THE REPUBLIC OF BELARUS
MINISTRY OF EDUCATION
BELARUSIAN STATE POLYTECHIC ACADEMY

UDC 629. 113-598

Lepeshko I.I., Hassan Abdulkadir Baba

CONCERNING THE CALCULATION OF ACTIVE HYDRAULIC BRAKE

Minsk 1993

I, Mikhail S. Shpakouski, sworn translator, do hereby certify that I am fluent in the English, Russian and Belarusian languages, and that the above document is an accurate and complete translation of the document entitled "Diploma of Higher Education" made this 16th day of February 2012.

Pr. Masherova 1-134, Minsk 220005 Belarus
Tel/fax: (375 17) 202-11-11, e-mail: 2021111@lan.by
Seal: Lantrading Plus Translation Services

БЮРО ПЕРЕВОДОВ
ЛАНТРЕЙДИНГ ПЛЮС

Nowadays in the automotive industry the hydraulic brakes of different schemes gain more and more popularity. All the class of such brake gears can be divided into two main groups, namely:

1. Brakes with parallel action, that are represented by direct action brake systems (hydraulic, as a rule) with a hydraulic booster. The main peculiarity of its work is the creation of control pressure in the system by driver's muscle power and the following hydraulic boosting of this pressure by a pump station. If the pump station doesn't work or there is no energy reserve than the brake system becomes a direct drive system.

2. Indirect action brake systems imply the creation of the force affecting the brake pads by the pump station pressure, at the same time the role of the driver is limited to the operation of pressure regulator (brake valve). If there is no pressure than such brake system is inoperable.

The analysis of the parameters of the abovementioned brake systems is usually made for static and dynamic modes of operation /1, 2/, at the same time the method of power take-off is not taken into account.

In order to calculate the influence of the method of power take-off on the selection of parameters and the analysis of the brake system let's see the possible variants of the pump station actuation.

a) Pump station actuation from vehicle engine.

Such method is widespread in motor vehicles and requires as obligatory components a pump, a hydraulic accumulator and pump station automatic control system, providing the necessary pressure thresholds in the hydraulic accumulator. Such a set has the following goals:

- decreasing of the generating capacity of the pump to the certain limits with the purpose of the reduction of power expenditure for brake system operation;
- increasing of brake system response at the account of significant instantaneous fluid consumption during the period of backlash elimination between brake pads and brake drum;
- guarantee of brake system operation for a certain period of time in case of engine stop.

During statistical and dynamic analysis of such brake system usually consider that the pressure in the hydraulic accumulator during braking stays permanent and the fluid consumption is determined by the resistance of lines and control equipment /2/.

I, Mikhail S. Shpakouski, sworn translator, do hereby certify that I am fluent in the English, Russian and Belarusian languages, and that the above document is an accurate and complete translation of the document entitled "Diploma of Higher Education" made this 16th day of February 2012.

Pr. Masherova 1-134, Minsk 220005 Belarus
Tel/fax: (375 17) 202-11-11, e-mail: 2021111@lan.by
Seal: Lantrading Plus Translation Services

БЮРО ПЕРЕВОДОВ
ЛАНТРЕЙДИНГПЛЮС
www.lan.by

b) Pump station actuation from electric engine fed by vehicle-mounted electric system. Such drive has no significant differences in comparison with the drive from vehicle engine with the exclusion of the issue of the increasing of energy reserve in case of engine stop. This reserve is determined by the capacity of the hydraulic accumulator and capacity of vehicle-mounted electric battery.

The calculation of brake system parameters and pump system parameters for such drive is identical to the drive with a pump station activated by an internal combustion engine (main engine of a motor vehicle).

c) The drive from vehicle's wheel or transmission. This drive has several peculiarities that influence the structure and component parts of such brake system on the whole and lead to the necessity of introduction of statistical and dynamical modes of operation into analysis method.

The main peculiarity of the brake system is the absence of pump generating capacity limit. This is conditioned, first of all, by the fact that the power consumed by the pump station facilitates the braking of a motor vehicle, i.e. it creates an additional braking torque. On the other hand, there is an almost unlimited energy reserve concentrated in the vehicle weight in the form of movement kinetic energy. The combination of the specified peculiarities of such drive allows rejection of hydraulic accumulators and systems of their automatic control; the necessary response of the brake system can be reached by the selection of parameters and place for the pump installation in respect to the wheels (a wheel) of a motor vehicle.

The main disadvantage of the abovementioned brake system is the presence of a "creeping" speed that is conditioned by the inevitable leakage of working fluid through the elements of the control equipment and fluid flow in the pump itself. It should be noticed that this phenomenon is a system's disadvantage and its positive property simultaneously.

The disadvantage of this phenomenon is the necessity of the usage of additional equipment in order to provide the holding of the motor vehicle at rest.

I, Mikhail S. Shpakouski, sworn translator, do hereby certify that I am fluent in the English, Russian and Belarusian languages, and that the above document is an accurate and complete translation of the document entitled "Diploma of Higher Education" made this 16th day of February 2012.

Pr. Masherova 1-134, Minsk 220005 Belarus
Tel/fax: (375 17) 202-11-11, e-mail: 2021111@lan.by
Seal: Lantrading Plus Translation Services

БЮРО ПЕРЕВОДОВ
ЛАНТРЕЙДИНГ ПЛЮС
www.lan.by

Among the disadvantages of such phenomenon there is the exclusion of the blocking of the braking wheel in the process of movement thus increasing its stability and safety.

In order to specify the main peculiarities of the calculation of the brake system with pump station actuation from a vehicle wheel let's see its structural scheme the fragment of which is provided on Fig. 1.

The active hydraulic brake gear consists of a hydraulic pump with a safety valve 2, a tracer valve 3, controlled by a remote direct action hydraulic drive with a main cylinder 4. The hydraulic pump is connected to a vehicle wheel through a matching gearbox 5.

During the vehicle movement and pressing the pedal 7 the tracer valve become activated and the fluid from the hydraulic pump gets to wheel hydraulic cylinders.

At the low speed of vehicle movement the tracer valve ceases its activity due to the insufficiency of fluid consumption; due to control effort the fluid flow through the pressure valve is stopped, at the same time the pump force main will become fully connected to wheel hydraulic cylinders. At the same time the pressure in hydraulic cylinders will be determined by fluid leakage through the valve gaps and leakages in the pump itself. Thus the balance between the vehicle speed and fluid pressure in the brake system will be achieved. If at this time the motor vehicle is located on a horizontal surface such slowing down will be determined by hydraulic drive parameters and road resistance. In the case of bearing surface slope the permanent speed of movement will be established.

Let's make an equation for the movement of the motor vehicle taking into account the braking action of the pump station in order to determine the regular patterns in the change of the motor vehicle movement from the movement of the ceasing of the of valve's tracer activity. During the making of the equation for the movement we will not take into account the air resistance as the movement speed is very low and the coefficient of efficiency of the matching gearbox we take equal to one.

The equation for the movement of the motor vehicle taking into account the made assumptions

$$m_a \frac{dV_a}{dt} + \sum_1^n J_k \frac{d\omega_k}{dt} \cdot \frac{1}{z_g} = \frac{M_H U_p}{z_g} + m_a \cdot g \cdot \psi + \sum_1^n p K_i \frac{1}{z_g}, \qquad (1)$$

I, Mikhail S. Shpakouski, sworn translator, do hereby certify that I am fluent in the English, Russian and Belarusian languages, and that the above document is an accurate and complete translation of the document entitled "Diploma of Higher Education" made this 16th day of February 2012.

Pr. Masherova 1-134, Minsk 220005 Belarus
Tel/fax: (375 17) 202-11-11, e-mail: 2021111@lan.by
Seal: Lantrading Plus Translation Services

where

m_a - vehicle weight;

$\frac{dV_a}{dt}$ - the deacceleration of a motor vehicle;

J_k - the moment of inertia of a vehicle wheel;

n - the number of wheels;

$\frac{d\omega_k}{dt}$ - the angular deacceleration of a vehicle wheel;

$r_д$ - wheel's dynamic radius;

M_H - the braking torque of the hydraulic pump;

U_p - the ratio of the matching gearbox;

g - the acceleration of gravity;

ψ - the cumulative rate of road resistance;

P - the pressure in the hydraulic brake system;

K_i - the conversion ratio of the brake mechanism.

The pressure in the hydraulic brake system depends on the fluid consumption and throttle resistance. We suppose that the fluid throttling in the pump and valve are collected by one hypothetic throttle and the pump is absolutely leak-proof, i.e. there are no leakages in the pump:

$$Q = \mu f \sqrt{\frac{2p}{\rho}}, \quad (2)$$

where

Q - fluid consumption through a hypothetic throttle;

f - conditional orifice size;

μ - consumption ratio;

ρ - the density of the working fluid.

On the other hand, the fluid consumption is determined by the frequency of pump rotation, its size and ratio of the matching gearbox located between the pump and a wheel:

$$Q = q\, n_k\, U_p \quad (3)$$

where

q - pump volume, m^3/vol.

n_k - the frequency of wheel rotation, rpm/s

By solving the equations (2) and (3) we will determine the dependency of the pressure in brake system hydraulic drives from the speed of vehicle movement:

$$P = \frac{q^2 U_p^2 \rho \cdot \omega_k^2}{8\pi^2 \mu^2 f^2}, \quad (4)$$

where

$\omega_k = 2\pi n_k$ - the angular speed of wheel rotation.

The braking torque of the hydraulic pump shall be determined by the equation of hydraulic and mechanical power:

I, Mikhail S. Shpakouski, sworn translator, do hereby certify that I am fluent in the English, Russian and Belarusian languages, and that the above document is an accurate and complete translation of the document entitled "Diploma of Higher Education" made this 16th day of February 2012.

Pr. Masherova 1-134, Minsk 220005 Belarus
Tel/fax: (375 17) 202-11-11, e-mail: 2021111@lan.by
Seal: Lantrading Plus Translation Services

$$pQ = M_н \omega_к \qquad (5)$$

where

$M_н$ - the braking torque created by the pump.

After the translation of formula (5) we will obtain:

$$M_н = \rho \frac{q^3 U_p^2}{16\pi^2 \mu^2 f^2} \cdot \omega_к^2 \qquad (6)$$

By inserting received formulas (4) and (6) in to formula (2) for the vehicle movement and making the translations with an assumption that the rolling radius is equal to the dynamic radius, we will obtain:

$$\left(1+\frac{\Sigma J_к}{m_a}\cdot\frac{1}{r_g^2}\right)\frac{dV_a}{dt} = g\psi + \frac{\rho}{m_a}\cdot\frac{q^3 U_p^3 + 2\Sigma q^2 U_p^2 K_i \pi}{16\pi^3 \mu^3 f^3 r_g^3} V_a^2 \qquad (7)$$

The canonical form of equation (7) is:

$$\frac{dV_a}{dt} = a^2 + b^2 V_a^2 \qquad (8)$$

Where

$$a = \sqrt{\frac{g\psi}{1+\frac{\Sigma J_к}{m_a r_g^2}}},$$

$$b = \sqrt{\frac{\rho}{m_a}\cdot\frac{q^3 U_p^3 + 2\Sigma q^2 U_p^2 K_i \pi}{16\pi^3 \mu^2 f^2 r_g^3 \left(1+\frac{\Sigma J_к}{m_a r_g^2}\right)}}$$

After the separation of variables, integration and translations we will obtain:

$$V = \frac{a}{b} tg\left[arc\,tg \frac{b}{a} V_0 - abt\right], \qquad (9)$$

where

V_0 - the initial speed of the motor vehicle when the ceasing of the tracer action of the valve takes place.

According to formula (9) the main parameters determining the nature of the vehicle speed change during the braking in the mode of absence of the tracer action of the valve shall be the ratio of the matching gearbox and the pump size. It is rational to make the calculation of these parameters based on the stated response of the brake system as a main factor with an influence on the braking length of the motor vehicle.

I, Mikhail S. Shpakouski, sworn translator, do hereby certify that I am fluent in the English, Russian and Belarusian languages, and that the above document is an accurate and complete translation of the document entitled "Diploma of Higher Education" made this 16th day of February 2012.

Pr. Masherova 1-134, Minsk 220005 Belarus
Tel/fax: (375 17) 202-11-11, e-mail: 2021111@lan.by
Seal: Lantrading Plus Translation Services

БЮРО ПЕРЕВОДОВ ЛАНТРЕЙДИНГ ПЛЮС www.lan.by

In order to determine the response of the brake system we suppose that the tracer valve opens immediately and all the fluid from the pump goes into the brake system hydraulic cylinders. In this case the volume coefficient is equal to one as the filling of the hydraulic cylinders is made on the condition of the low pressure.

The calculation of parameters shall be made based on the equivalence of the fluid consumption and volume of wheel hydraulic cylinders:

$$q\, n_{к}\, u_{р} = \frac{\Theta_i\, z}{\Delta t}, \qquad (10)$$

where

Θ_i - the volume of fluid necessary for the filling of hydraulic cylinders taking into account the moving of pistons to backlash elimination between the brake pads and brake drums or discs;

Δt - the time for filling of the hydraulic cylinders;

z - the number of moving pistons.

Having expressed the volume of the hydraulic cylinder through its geometric parameters and piston stroke

$$q\, n_{к}\, u_{р} = \frac{\pi\, d_{ц}^2\, z\, h}{4\Delta t}, \qquad (11)$$

where

$d_{ц}$ - the diameter of the hydraulic cylinder piston;

h - the moving of the hydraulic cylinder piston.

Having expressed $n_{к}$ in equation (II) through the vehicle speed and calculated it in respect to time we will obtain:

$$\Delta t = \frac{r_{к}\, \pi^2 d_{ц}^2\, z\, h}{2\, q\, u_{р}\, V_a} \qquad (12)$$

From the obtained equation you can see that the response of the brake system depends on the vehicle speed, the volume of hydraulic cylinders, the gap between the brake pads and brake drums and the pump size.

It shall be noted that the size of the hydraulic cylinders is determined based on the calculation of the brake mechanisms in respect to the necessary control effort and maximum pressure of the working fluid, limited by the selected type of the pump and abilities of the brake system flexible brake hoses.

I, Mikhail S. Shpakouski, sworn translator, do hereby certify that I am fluent in the English, Russian and Belarusian languages, and that the above document is an accurate and complete translation of the document entitled "Diploma of Higher Education" made this 16th day of February 2012.

Pr. Masherova 1-134, Minsk 220005 Belarus
Tel/fax: (375 17) 202-11-11, e-mail: 2021111@lan.by
Seal: Lantrading Plus Translation Services

The ratio of the matching gearbox is determined based on the structural peculiarities of the pump taken for the installation into the brake system. The selection of the pump is determined by its price and operation parameters.

In all cases, independently from the mentioned criteria for the selection of the pump, it shall be noted that any structure of the pump is limited by the maximum speed of its rotation. Based on this the necessary ratio of the matching gearbox is determined based on the kinematic relations between the maximum speed of vehicle movement and maximum rotation speed of the pump shaft, i.e.

$$U_p = \frac{\omega_H^{max}}{V_a^{max}} r_k, \qquad (13)$$

where

ω_H^{max} - the maximum speed of the pump shaft rotation;

V_a^{max} - the maximum speed of the vehicle movement.

Next, the choice of parameters is amounted to the response at the stated minimum speed of vehicle movement. Having solved equation (12) in respect to (q) we will get:

$$q = \frac{\pi^2 d_u^2 Z h}{\Delta t \cdot 2\omega_H^{max} \frac{V_a^{min}}{V_a^{max}}}, \qquad (14)$$

where

V_a^{min} - the minimum speed of vehicle movement.

The value of the pump size obtained in equation (14) forms the basis for its design or the utilization of one of the existing types. When using an off-the-shelf pump different from the calculated one in respect to its size, the necessary alliance of properties is performed by the adjustment of the ratio of the matching gearbox.

It shall be noted that during the assignment of the required response of the brake system it is necessary to take into account the actuation time of the brake system that is determined by the sum of the actuation time of the valve control drive and the actuation time of the drive itself, i.e.

$$\tau = \Delta t_e + \Delta t, \qquad (15)$$

where

τ - the actuation time of the brake system;

Δt_e - the actuation time of the tracer valve control drive.

I, Mikhail S. Shpakouski, sworn translator, do hereby certify that I am fluent in the English, Russian and Belarusian languages, and that the above document is an accurate and complete translation of the document entitled "Diploma of Higher Education" made this 16th day of February 2012.

Pr. Masherova 1-134, Minsk 220005 Belarus
Tel/fax: (375 17) 202-11-11, e-mail: 2021111@lan.by
Seal: Lantrading Plus Translation Services

БЮРО ПЕРЕВОДОВ ЛАНТРЕЙДИНГ ПЛЮС

UDC 629. 113-598

To the calculation of active hydraulic brake/ Lepeshko I.I., Hassan Abdulkadir Baba; Belarusian State Polytechnic Academy. – Minsk, 1993. -12 p. - : 3 figures – Bibliography: 2 names – In Russian.

Stored in Belarusian Scientific Research Institute for Scientific Information and Technical and Economic Research.

The analysis of active hydraulic brake systems with various methods of power take-off for the pump station was performed. It was demonstrated that the pump station actuation from the vehicle wheels expands the area for the application of such drives with the simultaneous simplification of brake system structure.

The methodology of calculation of the main parameters of the brake drive maintaining the main values of efficiency and response was provided.

The rules for the change of speed and relative wheel slipping in case of emergency braking were demonstrated.

Authors: */signed/ /Lepeshko/*
/signed/ Hassan A.B.

I, Mikhail S. Shpakouski, sworn translator, do hereby certify that I am fluent in the English, Russian and Belarusian languages, and that the above document is an accurate and complete translation of the document entitled "Diploma of Higher Education" made this 16th day of February 2012.

Pr. Masherova 1-134, Minsk 220005 Belarus
Tel/fax: (375 17) 202-11-11, e-mail: 2021111@lan.by
Seal: Lantrading Plus Translation Services

БЮРО ПЕРЕВОДОВ
ЛАНТРЕЙДИНГ ПЛЮС
www.lan.by

The obtained equation determines the change of the vehicle speed on the condition of the realization of the adhesion coefficient to the bearing surface. The diagram of the mentioned dependence is provided in Fig. 3. By comparison of the diagrams of the change of the vehicle speed due to the actuation of the active brake drive and threshold value according to the condition of the realization of the adhesion coefficient to the bearing surface it can be concluded that at the identical initial speeds there is a significant slipping of the braking wheel in regard to the bearing surface. In order to escape such a phenomenon it is necessary to apply special devices regulating the throttle resistance in the hydraulic drive.

Based on the performed analytical analysis we can make the following conclusions:

1. By using the active hydraulic brake drive with the pump station actuation by a vehicle wheel there is a possibility to satisfy the requirements to the response of the brake system without using of the hydraulic accumulator by the means of the corresponding selection of the pump station parameters and matching gearbox;

2. There is a phase of non-controlled braking in case of the active hydraulic drive conditioned by the pump station parameters and the tracer valve;

3. In order to eliminate the significant slipping of a wheel in respect to the bearing surface the special regulating devices shall be introduced into the brake system;

4. In order to hold the vehicle immovable the brake system shall include the possibility of direct action on the brake mechanisms by the driver.

Literature

1. Motor vehicles. Structure, design and calculation. Control systems and chassis. Under the editorship of A.I. Grishkevich. Minsk, High school, 1987. – 200 p.

2. Metliuk N.F., Avtushko V.P. Dynamics of pneumatic and hydraulic drives of motor vehicles. – Moscow, Machine industry, 1980. – 231 p.

I, Mikhail S. Shpakouski, sworn translator, do hereby certify that I am fluent in the English, Russian and Belarusian languages, and that the above document is an accurate and complete translation of the document entitled "Diploma of Higher Education" made this 16th day of February 2012.

Pr. Masherova 1-134, Minsk 220005 Belarus
Tel/fax: (375 17) 202-11-11, e-mail: 2021111@lan.by
Seal: Lantrading Plus Translation Services

After the achievement of the drive pressure determining the corresponding value of the initial speed of non-controlled movement according to the diagram provided in Fig. 2, the resulting braking action will be performed in conformity with the rules established by movement equation (1), at the same time the vehicle speed will change in accordance with equation (8) (see the diagram in Fig. 3).

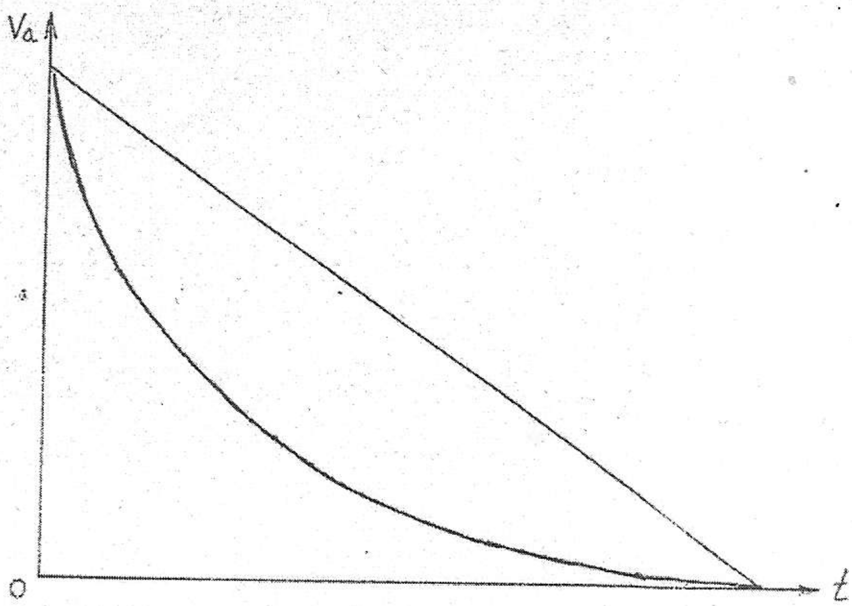

Fig. 3. The change of non-controlled movement speed.

It shall be noted that the movement equation and the obtained diagrams reflect the process of the vehicle movement without limitation to the condition of wheel's adhesion to the bearing surface.

Taking into account the limitation to the condition of wheel's adhesion to the bearing surface equation (1) gains the following look:

$$m_a \frac{dV_a}{dt} = m_a g f + m_a g \varphi, \qquad (16)$$

where

φ - the wheel's adhesion coefficient to the bearing surface;

f - the coefficient of resistance to the rolling movement.

Having made the translations and solved the equation in respect to the vehicle movement speed we will get

$$V_a = V_0 - (\varphi + f) g \cdot t. \qquad (17)$$

I, Mikhail S. Shpakouski, sworn translator, do hereby certify that I am fluent in the English, Russian and Belarusian languages, and that the above document is an accurate and complete translation of the document entitled "Diploma of Higher Education" made this 16th day of February 2012.

Pr. Masherova 1-134, Minsk 220005 Belarus
Tel/fax: (375 17) 202-11-11, e-mail: 2021111@lan.by
Seal: Lantrading Plus Translation Services

In the process of deacceleration control the driver shall press the brake pedal taking into account the present situation, at the same time in the required pressure is created in the tracer valve control system. The controlled tracer valve will maintain the stated pressure in the brake system by the means of throttling of the delivery section of the pump station towards discharge until the moment of disbalance between the control pressure and the pressure of the brake drive conditioned by the insufficient productivity of the pump station due to the reduction of the vehicle speed. Next, the brake process will be realized in accordance with the rule established by formula (9), at the same time the initial speed will depend on the pressure on the control pedal.

In order to carry out a qualitative analysis of the formation of the initial speed of the actuation of non-controlled braking let's examine the diagram of its change built in accordance with equation (4) and provided in Fig. 2.

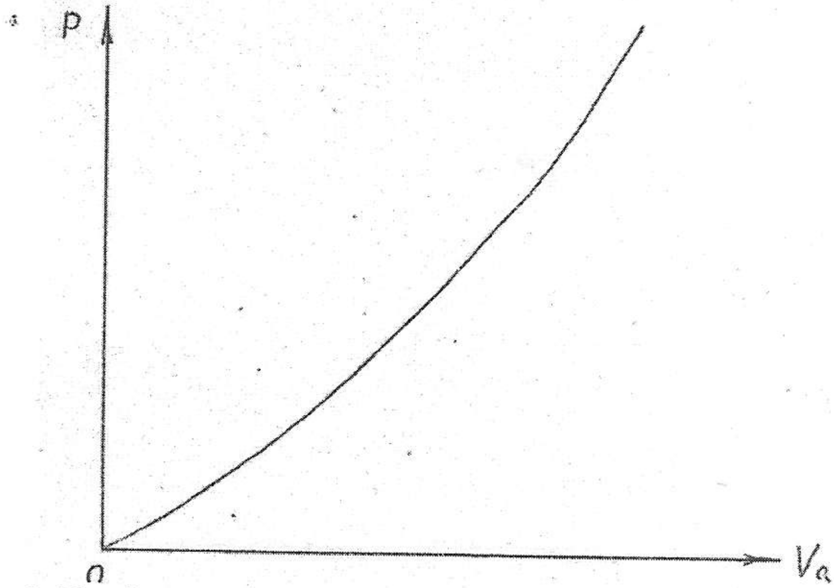

Fig. 2. The diagram of terminal pressures in the system of the active hydraulic brake drive.

From the diagram above we can see that the driver can facilitate the braking action for any speeds located to the right from the diagram of terminal pressures. The deacceleration of the vehicle with the hydraulic drive pressure and speeds located to the left of the diagram of terminal pressures is impossible.

I, Mikhail S. Shpakouski, sworn translator, do hereby certify that I am fluent in the English, Russian and Belarusian languages, and that the above document is an accurate and complete translation of the document entitled "Diploma of Higher Education" made this 16th day of February 2012.

Pr. Masherova 1-134, Minsk 220005 Belarus
Tel/fax: (375 17) 202-11-11, e-mail: 2021111@lan.by
Seal: Lantrading Plus Translation Services