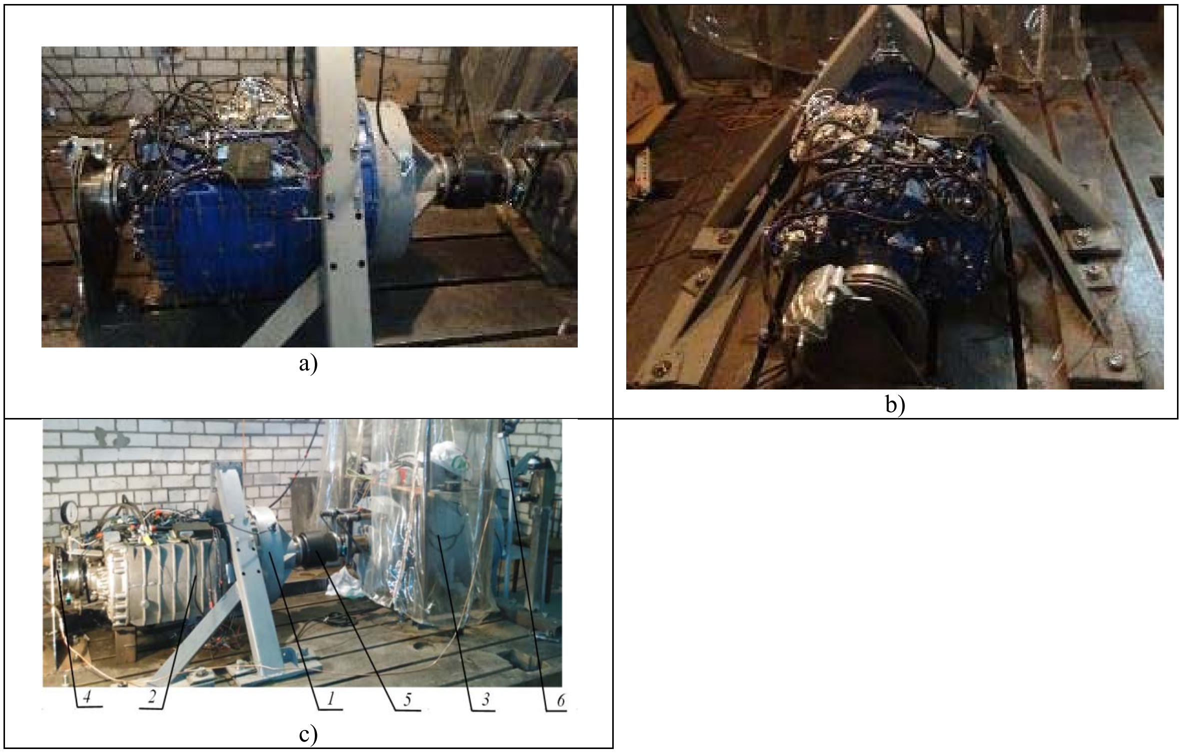

1 - the researched clutch control mechanism, 2 - the multi-stage transmission; 3 - the drive motor; 4 -the disc friction brake with a hydraulic drive; 5 - the driveline; 6 - remote control stand.

Figure-7. The test bench for the study of the efficiency of the automatic clutch control mechanism of the transmissions TM16-2000.

Figure-8 shows the fragments of oscillograms showing changes in the main adjustable clutch parameters at the moment smooth starting: 1 - position of the clutch fork rod; 2 - pressure in accelerating mechanism; 3 - pressure in a clutch control power chamber; 4 - the angular velocity of rotation of the motor shaft [rpm.]; 5 - torque on the motor shaft [Nm], 6 - the angulare position of the gas pedal [%].

Clutch plug rod movement (line 1) plotted on the supporting vertical axis and other parameters - by main. As a resul of studies on the test stand we found that at initial pressures in the receiver of 950 kPa, and at the angular velocity of the motor shaft 650 rpm, time of starting off the vehicle (gross weight of 20 t.) for second gear was 1.8 s.

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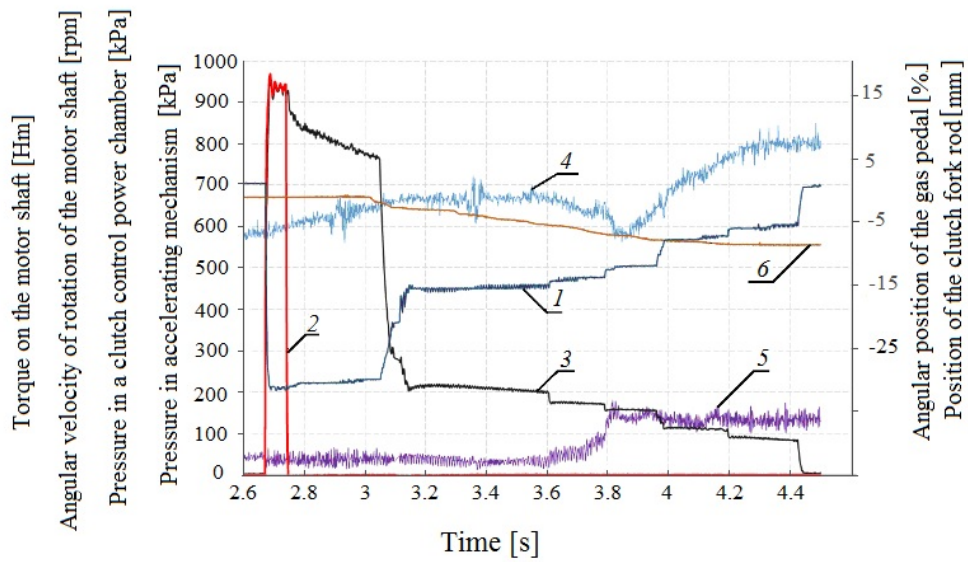

Figure-8. Experimental data of the parameter changes in the process of smooth starting.

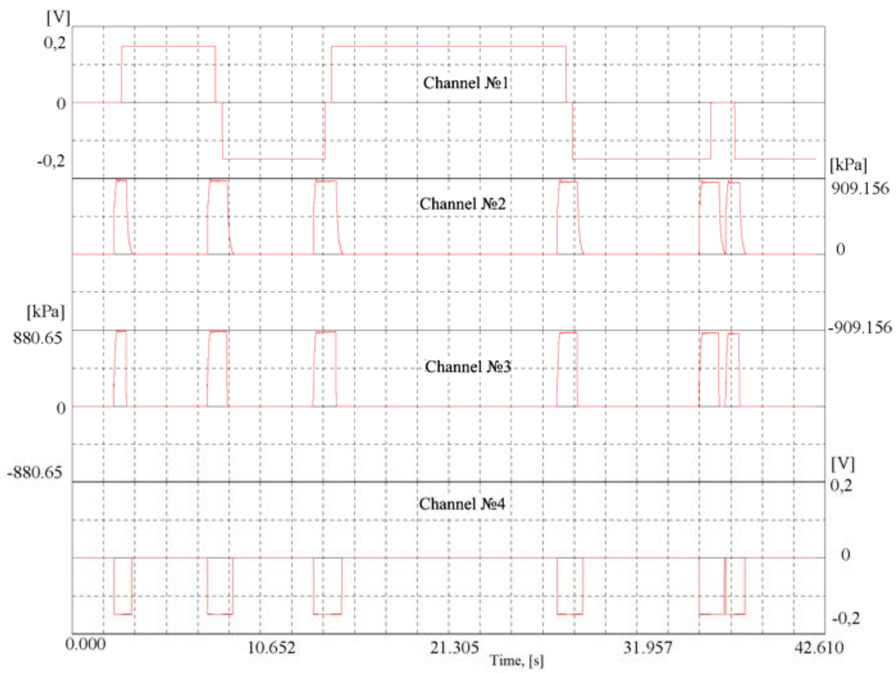

Figure-9. The experimental data on the stand on the process of a gear shift in the transmission TM16-2000.

The results of the experimental studies have shown:

- The time of the clutch turning on / off realized by the developed mechanism of automatic clutch control complies with the requirements;
- The established dependence of the clutch fork rod from the position of the gas pedal provides a smooth start;
- We determined the value of the rational engine shaft speed for the vehicle starting. Within this value the movement with a clutch slip is carried out.- The developed algorithms of the clutch control of the vehicle on the stand allowed us to reduce 4 times the dynamic loads at the vehicle starting, as well as reduce the time and the clutch slipping, which significantly increases the longevity of the transmission parts.

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- the total turn-on time of the transmission
- time of the pressure rise in the power cylinder, the main gearbox, the divider, the splitter;
- the response time of the main gear mechanism and a divider in the selection and a given transmission;
- the dynamic moments on the motor shaft and the output shaft of the transmission;
- the rotational speed of the motor shaft and the output shaft of the transmission.

Figure-8 shows the results of experimental studies of sixteen staged transmission TM16-2000 with experimental samples of the automatic and command drive control in the following sequence 2-4-6-8. A signal from the microswitch of gears of the main gearbox is represented at channel №1, a signal from the pressure sensor in the accelerating mechanism of the clutch control is represented at channel №2, a signal from the pressure sensor in the power chamber of the clutch control mechanism is represented at channel №3 and a signal from the microswitch of the clutch mechanism is represented at channel №4. It is important to note that in the developed automatic transmissions the gearshift is always performed with the automatic clutch control (channels 2-4, Figure-9).

The results of road test

After obtaining a satisfactory result at the test stand the sixteen speed transmission TM 16-2000 was installed on the vehicle KAMAZ-65117 (6x4) (Figure-1) (Blokhin *et al*., 2015 c, 2015 d, 2015 f, 2015 g). Its main technical parameters are presented in Table-1.

Figures 10 and 11 show the waveforms, matching several startings of the vehicle KAMAZ-65117, having TM16-2000 transmission with automatic control on a flat supporting surface.

On channel № 1 (Figure-10) we can see the value of the pressure in the power chamber of the automatic clutch control mechanism. The maximum value of pressure is about 8 atm. The pressure at which the regulation is provided by the clutch position corresponds to 2-2.2 bar. On channel № 2– we can see the value of the pressure in the accelerator mechanism of the clutch control. Maximum pressure is a bit more than 8 atm., the pressure in the valves, when the clutch position is being adjusted is 2.6...2.8 atm.

The movement of the clutch fork rod represents channel № 3. At the time of clutch engagement for gentle start of the vehicle, the friction clutch plates differ by the amount of 70-80% of their maximum speed. Channel № 4 presents data on the response time of the microswitch of the clutch based on the analysis which will determine the time of the on / off of the clutch and the total time of a gear change (on average about 0.7 seconds.)

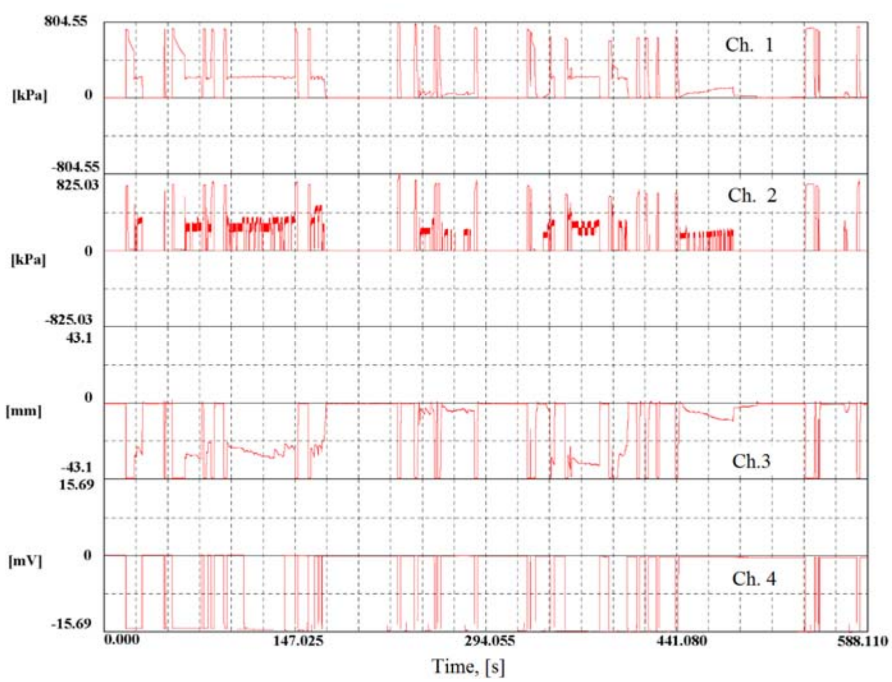

Figure-10. Waveform of starting process of KAMAZ-65117 (6x4) with TM16-2000 transmission Channel №1 - The pressure in the power chamber of the automatic clutch control mechanism; Channel №2 - the pressure in the accelerator of the clutch control mechanism; Channel №3 - the movement of the clutch fork; Channel №4 - response time of the clutch mechanism.

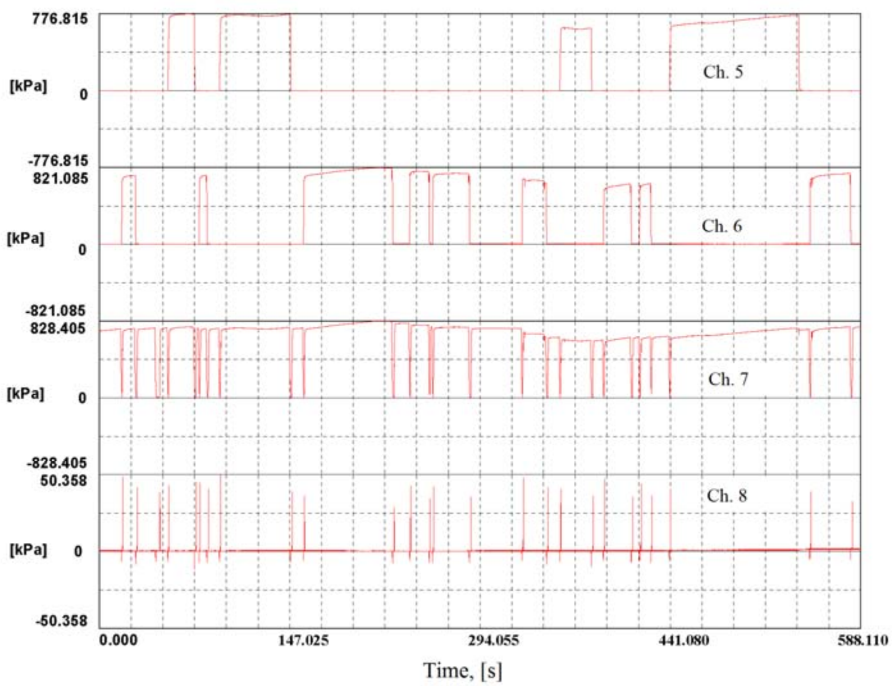

Figure-11. Waveform of starting process of KAMAZ 65117 (6x4) with TM16-2000 transmission
Channel №5 - The pressure in the power chamber of the main gear at turning on even gears;
Channel №6 - The pressure in power chamber of the main gear at turning on even gears;
Channel №7 - The pressure in the power chamber at the lower range of the splitter;
Channel №8 - The pressure in the power chamber at the top range of the splitter.

Figure-11 shows the nature of the pressure measurement in the power of the main gearbox and cavities divider. The movement took place on 1 and 2 in the main transmission gearbox (respectively channels №5 and №6, Figure-10) and the lower divider range (channel №7 Figure-11).

Analysis of the results of the road research showed that at ambient temperatures above 0C the difference is 5... 10% max due to a more rapid decrease in pressure in the vehicle's receiver by several gear changes. At low outside temperatures, from -15C .to -20C, the data of the laboratory and road conditions differ significantly, up to 35%. To some extent, this is due to additional freezing of the pipes when the vehicle is moving, which is difficult to simulate in our laboratory.

Thus, in general, there is a high qualitative and quantitative convergence with the results of laboratory tests on the stand, and the results of theoretical research.

CONCLUSIONS

a) It considers the basic design features of the unified technical multistage gearboxes based on the leading technology solutions. We have found out that considering the number of gears of synchronized stages, transmitted maximum torque, power, range, density, number of gear ratios, and mass-dimensional parameters the gearboxes from the standard series rank with the best world analogues.
b) The paper contains a schematic diagram of automatic or command control of the developed multi-staged manual transmissions with microprocessor control.
c) It represents the results of experimental studies on the specialized stand, as well as comparison of theoretical and experimental data. The discrepancy between the data is max. 10-25%.
d) In general, there is a high qualitative and quantitative convergence of the road test results with the results of laboratory tests carried out on a special stand, and the results of theoretical research. At ambient temperatures above 0C the difference is 5 ... 10% max, at lower temperatures, from -15 C .to. -20 C, the difference increases to 35%.

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Nomenclature

U_i	gear ratio	
n'_e	lowest engine speed, within which the acceleration at the gear takes place	[r/min]
n''_e	highest engine speed, within which the acceleration at the gear takes place	[r/min]
v_0	vehicle speed before gear shifting	[m/s]
v	vehicle speed after gear shifting	[m/s]
q	progression denominator	
λ_h	actual rpm interval in higher gear	
λ_l	actual rpm interval in lower gear	
g	gravitational acceleration	[m/s^2]
G	the full weight of the vehicle	[N]
ξ	the coefficient taking into account the inertia of the rotating masses of the transmission and the chassis parts	[N·s^2/м^4]
k	wind drag coefficient	
Fm	Middel's area	[m^2]
Ψ	road resistance coefficient	
x	the rod movement of the considered nod (the power cylinder rod)	[Mm]
p_p	the pressure in the receiver, from which the air outflows	[Pa]
p	the current pressure in the power cylinder	[Pa]
V	the volume of the cylinder into which the air outflows	[m^3]
T	absolute temperature of the air	°[K]
R	the gas constant	[J/ mol K]
k	heat capacity ratio for the air	
μ	discharge coefficient	
F	the jet section, through which the air outflows	[m^2]
Mc	the friction torque in the transmission synchronizer	[Nm]
M_{TP}	the friction torque of the leading parts of the box given to the selector gear caused by the losses due to oil splashes and friction in the bearings are determined experimentally	[Nm]
$M_{B.C}$	motor torque transmitted to the driven clutch parts when it is deactivated (due to imperfections of the structure)	[Nm]
M_M	the friction torque of the fork against the synchronizer clutch	[Nm]
$M\psi$	the moment of resistance of the vehicle, given to the output shaft of the transmission.	[Nm]

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