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Mathematical Description of the Machine-Tractor Aggregate Diesel Engine with Electronic Positional Impact Regulator

This article presents a mathematical description of the diesel engine of the machine-tractor unit with a developed electronic regulator of the positional action on the fuel pump rack. The developed mathematical model of a diesel engine with a modernized electronic regulator using an additional effect from the crankshaft twisting value makes it possible to determine the limits of fuel feed regulation at unsteady operating modes for engines of various power capacities and designs. The principle of operation of the advanced electronic positional impact regulator is described and its functional block diagram is given.

Keywords: *diesel engine; fuel supply system; the electronic regulator; torque; mathematical model of a diesel engine.*

1. INTRODUCTION

It is possible to implement intensive operating pro– cedures effectively in the agro-industrial complex and address the problem of import substitution only if modern requirements are met: the machinery must be fuel- and energy-efficient, reliable, high-precision and less material-intensive.

Diesel engines having a conventional split-pumping fuel feeding system with a mechanical regulator which has become widely used in agricultural combine harvesters and tractors due to high reliability in operation and ease of maintenance.

Many engine manufacturing facilities produce diesel engines that are fully or partially automated, depending on the requirements. With the increase in the degree of automation, very high requirements are imposed on the regulators to ensure fuel efficiency in a wide range of high-speed and load-bearing modes of operation of the diesel engine.

However, unfortunately, a significant part of the diesel engines have insufficiently high performance in terms of fuel efficiency in partial and unsteady operating modes [1,2].

Surging of the crankshaft, and, as a consequence of the camshafts and regulator shafts, as well as the travel of the high-pressure fuel pump rack, caused by the uneven loads on the machine-tractor aggregate, are the reason for the deterioration of the power and economic parameters of the diesel engines.

Further increase in the efficiency of diesel engines with the direct fuel feeding system is possible by improving the fuel systems, taking into account their operation in unsteady modes for considerable time, as well as in modes of partial loads and low rotating velocities of the crankshaft. This can be achieved, first of all, by reducing the inertia of the diesel-fuel pump regulator system [3,4].

Modern microprocessor technologies enable to fully implement regulation of the fuel feeding system and the engine in automatic control systems, with regard to the load variation as the main disturbance input [5,6].

The fuel feed systems of direct action with electronic control are divided into systems with discrete and positional input. [3,5]

In the systems with a discrete input, the output pulsed signal determines directly the timing of the start and end of the fuel injection. The R. Bosch electromagnetic actuator is used in in-line fuel pumps of MV and P type [7]. It is a precision linear electromagnet mounted coaxially with the metering rack of the high pressure fuel injection pump (HPFIP). The rack is moved by a return spring toward the fuel feed reduction. The required cyclic feeding characteristics are formed in the electronic block of the automated control system on the basis of signals coming from the crankshaft rotation sensors, the position of the HPFIP rack and the regulator control lever.

A similar microprocessor control system was developed by Lucas CAV (UK) and Central Research and Design Institute of Fuel Equipment for Automotive and Stationary Engines (St. Petersburg, Russia) for distribution fuel injection pumps.

A number of companies note a significant simplification of the fuel pump when implementing an electronic injection control system; the Lucas CAV distribution pump (similar to the DRS pump) with electronically controlled fuel feed and an injection timing angle has 100 parts instead of 220 for the DRS pump.

Even more simplified design of the fuel pump and reduced cost can be achieved by using the same solenoid valve to control both the filling of the plunger space and the amount of plunger delivery to the injector.

Ford Motor used this control method in the SME-4 system of the four-cylinder PROCO diesel engine. The

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company notes that with this system, the toxicity of the engine exhaust gases was the same as with the conventional system. The unevenness of the fuel feed has decreased significantly.

The systems with electromagnetic actuators located in the injection line between the HPFIP and the injector have become widely known. Such a device was used, for example, by the Japanese company Nippo Denso.

Electronically controlled systems with positional impact on the fuel feed control (high pressure rack) are commonly known as systems with electronic speed controllers.

Speed controllers manufactured by ADECO, Robert Bosch GmbH and others are recognized by the consumer; electromagnetic actuators of such controllers can provide a movement of the HPFIP rack for 12 ... 10 mm over 15 msec and returns to the former position within 100 msec.

The principle of regulating multiplunger pumps by a shut-off clutch was applied by Caterpillar, Diesel Kiki, Bosch and others. It enabled not only to eliminate the oblique helixes on the plunger, but also to significantly improve the accuracy of fuel dosing by the pump sections and improve the fuel feed process.

When controlling the fuel feed with the oblique plunger helix and the rack-and-pinion mechanism, the cut-off cut off bushing can be used to influence the injection advance (using electronic control).

At the same time, the camshaft drive stiffness is significantly increased (a flexible element of the advance coupling is eliminated) and a stable value of the advance angle is provided with high accuracy and speed response (more than 160 degrees/s).

The cut-off bushing for adjusting the advance angle is usually introduced in fuel pumps with a high level of injection energy. Its placement increases the height of the pump casing by 17...24 mm.

Bosch used an electromagnetic mechanism to control the rack and the cut-off bushing.

The actuator mechanisms are controlled by means of an electronic unit made on the basis of microprocessor technology (according to the signals of a number of sensors), as in the above-mentioned cases.

The accumulator injection systems such as Common Rail of Daimler Benz, Caterpillar, and others meet high requirements for fuel feeding systems of diesel engines, combined with electronically controlled injection process. However, these systems are characterized by very high costs and are not always reliable in the machine and tractor aggregates under conditions of actual operation; they are very sensitive to the quality of fuel and to working conditions in agriculture.

Establishing the relationship between the load applied to the engine and the amount of fuel supplied into the combustion chamber is the most difficult task so far, which has not got a technical solution providing a large-scale production due to the extremely complex nature of the load changes, particularly in diesel machine and tractor aggregates.

In practice, this problem is solved by increasing the torque backup, increasing the rated speed of the crankshaft or reducing the traction resistance of the aggregate. Unfortunately, all these factors contribute to an increase in fuel consumption, especially in the partial engine load modes.

The fuel feed regulation depending on the amount of load or torque on the engine crankshaft is a well-known and most effective method. It is better known as the N-regulator or disturbance input regulator. In modern technology, this method is not widely used, and it has not found application in the control systems for diesel engines.

Recently, leading engine manufacturers have begun to carry out research aimed at the implementation of this method. In the research centers of the General Motors Company, experimental work is under way to fine-tune the torque flexplate system and to create a torque control system for engines with automatic transmission. They used Transense/HoneyWell Wireless SAW Sensors to measure the load on the engine flywheel. The possibility of non-contact measurement and signal transmission is a distinctive feature of sensors of this type [8,9].

At the same time, it should be noted that the possibilities of adapting the currently used fuel feed systems of direct action that are very reliable in operation for efficient operation of diesel engines in partial and unsteady loads by electronic regulation are far from implementation.

Based on numerous studies of unsteady modes, the authors have developed a dynamic model of the transient process for the operation of diesel engines of the machine-tractor aggregate (MTA), which made it possible to identify the interrelationship between the design and operational parameters of the diesel-regulator system, and also to determine the factors whose effect can lead to an increase in the efficiency of the fuel feeding systems [2,4,10,11].

The aim and objective of this study is to improve the quality of controlling fuel feeding parameters of the MTA diesel engine by positional increased-precision electronic regulation. The main task is to improve the devices of the fuel feed control system, allowing for automatic movement of the fuel feed control unit with precise positioning and timely monitoring of load changes depending on the operating modes.

2. METHODOLOGY

Mathematical modeling was based on the structural and functional analysis of the transient processes of a diesel engine as a dynamic system consisting of mechanical and automatic control subsystems and on the theory of automatic control of engine crankshaft speed.

The authors developed an electronic device for positional impact on the fuel feed controls of a diesel engine with an additional load action applied to the D-243 four-cylinder tractor diesel engine, which made it possible to reduce the rotation overspeed and the duration of the transient process of the diesel engine in unsteady operating modes [11]

However, this developed device with additional load action did not take into account the input value of the perturbation of the torque oscillations, thereby reducing the efficiency of electronic position control operation.

Increasing the diesel engine efficiency can be provided taking into account all the factors affecting the fuel feed, in particular the crankshaft twist angle and its use as an input signal in the automatic control system [11].

In this regard, the authors developed a mathematical model of a multi-cylinder diesel engine that is load regulated depending on the crankshaft twist angle.

The simplest technological solution can be given by determining the torque directly on the crankshaft. This enables to exclude a lot of negative impacts that affect the quality of the received signal. The existing theories and developed mathematical models do not take into account the possibility of regulating the shaft torque, and moreover they do not take into account the torque variation non-uniformity along the various crankpins.

The diesel engine is a controlled object of the system; air supply G_a and fuel injection g_c are its input actions, and angular velocity of the engine crankshaft ω_D and the developed torque M_e are its output actions. The engine is controlled by the regulator whose input action includes the angular velocity ω_D and the relative variation of torque $\Delta M = M_e - M_R$, and the position of the fuel equipment control h_{fc} is its output parameter.

Measuring the relative variation of torque is rather difficult to implement in practice. It is technically easier to determine the value of crankshaft torquability.

Representing the crankshaft as a balanced solid rod (figure 1), which is twisted by the z-th number of torques (equal to the number of cylinders) distributed along its length, let us write down the equation of its twist angle in the boundaries of elastic deformations, determined from the torsion formula of Hooke's law:

$$\gamma_z = \frac{M_{fz} l_z}{J_0 G}, \tag{1}$$

where J_0 is geometric polar moment of inertia;
l_z – distance between the flywheel and the axis of the z-th cylinder;
G – shear modulus;
M_{fz} - z-th cylinder torque.

In accordance with the above equation the crankshaft twist angle will be equal to:

$$\gamma = \gamma_1 + \gamma_2 + ... + \gamma_z = \sum_1^z \gamma_z = \frac{\sum_1^z M_{fz} l_z}{J_0 G} \tag{2}$$

Considering that each torque equals to $M_{f1} = M_{f2} = \ldots = M_{fz}$, the equation can be simplified:

$$\gamma = \frac{M_f (l_1 + l_2 + ... + l_z)}{J_0 G} \tag{3}$$

The length of the rod to which each torque is applied can be expressed through the distance between them

$$l_z = l + \frac{z-1}{z} l + \frac{z-2}{z} l + ... + \frac{z-(z-1)}{z} l = \sum_{f=0}^{z-1} \frac{z-f}{z} l, \tag{4}$$

where z is a number of cylinders,
f – the summation index.

With this consideration in mind let us write down:

$$\gamma = \frac{M_f \cdot \sum_{f=0}^{z-1} \frac{z-f}{z} l}{J_0 G}, \tag{5}$$

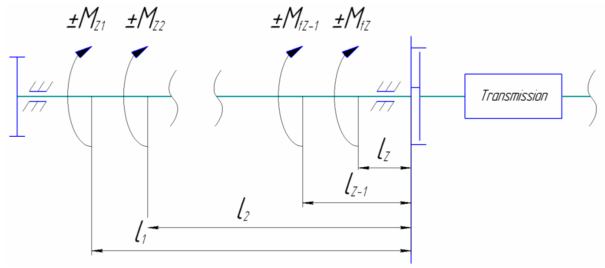

Figure 1. Crankshaft computational pattern

Assuming that

$$M_f = \frac{1}{z} M_e, \tag{6}$$

we obtain:

$$\gamma = \frac{M_e \cdot \sum_{f=0}^{z-1} \frac{(z-f)}{z} l}{J_0 G \cdot z} \tag{7}$$

Let us linearize the equation:

$$\partial M_e = \frac{z J_0 G}{\sum_{f=0}^{z-1} \frac{(z-f)}{z} l} \cdot \partial \gamma \tag{8}$$

By setting $R = \dfrac{z \cdot J_0 G}{\sum_{f=0}^{z-1} \frac{(z-f)}{z} l}$ (9)

"specific crankshaft torque of engine consisting of z-cylinders" in equation (8), we arrive at:

$$\partial M_e = R \partial \gamma \tag{10}$$

Under the real operating conditions the MTA diesel engine workflow can be described by the equation for unsteady modes that has the following form [11,12]:

$$J_D \frac{d \Delta \omega_D}{dt} = \Delta M_e - \Delta M_R \tag{11}$$

where $J_D \frac{d \Delta \omega_D}{dt}$ is variation of the moving parts inertia moment;
ΔM_e – torque variation;
ΔM_R – resistance moment variation.

With regard to equations (9, 10) incremental mo–ments are determined by equations:

$$\Delta M_e = \frac{R \partial \gamma}{\partial \omega_D} \Delta \omega_D + \frac{R \partial \gamma}{\partial g_c} \Delta g_c, \tag{12}$$

$$\Delta M_R = \frac{\partial M_R}{\partial \omega_D} \Delta \omega_D + \frac{\partial M_R}{\partial N} \Delta N, \tag{13}$$

where $\Delta \omega_D$ is angular velocity variation;

Δg_c - cyclic fuel feed variation;

ΔN - load variation.

Then equation (11) will take the following form:

$$J_D \frac{d\Delta\omega_D}{dt} = \frac{R\partial\gamma}{\partial\omega_D}\Delta\omega_D + \frac{R\partial\gamma}{\partial g_c}\Delta g_c - \frac{\partial M_R}{\partial\omega_D}\Delta\omega_D - \frac{\partial M_R}{\partial N}\Delta N. \quad (14)$$

Taking into account load control of the diesel engine with respect to the MTA twist angle, Δg_c variation can be expressed through load variation ΔN:

$$\frac{R\partial\gamma}{\partial g_c}\Delta g_c = \frac{R\partial\gamma}{\partial g_c}\cdot\frac{\partial g_c}{\partial N}\cdot\Delta N \quad (15)$$

Based on the above the equation will look like:

$$J_D \frac{d\Delta\omega_D}{dt} = \frac{R\partial\gamma}{\partial\omega_D}\Delta\omega_D + \frac{R\partial\gamma}{\partial g_c}\cdot\frac{\partial g_c}{\partial N}\Delta N - \\ -\frac{\partial M_R}{\partial\omega_D}\Delta\omega_D - \frac{\partial M_R}{\partial N}\Delta N \quad (16)$$

$$J_D \frac{d\Delta\omega_D}{dt} + \left[\frac{\partial M_R}{\partial\omega_D} - \frac{R\partial\gamma}{\partial\omega_D}\right]\cdot\Delta\omega_D = \\ = \left[\frac{R\partial\gamma}{\partial g_c}\cdot\frac{\partial g_c}{\partial N} - \frac{\partial M_R}{\partial N}\right]\cdot\Delta N \quad (17)$$

Let us set:

$F_g = \frac{\partial M_R}{\partial\omega_D} - \frac{R\partial\gamma}{\partial\omega_D}$ - the engine stability factor exp–ressed through crankshaft twist angle; (18)

$K_{R\gamma g_c N} = \frac{R\partial\gamma}{\partial g_c}\cdot\frac{\partial g_c}{\partial N}$ - the coefficient of specific shaft torque influence

$R\gamma \leftarrow g_c \leftarrow N$; (19)

$K_{M_R N} = \frac{\partial M_R}{\partial N}$ - the coefficient of resistance moment influence $M_R \leftarrow N$; (20)

Then we arrive at:

$$J_D \frac{d\Delta\omega_D}{dt} + F_g\cdot\Delta\omega_D = \left[K_{R\gamma g_c N} - K_{M_R N}\right]\cdot\Delta N \quad (21)$$

Let us introduce the following notation:

$$\phi = \frac{\Delta\omega}{\omega_o}, \quad (22)$$

where ϕ is a relative variation of crankshaft speed

$$\alpha_g = \frac{\Delta N}{N_o}, \quad (23)$$

where α_g is a relative variation of load

$$J_D \frac{d\Delta\omega_D}{dt}\cdot\frac{\omega_0}{\omega_0} = J_D\frac{d\phi}{dt}\omega_0, \quad (24)$$

Upon substitution the equation can be written as follows:

$$J_D\frac{d\phi}{dt}\omega_0 + F_g\cdot\phi\cdot\omega_0 = \left[K_{R\gamma g_c N} - K_{M_R N}\right]\cdot\alpha_g\cdot N_0 \quad (25)$$

If all the equation terms are divided by $\left[K_{R\gamma g_c N} - K_{M_R N}\right]\cdot N_0$, then

$$\frac{J_D\frac{d\phi}{dt}\cdot\omega_0}{\left[K_{R\gamma g_c N} - K_{M_R N}\right]\cdot N_0} + \frac{F_g\cdot\phi\cdot\omega_0}{\left[K_{R\gamma g_c N} - K_{M_R N}\right]\cdot N_0} = \alpha_g \quad (26)$$

Let us set:

$T_g = \frac{J_D\cdot\omega_0}{(K_{R\gamma g_c N} + K_{M_R N})\cdot N_0}$ - engine dynamic time as to load with respect to the crankshaft twist angle, (27)

$K_g = \frac{F_g\cdot\omega_0}{(K_{R\gamma g_c N} + K_{M_R N})\cdot N_0}$ - dynamic self-regulation coefficient as to load with respect to the crankshaft twist angle. (28)

Then the equation will take the following form:

$$T_g\cdot\frac{d\phi}{dt} + K_g\cdot\phi = \alpha_g, \quad (29)$$

The solution of this equation is given by:

$$\phi = \frac{\alpha_g}{K_g}\left(1 - e^{-\frac{K_g\cdot t}{T_g}}\right) \quad (30)$$

Violation of the equilibrium state can also occur as a result of processes unrelated to the fuel feeding system; therefore, equation (29) will be considered together with the classical mathematical model of a diesel engine, which is controlled on the basis of angular velocity:

$$T_g'\cdot\frac{d\phi}{dt} + K_g'\cdot\phi = \varphi_g - \alpha_g\cdot\Theta_g \quad (31)$$

where $T_g' = \frac{J_D\cdot\omega_0}{K_{R\gamma g_c}\cdot g_{c0}}$ - engine dynamic time (32)

$K_g' = \frac{F_g\cdot\omega_0}{K_{R\gamma g_c}\cdot g_{c0}}$ - engine self-regulation coefficient (33)

$\Theta_g = \frac{K_{M_R N}\cdot N_0}{K_{R\gamma g_c}\cdot g_{c0}}$ - load gain factor with respect to the crankshaft twist angle (34)

$\varphi_g = \frac{\Delta g_c}{g_{c0}}$ - relative variation of the rack position. (35)

We obtain:

$$T_g'\cdot\frac{d\phi}{dt} + K_g'\cdot\phi = \varphi_g - (T_g\cdot\frac{d\phi}{dt} + K_g\cdot\phi)\cdot\Theta_g \quad (36)$$

On rearrangement the equation will look like:

$$\left(T_g' + T_g\cdot\Theta_g\right)\cdot\frac{d\phi}{dt} + \left(K_g' - K_g\cdot\Theta_g\right)\cdot\phi = \varphi_g \quad (37)$$

where

$$T_g'' = T_g' + T_g\cdot\Theta_g = \frac{J_Д\cdot\omega_0}{K_{R\gamma g_c}\cdot g_{c0}} + \\ + \frac{J_Д\cdot\omega_0}{(K_{R\gamma g_c N} + K_{M_R N})\cdot N_0}\cdot\frac{K_{M_c N}\cdot N_0}{K_{R\gamma g_c}\cdot g_{c0}} =$$

$$= J_D \cdot \omega_0 \cdot \left(1 + \frac{K_{M_R N}}{K_{R\gamma g_c N} + K_{M_R N}}\right)$$ - engine dynamic time;

$$K_g'' = K_g' + K_g \cdot \Theta_g = \frac{F_g \cdot \omega_0}{K_{R\gamma g_c} \cdot g_{c0}} - \\ - \frac{F_g \cdot \omega_0}{(K_{R\gamma g_c N} + K_{M_R N}) \cdot N_0} \cdot \frac{K_{M_R N} \cdot N_0}{K_{R\gamma g_c} \cdot g_{c0}} = \tag{38}$$

$$= F_g \cdot \omega_0 \cdot \left(1 - \frac{K_{M_R N}}{K_{R\gamma g_c N} + K_{M_R N}}\right)$$ - engine self-regulation coefficient. (39)

Considering the above, relative variation of the rack position is found by equation:

$$T_g'' \cdot \frac{d\phi}{dt} + K_g'' \cdot \phi = \varphi_g \tag{40}$$

The solution of this equation is given by:

$$\phi = \frac{\varphi_g}{K_g''} \left(1 - e^{-\frac{K_g'' \cdot t}{T_g''}}\right) \tag{41}$$

The resulting new mathematical description is a model of the diesel engine and expresses the angular velocity variation of the crankshaft due to the relative variation of the position of the fuel feed control of the fuel pump with regard to the crankshaft load, determined by measuring the crankshaft twist angle.

The dependence of engine crankshaft twist angle on the torque is shown in Figure 2.

The curve shows that the greater the engine torque, the greater the twist angle of the engine crankshaft. Thus, a twisting angle of 0.00007 rad corresponds to the torque of 150 Nm.

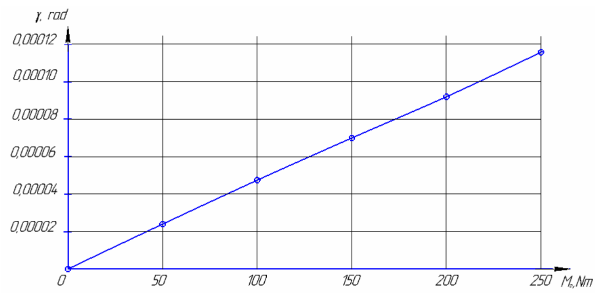

Figure 2. Twist angle-versus-engine torque curve

In accordance with the obtained mathematical model, the authors developed an electronic device for positional input action on the fuel feed controls of a diesel engine with an additional load impact depending on the crankshaft twist value [11].

The technical result consists in increasing the accuracy and timeliness of monitoring load changes, improvement of the dynamic characteristics of the regulator and increased acceleration capacity of the engine.

Figure 3 shows the function block diagram of the fuel feed control system equipped with an advanced electronic rpm governor for the diesel engine with an additional impact due to the value of the crankshaft twist angle.

The rack (1) of the high-pressure fuel pump (2) is connected to the armature (3) of the electromagnet (4). The regulator is equipped with engine crank revolutions detectors (5), the mass sensor for air entering the cylinders (6), the rack position indicator (7), the position indicator for fuel feed control pedals (8), the crankshaft twist angle sensor (9) with two sensing elements (10) installed at different ends of the crankshaft, and the electronic controller (11).

The operating principle of the advanced electronic regulator is as follows: the required position of the rack (1) is determined by the controller (11) according to the signals of the position indicators for fuel feed control pedals (8), the crank revolutions detectors (5), the mass sensor for air entering the cylinders (6) and the crankshaft twist angle sensor (9). The actual position of the rack (1) is determined by means of the rack position indicator (7). If there is a mismatch, the controller changes the signal applied to the electromagnet (4) by changing the position of the fuel pump rack. The controller is able to additionally adjust the position of the fuel pump rack according to the value of crankshaft torquability.

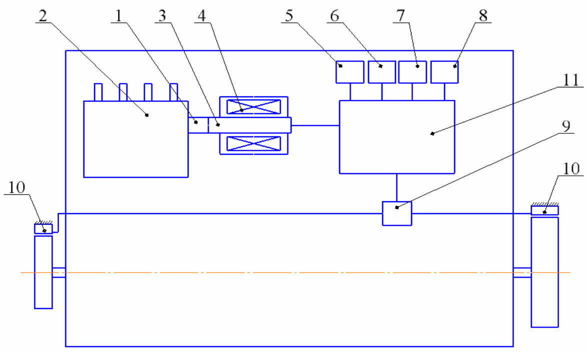

Figure 3. Function block diagram of the fuel feed control system having an advanced electronic crankshaft rpm governor for the diesel engine with an additional input action depending on the value of the crankshaft twist angle

1 – rack; 2 – high-pressure fuel pump; 3 – armature; 4 – electromagnet; 5 – engine crank revolutions detector; 6 – mass sensor for air entering the cylinders; 7 – rack position indicator;8 – position indicator for fuel feed control pedal; 9 – crankshaft twist angle sensor; 10 – sensing elements installed at different ends of the crankshaft; 11- electronic controller.

The accuracy of determining the engine load is increased since the electromagnet is controlled by the signals of the regulator, which makes an additional input action due to the value of the crankshaft twist angle. The crankshaft twist angle is proportional to the load on the machine-tractor aggregate and is determined by means of a twist angle sensor, which consists of two sensing elements installed at different ends of the crankshaft. The sensor generates a signal in the form of pulses, which are compared by the electronic controller, and determines the phase lag of the signals from each other. The magnitude of the lag is directly proportional to the magnitude of the load applied on the crankshaft. The obtained value of the phase lag magnitude is used by the controller as an input parameter of the diesel engine automated control system to adjust the fuel feed control signal within the specified speed mode. This design eliminates interference in the signal from the influence of oscillatory processes in the transmission, the effect of multidirectional forces on the sensor when the machine and tractor aggregate is moving over the uneven surface of the field, making turns and lifts, as well as the presence of backlashes in mechanical connections and distortion of the signal during transmission to the control system.

3. RESULTS

Benchmark comparative tests of the 4UTNI high-pressure fuel pump (HPFIP) showed significant advantages of the developed electronic regulator with a positioning device as compared to the serial all-mode mechanical regulator for reducing the controlling response time. Thus, in the event of equilibrium state perturbation, due to an increase in the load (up to 8%) in the stationary mode of the diesel engine operation, the decrease in the angular crankshaft speed was lower by 6%, the relative displacement of the HP fuel pump rack was reduced by 1.1 mm, and the total duration of the transient process was 2.2 s, which is by 37% lower than when working with a mechanical regulator.

The amplitude of the rack oscillations of the pump with the mechanical regulator reached Δ = 0.51 mm; after installation of the electronic positioning device it amounted to only Δ = 0.12 mm, which is by 45% lower, and as a consequence, non-uniformity of fuel feeding decreased by 7%.

A mathematical model of a diesel engine with an advanced electronic regulator has been developed which makes additional impact due to crankshaft twist value to determine the engine load. The twist angle-versus-engine torque plot is given. The principle of operation of the advanced electronic regulator of the positional impact is described and its function block diagram is presented.

4. CONCLUSION

The mathematical model of the machine-tractor aggregate diesel engine enables to determine the optimal limits for fuel feed control when operating at non-steady-state modes. The scientific value of the model lies in the possibility of calculating the fuel feed amount based on the crankshaft twisting value, which is proportional to the magnitude of the load on the diesel engine. The model can be used to improve the accuracy of fuel feed control in diesel engines of any design and power capacity. The tech–nical effect of the developed electronic regulator of the positional action is in reduction of the fuel feed control inertia, the increase in accuracy and timeliness of monito–ring the load changes and, as a consequence, increased fuel feed uniformity and reduced fuel consumption.

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МАТЕМАТИЧКИ ОПИС ДИЗЕЛ МОТОРА ТРАКТОРСКО-МАШИНСКОГ АГРЕГАТА СА ЕЛЕКТРОНСКИМ РЕГУЛАТОРОМ ПОЗИЦИЈЕ

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Рад приказује математички опис дизел мотора тракторско-машинског агрегата са развијеним електронским регулатором позиције постоља са пумпом за убризгавање горива. Развијени математички модел дизел мотора са осавремењеним електронским регулатором, применом додатног ефекта вредности увијања радилице, омогућава да се одреде границе регулације убризгавања горива у условима нестабилног начина рада за моторе различитог степена снаге и дизајна. Описан је принцип рада унапређеног електронског регулатора позиционог утицаја и дат је његов блок дијаграм.