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ESTIMATION MODEL OF THE DIESEL ENGINE FUEL SYSTEM WITH AN ELECTROMECHANICAL DEVICE TO INTENSIFY FUEL SUPPLY

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Окреслені ключові напрямки розвитку та вдосконалення паливної системи для двигунів внутрішнього згорання із запаленням від стиску паливоповітряної суміші. Доведена необхідність всебічного впровадження електромеханічних принципів керування процесом паливоподачі у відповідності до варіативних умов експлуатації дизелів. Висвітлена актуальність подальшого вдосконалення та модернізації гідромеханічних паливних систем на фоні постійно зростаючого інтересу до електрокерованих аналогів. Наведено перелік не використаних потенційних можливостей гідромеханічної паливної апаратури для покращення умов протікання процесу подачі палива. Запропоновано електромеханічний засіб для інтенсифікації паливоподачі, що вмонтовується до паливонагнітального трубопроводу та видозмінює фазо-амплітудну характеристику хвильового процесу розповсюдження одиночного імпульсу подачі між паливним насосом високого тиску та гідромеханічною форсункою. Висвітлено основні аспекти методики уточнення розрахункової моделі паливної системи безпосередньої дії розподільного типу з новим засобом інтенсифікації подачі палива. Запропоновано розглядати процес паливоподачі в декілька етапів з врахуванням особливостей функціонування окремо взятого гідравлічного вузла паливної системи, включаючи запропонований засіб інтенсифікації. Приведені системи диференціальних та аналітичних рівнянь, що дозволяють проводити математичне моделювання процесу розповсюдження та взаємовпливу хвиль тиску в паливонагнітальному тракті. Отримані системи дозволяють отримувати характеристики зміни гідравлічного тиску в різних паливних об'ємах, кінематики руху запірних елементів насосу високого тиску та форсунки, тощо.

У ході порівняльних досліджень, що були проведені на основі сформованої розрахункової моделі, процесу паливоподачі для штатної та вдосконаленої паливної системи турбодизеля були виявлено значне покращення якості впорскування за значною кількістю показників. За результатами розрахунків прослідковується наявність стрімкоподібного характеру зростання та спадання тиску впорскування на початковій та завершальній фазі процесу подачі палива до циліндрів дизеля. Відмічено, що швидкість зміни тиску може досягати 170 МПа/град, максимальний та середній тиск впорсквання зростає до 75 МПа та 30…40 МПа відповідно. Розрахункові дослідження проводились із залученням чисельного методу інтегрування – інтерполяційного методу Адамса, вибір якого обумовлено потребою отримання стіких рішень при вирішенні сформованих систем диференціальних рівнянь, що відносяться до категорії жорстких

Ключові слова: розрахункова модель, інтенсифікація паливоподачі, паливний насос, електромеханічний засіб, хвилі тиску

1. Introduction

The key and prerogative tendency in the development of automotive transport industry is the constantly-growing requirements to the level of emissions of toxic substances in exhaust gases of vehicles equipped with internal combustion engines. This fact outlines and determines the direction of the development and improvement of the key systems of power plants that have direct or indirect influence on the environmental friendliness of a plant. In addition, it remains a relevant problem to improve the indicators of the fuel saving component of operation of vehicles, which in recent years is acquiring an acute character amid a growing rate of rising costs of petrol fuel. An additional motivating factor in the need to enhance the profitability is the rapid development and spreading the vehicles with the electromechanical drive that are more economical in their operation.

These fuel-saving and environmental aspects of the development of modern engine design, in particular regarding creation of power plants with the ignition of operating mixture from compression are manifested in the development of integrated scientific and technical approaches to the improvement of the supply system. The main directions to improvement include the intensification of the fuel supply process, increasing flexibility of control and improvement of adaptivity to various conditions of operation of the internal combustion engine in different speed-load modes, etc. In this case, the practical opportunity to implement the necessary performance characteristic of fuel systems is realized using electromechanical executing devices and microcontroller devices for monitoring, control and multifactor management of the fuel supply processes.

At the same time, the traditional fuel system with hydromechanical control, despite the existing shortcomings, still takes a considerable share in the total volume of power supply systems mounted on the internal combustion engine of a large numbers of vehicles. They are especially spread in the power plants of the older generation and are actively operated in national equipment. According to their functional characteristics and the degree of flexibility of control, these systems are significantly inferior to their electromechanical counterparts, but potentially they still have some reserve for their development, which will extend the operation time and improve the performance of internal combustion engines.

Therefore, it is a relevant problem to improve the existing hydromechanical power supply systems with realization of the potential for improvement in the fuel supply process, with the purpose of increasing the competition in comparison to electromechanical equivalents.

2. Literature review and problem statement

As noted in paper [1], one of the traditional directions of modernization of hydromechanical fuel systems of direct action is the intensification of fuel supply, which is achieved by the optimization of characteristics of the fuel pump drive. Thus, in paper [2], the design of the hypocycloid drive, which gives significant improvement of kinematic parameters of the pumping plunger in comparison with the traditional cam drive, is proposed. A similar approach is outlined in paper [3], where, in order to increase pressure in the fuel supply system, it is proposed to increase the rate mode of the fuel pump operation. But the main problem in the implementation of these and similar ideas is raising the level of wear of contact elements, the emergence of critical contacts and dynamic stresses, etc. This leads to making the design more expensive and increasing the weight and dimensional parameters of the fuel system.

Another option to improve the fuel supply parameters is the modernization of the hydro-mechanical nozzles. One of the variants of this modernization is presented in study [4], where the structural change of the outlet nozzles of the sprayer positively influences the improvement of the maximum and medium injection pressure. The variant, described in paper [5], is similar to this approach. According to this work, it was found that the change in the ratio of the diameters of the inlet and outlet sections of the sprayer nozzles makes it possible to adjust the depth of penetration of a fuel jet to the combustion chamber volume and to improve the fuel distribution in it. The alternative variant of the modernization of the fuel injection node is presented in papers [6–8], in which it is proposed to use the double-spring injector with the differential piston in order to intensify and implement the multi-stage injection. But the presented methods for the improvement of the conditions of fuel supply involve significant structural variations for fuel-injection means.

The fundamental solution to the task on improving the fuel supply conditions is to use a fuel system of direct action with electromechanical nozzles, which are the focus of quite a large number of papers and publications [9–12]. But re-equipment of the existing vehicles that apply the separated fuel system is an expensive measure and requires coordination with other systems of power plants.

There is another variant of solving the problem of the fuel supply intensification that is implemented without any significant structural changes to the fuel system and significant capital investment. It is the use of technical means, which influence the wave processes in the discharge fuel line and improve the fuel supply process without introducing the changes in the structure of nozzles and the fuel pump. According to paper [13], this technique to improve fuel supply is almost the only effective step of improvement of the fuel systems of separate type.

These technical devices represent the hydraulic node with the internal fuel volume with one or more plunger elements that cause a mechanical movement under the influence of a variety of force efforts. These devices are mounted in the fuel discharge line between the fuel pump and the nozzle. They include the technical means of fuel supply adjustment that are diverse in their execution [13–18]: dampers, compensators, resonators, modulators, and others. The evident option of these devices is the modulator of pressure pulses, introduced and studied in [13].

As a rule, the outlined technical means are used to obtain a positive effect in a certain speed-load mode of engine operation, which somewhat lessens its use in other modes. That is why there occurs a need for the development of similar means of the fuel-supply intensification, but with the adaptive principle of their control under a wide range of engine operation modes.

3. The aim and objectives of the study

The aim of this research is to refine an estimation model of the fuel supply process of the fuel system of direct action using the new electromechanical device to intensify fuel supply. Achievement of this aim will make it possible to form an idea about the degree of an essential change in qualitative indicators of the fuel supply process, in particular, the maximum and medium injection pressure and duration of the specified hydrodynamic process.

To accomplish the aim, the following tasks have been set:

– to provide the technical description of the new electromechanical means for fuel supply intensification for the traditional hydromechanical fuel systems;

– to present the hydraulic circuit of the fuel system with a new means of the fuel supply intensification;

– to display the nuances of operation of the fuel system with the proposed electromechanical node and the influence of the latter on the formation of the process of fuel supply to the cylinders of the diesel engine by the refinement of the calculation model of separate hydraulic fuel supply systems;

– to conduct a comparative calculation study of the process of fuel injection to the cylinders of the diesel engine for the standard and improved versions of the separate fuel system of the diesel engine with the comparison of the key hydrodynamic indicators and characteristics.

4. Technical description of the electromechanical device for fuel supply intensification and representation of the hydraulic circuit of the fuel system with the new electro-controlled hydraulic node

The developed electromechanical device [19, 20] for the fuel supply intensification is an alternative option of the known modulator of pressure pulses [13].

By analogy to the modulator of pressure pulses, the electromechanical device (hereinafter, hydraulic corrector) is a part of the fuel discharge tract and is located between the high-pressure fuel pump and the hydromechanical nozzle. Hydraulic circuit of the fuel system with the hydraulic corrector is shown in Fig. 1, *a*. By its design implementation, the hydraulic corrector is a cylindrical rod (Fig. 1, *b*) with a ring groove in its middle part, to which the inlet and the outlet fuel ducts are connected, which, accordingly, join the hydraulic corrector with the fuel pimp and the injector.

In this case, in one of its extreme positions, the cylindrical part of the rod shuts off the outlet duct, at the same time leaving the inlet duct open. When moving the rod to the other extreme position, both ducts are open and joined with the ring groove of the rod, thus ensuring uninterrupted flowing of fuel streams from one duct to another. Transition of the rod from one position to another is carried out by electromagnetic force generated by the electric magnet, the anchor of which is the specified rod.

The main difference of the improved fuel system in comparison with its serial option is the existence in the discharge fuel line of the additional fuel volume, which affects the fuel supply process throughout its course, especially in the initial phase. Taking into consideration the impact of this volume on the fuel supply process, this process can be separately considered in the two ducts of the discharge fuel line. In this case, the hydraulic corrector with the variable flow cross section of the outlet duct acts as the separation boundary and the additional fuel volume of the ring groove of the rod. The magnitude of the additional fuel volume affects not only the magnitude of supply pulse and the character of the fuel flow motion in both ducts, but also determines the character of a change in pressure in the volume of the nozzle.

Thus, when carrying out mathematical modeling of the motion of supply pulse from the fuel pump to the nozzle, there occurs the need to describe mathematically the pulse propagation in ducts of the discharge fuel line. In this case, it is necessary to take into consideration the effect on the pulse of fuel supply of the fuel volume of the hydraulic corrector.

To simulate the motion of the supply pulse along the discharge tract, we chose the hydrodynamic model, which considered the process of fuel supply as the pulse that occurs in the inlet section of the high-pressure fuel pump and is sent along the fuel line at sonic speed to the hydraulic nozzle. In the nozzle, the jump-like transition from the flow section of the discharge pipeline to the sprayer holes inhibits the fuel motion and increases its pressure, which leads to emergence of hydraulic impact. This makes it possible to describe the fuel motion in the fuel discharge line by equations from the theory of a hydraulic impact, which take the following form [21]:

$$\begin{cases} \dfrac{\partial c}{\partial t} = -\dfrac{1}{\rho} \cdot \dfrac{\partial p}{\partial x}; \\ \dfrac{\partial c}{\partial x} = -\dfrac{1}{a^2 \rho_{pp}} \cdot \dfrac{\partial p}{\partial t}, \end{cases} \tag{1}$$

where c is the rate of fuel motion towards the nozzle; p is the fuel pressure; ρ is the fuel density; a is the local rate of pressure wave propagation; x is the coordinate by the length of pipe line; t is the time.

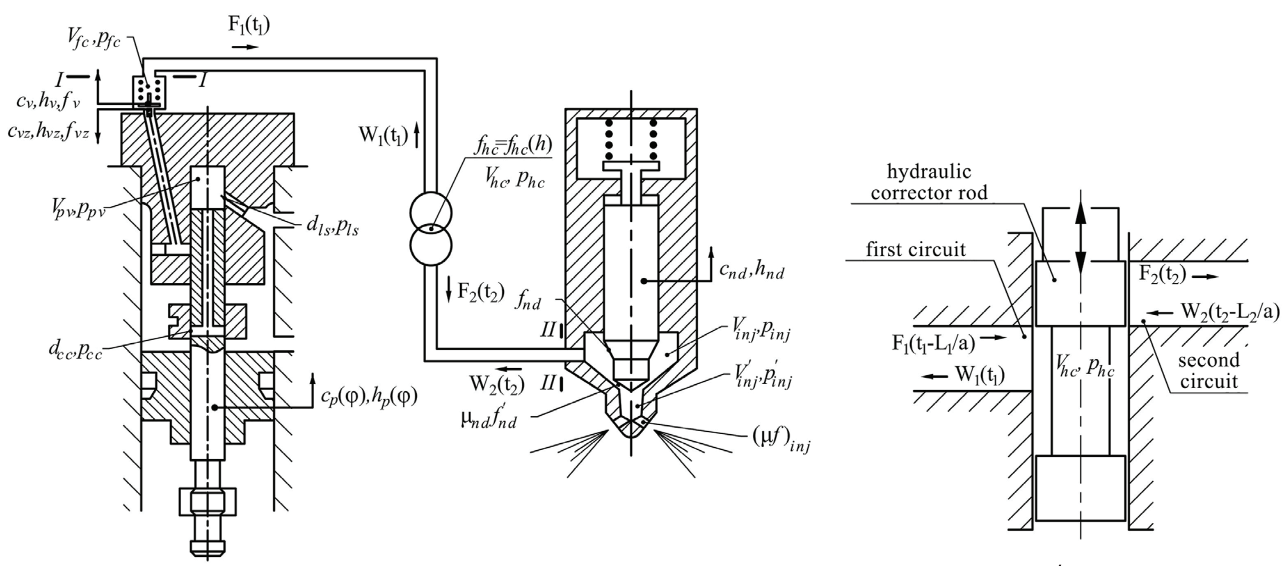

Fig. 1. Hydraulic scheme of fuel system of the diesel engine: *a* — relations between key hydraulic nodes of the system; *b* — internal structure of the electromechanical device of fuel supply intensification

In the general case at a hydraulic impact, for complex boundary conditions and a single supply pulse, it is appropriate to represent a solution to the system of equations (1) in the form of algebraic equations by d'Alembert:

$$\begin{cases} p - p_r = F\left(t - \dfrac{x}{a}\right) - W\left(t + \dfrac{x}{a}\right); \\ c - c_r = \dfrac{1}{a\rho}\left[F\left(t - \dfrac{x}{a}\right) + W\left(t + \dfrac{x}{a}\right)\right], \end{cases} \quad (2)$$

where p, c are the pressure and rate of fuel flow in the line; p_r, c_r are the residual pressure and rate of fuel flow in the line at the beginning of the new injection process.

Functions $F\left(t - \dfrac{x}{a}\right)$ and $W\left(t + \dfrac{x}{a}\right)$ model the motion of the direct and return pressure wave along the line, Return pressure waves propagate in the direction that is opposite to the main supply pulse and occur as a result of an abrupt change of the effective flow area.

5. Refinement of the estimation model of the separated hydraulic fuel supply system with the new electromechanical device of fuel injection intensification

Consider the flow of the process of pressure waves propagations in above areas of the discharge fuel line, according to the hydraulic circuit, as shown in Fig. 1, *a*.

Thus, with a gradual shut-off of the suction sleeve window by the end edge of the plunger, the pressure in the over-plunger volume increases with the discharge valve getting off from the seat. The motion of the valve provokes an increase in pressure in the fitting cavity (p_{fc}) and the emergence of unsteady fuel motion in the inlet section of the first duct of the fuel line, which causes the formation of the direct pressure wave F_1. The formed wave F_1 propagates at local sonic speed a along the first duct and over a period of time t_1 reaches the internal volume of the hydraulic corrector.

As a result of a sudden expansion of the volume of the discharge line in the hydraulic corrector, which is a local hydraulic resistance to the fuel flow, return wave W_1, directed opposite the main wave F_1, occurs in the first duct:

$$W_1(t_1) = F_1\left(t_1 - \frac{L_1}{a}\right). \quad (3)$$

Prior to the arrival of return wave W_1, the magnitude of pressure in direct wave F_1 will be determined from the known expression:

$$F_1 = p_{fc} - p_r, \quad (4)$$

where p_r is the residual pressure in the line; p_{fc} is the pressure in the fitting cavity.

In this case, the rate of fuel arrival c_1 to the fuel pipeline in the inlet section of the first duct, taking into consideration the system of equations (2), is determined from:

$$c_1 = \frac{F_1}{a\rho}, \quad (5)$$

where ρ is the fuel density.

Wave W_1, returning to the inlet section of fuel line, is a wave of compression, which makes the fuel from the fuel line come back to the fitting cavity, thereby reducing the rate of fuel flow in the main direction.

Taking into consideration system (2), the rate of fuel flow c_1 can be found from expression:

$$c_1 = \frac{1}{a\rho}\left(F_1(t) - W_1\left(t - \frac{L_1}{a}\right)\right), \quad (6)$$

where L_1 is the length of the first duct.

A decrease in the rate of the fuel flow motion in the inlet section of the first duct on the background of an increase in the volume fuel supply from the over-plunger volume leads to increased pressure in the space of the fitting:

$$p_{fc}(t) = F_1(t) + W_1\left(t - \frac{L_1}{a}\right) + p_r. \quad (7)$$

Besides, return wave W_1 prevents the growth of the amplitude of pressure of direct wave F_1.

In the fuel volume of the hydraulic corrector at the closed fuel flow duct, direct wave F_1 causes an increase in hydraulic pressure, the rate of an increase of which depends on the amplitude of the direct wave. Pressure in the inlet section of the hydraulic corrector can be determined from formula:

$$p_{hc}(t) = F_1\left(t - \frac{L_1}{a}\right) + W_1(t) + p_r. \quad (8)$$

The character of formation of direct F_1 and return wave W_1 at the inlet section of the volume of the hydraulic corrector remains constant until the rod of the hydraulic corrector starts moving and opening the flow duct. After the beginning of lifting the rod of the hydraulic corrector, the area of the flow duct gradually begins to increase. This leads to intense excitation of the fuel flow by the hydraulic corrector and the formation in its outlet section of the direct pressure wave F_2, the amplitude of which can be found from formula:

$$F_2 = p_{hc} - p_r. \quad (9)$$

Direct wave F_2, moving along the second duct of the length of L_2 is directed to the fuel volume of the hydro-controlled nozzle, causing an increase in pressure in it.

At the same time, due to a sudden transition from the section of the fuel ducts of the hydraulic corrector to the closed holes of the sprayer, there occurs return wave W_2 that propagates towards the hydraulic corrector.

In this case, the fuel rate c_{inj} in the section near the nozzle and the change of pressure in the return wave W_2 can be determined from the following equations:

$$\begin{cases} W_2(t_2) = p_{inj} - p_r - F_2\left(t_2 - \dfrac{L_2}{a}\right); \\ c_{inj} = \dfrac{1}{a\rho}\left[2 \cdot F_2\left(t_2 - \dfrac{L_2}{a}\right) - p_{inj} + p_r\right], \end{cases} \quad (10)$$

where p_{inj} is the pressure in the volume of the injector.

Wave W_2 moves to the volume of the hydraulic corrector and affects the law of change of hydraulic pressure, the rate of the fuel flow out of this volume and nature of the formation of direct wave F_2. All changes of the listed parameters can be calculated by using the system of equations:

$$\begin{cases} p_{hc} - p_r = F_2(t_2) + W_2\left(t_2 - \dfrac{L_2}{a}\right); \\ c_{hco} - c_r = \dfrac{1}{a\rho}\left[F_2(t_2) - W_2\left(t_2 - \dfrac{L_2}{a}\right)\right], \end{cases} \quad (11)$$

where c_{hco} is the rate of fuel flow at the outlet from the hydraulic corrector.

It should be noted that the area of the fuel flow duct of the hydraulic corrector is not always constant, but is in the functional dependence on the coordinates of the linear motion of the rod of the hydraulic corrector.

The fuel flow area of the fuel duct can be determined by a geometric figure (a circle segment), formed by the circle section of the flow duct and the boundary line of the ring groove of the rod:

$$f_{hc} = \frac{r^2}{2} \cdot (\alpha - \sin\alpha), \quad (12)$$

where r is the radius of the flow duct; α is the central angle of the circle segment, rad.

Angle α is determined from expression:

$$\cos\frac{\alpha}{2} = 1 - \frac{h}{r}, \quad (13)$$

where h is the current height of the lift of a hydraulic corrector rod.

Typically, the current value of the height of lifting the hydraulic corrector rod is determined from the joint solution of the dynamics of electromagnet work and the calculation of its magnetic chain [22]. To avoid complex mathematical representations regarding providing functional dependence $h_{hc}=f(t)$ in the analytical form and to facilitate the theoretical research, this dependence can be determined experimentally and represented in the form of polynomial dependence of n-th degree:

$$h = a_0 + \sum_{n=1}^{k} a_n \cdot t^n. \quad (14)$$

The nature of a pressure change in the fitting of the pump p_{fc}, in the volume of hydraulic corrector p_{hc} and in the volume of the nozzle p_{inj} is determined from the equations of boundary conditions, formed for these fuel volumes of the fuel supply system.

Equations of boundary conditions combine equations of fuel balance in a certain section of the discharge pipeline and equations of motion of shut-off parts of the fuel system. In connection with the closing and opening of the suction and cut-off windows, the motion of the discharge valve and the nozzle sprayer needle, the boundary conditions are not constant, so there is a need to divide the fuel supply process into several stages. The number of these stages depends on the design of the system and the ratio between its elements. In this case, the principle of making the equations of boundary conditions for all the stages is the same. Only the number of components in each of the formed equations and the number of equations change.

It is common to consider the fuel supply process separately for the fuel pump and the hydro-controlled nozzle [21]. Thus, the fuel supply process for the studied fuel supply can be divided into the following stages:

– stage 1 – from the beginning of the plunger motion to the beginning of the motion of the discharge valve;

– stage 2 – from the beginning of the motion of the discharge valve to complete shut-off of the suction windows;

– stage 3 – from the beginning of the opening of the cut-off window to the moment of setting the discharge valve on the seat;

– stage 4 – from the moment of setting the discharge valve on the seat to the moment of closing the return valve;

– stage 5 – from the moment of setting the return valve to the initial position to the full stop of the fuel motion in the pump tract.

The process in the nozzle is characterized by the following steps:

– from the beginning of the motion of the sprayer needle to the moment of reaching the maximum height of the lift;

– from the moment of reaching the maximum height of the lift by the needle to the beginning of its setting;

– from the beginning of the motion of the needle in the return direction to the moment of its setting on the shut-off cone.

The fuel volume of the hydraulic corrector is characterized by the following two stages. The first stage takes place in the period when the flow duct, in which there is accumulation of the fuel flow, is closed. In turn, the second stage lasts from the moment of joining the volume with the flow duct of the hydraulic corrector up to the moment of the cessation of motion of pressure waves in the fuel pipeline.

When constructing the boundary conditions for each of the listed stages of the structural elements of the fuel system and the fuel supply process in general, the following simplifications are allowed:

1. The discharge and return valve and the nozzle needle move with the uniform acceleration.

2. Fuel pressure in the volume of the fitting and in the inlet section of the discharge pipeline are identical.

3. The pressure in the inlet section of the discharge pipeline and in the volume of the nozzle are equal.

4. The pressure in the inlet and outlet section of the hydraulic corrector is equal to the pressure in its fuel volume.

5. Instantaneous values of the rate of non-constant fuel motion are determined from formulas of constant motion.

6. Fuel throttling at the fuel arrival from the over-plunger volume to the cavity of the fuel nozzle, as well as throttling in the shut-off and suction widows, are taken into consideration by special coefficients of consumption.

7. The influence of the chamber volume before the nozzle holes of the sprayer on the injection process is not taken into consideration.

8. The force of friction between the shut-off needle and the body of the nozzle sprayer is not considered due to its insignificance compared to other power components of the equation of dynamic equilibrium.

9. As a result of relatively small excess pressure of injection for the studied fuel supply system, fuel losses through the gaps in the connection of the plunger-sleeve and needle-sprayer are not taken into account when constructing the equations of the fuel balance for the calculation sections of the discharge pipeline.

10. Based on the condition of small injection pressure, volume deformation of pipelines, fittings and the sleeve of the discharge plunger is not taken into account. For this same reason, elasticity of the cam shaft of the fuel pump is not taken into account either.

To avoid giving the description of boundary conditions for each fuel supply stage, we will make the following equations of boundary conditions, which would contain the maximum number of functional components that describe one or another stage of the process of fuel supply for the fuel pump. To implement the possibility of applying these equations to describe each of the stages, we will introduce to the composition the special step functions σ_i, where $i=1,...5$, in accordance with the recommendations given in papers [21, 23]. These functions take only two fixed numerical values of zero and unity, depending on the logical conditions imposed on them, and thereby establish the type and the number of key equations for each stage of calculation.

The generalizing system of equations of boundary conditions of the fuel supply process of the fuel pump and the inlet section I-I of the discharge pipeline will take the following form:

$$\begin{cases} \alpha V_{pv}\dfrac{dp_{pv}}{dt} = f_p\cdot c_p - \sigma_1\cdot sign\left(p_{pv}-p_{ls}\right)\cdot\mu_{ls}\cdot f_{ls}\times \\ \times\sqrt{\dfrac{2}{\rho}}\cdot\sqrt{p_{pv}-p_{ls}} - \sigma_2\cdot\mu_{cc}\cdot f_{cc}\cdot\sqrt{\dfrac{2}{\rho}}\cdot\sqrt{p_{pv}-p_{cc}} - \\ -\sigma_3\cdot sign\left(p_{pv}-p_{fc}\right)\cdot\mu_{fc}\cdot f_{fc}\cdot\sqrt{\dfrac{2}{\rho}}\cdot\sqrt{p_{pv}-p_{fc}} - \\ -\sigma_4\cdot f_v\cdot\dfrac{dh_v}{dt} + \sigma_5\cdot f_{rv}\cdot\dfrac{dh_{rv}}{dt}; \\ \alpha V_{inj}\dfrac{dp_{inj}}{dt} = \sigma_3\cdot sign\left(p_{pv}-p_{fc}\right)\cdot\mu_{fc}\cdot f_{fc}\times \\ \times\sqrt{\dfrac{2}{\rho}}\cdot\sqrt{p_{pv}-p_{fc}} + \sigma_4\cdot f_v\cdot\dfrac{dh_v}{dt} - \sigma_5\cdot f_{rv}\cdot\dfrac{dh_{rv}}{dt} - f_{pp}\cdot c_1; \\ \sigma_4\cdot M_v\dfrac{d^2h_v}{dt^2} = f_v\cdot\left(p_{pv}-p_{fc}\right) - \delta_v\cdot\left(h_{v0}+h_v\right); \\ \sigma_5\cdot M_{rv}\dfrac{d^2h_{rv}}{dt^2} = f_{rv}\cdot\left(p_{fc}-p_{pv}\right) - \delta_{rv}\cdot\left(h_{rv0}+h_{rv}\right), \end{cases} \quad (15)$$

where V_{pv}, V_{fc} are the volume of the pump chamber and the fitting of the fuel pump; $f_p, f_{ls}, f_{cc}, f_v, f_{rv}, f_{fc}$ are the corresponding areas of the section of the plunger, suction of cut-off windows, discharge and return valves and flow duct around shut-off valves; μ_{ls}, μ_{cc}, μ_{fc} are the coefficients of fuel consumption through the suction and cut-off windows, shut-off valves; $p_{pv}, p_{ls}, p_{cc}, p_{fc}$ are the pressure in the over-plunger volume, the low-pressure system, cut-off cavity and the fitting; c_p, c_1 are the rates of motion of the plunger and the fuel flow in the inlet section of the pipeline; h_v, h_{v0}, h_{rv}, h_{rv0} are the current and the initial height of lifting of the discharge and return valves; M_v, M_{rv} are the total weight of the parts that move along with the discharge and return valves; δ_v, δ_{rv} are the coefficients of rigidity of the spring of the discharge and return valves.

Let us give a brief description of the functional components of the equations, included in the system of equations (15). The first term of the first equation characterizes the amount of fuel that remains in the compression chamber. The second one determines the volumetric rate of pumping fuel by the pump plunger. The third and fourth ones express the volumetric rate of fuel flowing through the suction and cut-off windows. The fifth is the volumetric rate of fuel flowing from the over-plunger volume into the volume of the fitting through the flow duct near the return valve. The sixth provides the numerical feature of the rate of filling with fuel the volume, which is released after lifting the discharge valve. The seventh determines the amount of fuel that is pumped to the over-plunger space by the return valve during its motion under the influence of the return pressure wave, generated at the nozzle. Thus, the first equation describes the volume balance of fuel balance between the amount the plunger pumped during its motion and what this amount of fuel was spent for.

The second equation of system (15) is both the equation of fuel balance for the pump fitting and for the inlet section of the pipeline. The first term of this equation determines the amount of fuel that remains compressed in the volume of the fitting. The second determines the amount of fuel that arrived at the volume of the fitting through the flow section of the return valve. The third describes the amount of fuel that is pumped by the discharge valve in the fitter during its motion during fuel supply. The fourth reflects the volume rate of filling the volume, emptied by the return valve during its motion under the influence of the return pressure wave after stopping the fuel supply and setting the discharging valve on the seat. The fifth describes the amount of fuel that entered the discharge pipeline.

The last two equations of the system (15) characterize the dynamic balance, respectively, of the discharge and return valve. The first component of these equations determines the force of inertia of the weight of the valve and accompanying parts that move with them. The second component determines the resulting force of hydraulic pressure that influences the valves from the discharge chamber and the fitting of the fuel pump. The third component takes into consideration the elasticity of springs on the valves.

It should be noted that the flow sections the suction and cut-off windows during the flow of the discharge plunger change their area and can be determined from the expression [21]:

– the area of suction windows:

$$f_{ls} = i\cdot\frac{r^2}{2}\cdot\left(2\pi - 2\chi + \sin 2\chi\right), \quad (16)$$

– the area of cut-off windows:

$$f_{cc} = i\cdot\frac{r^2}{2}\cdot\left(2\chi - \sin 2\chi\right), \quad (17)$$

where r are the radii of the suction and cut-off windows; i is the number of windows:

Angle χ is determined from expression:

$$\cos\chi = 1 - \frac{h - h_{beg}}{r}, \quad (18)$$

where h, h_{beg} are the current height of lifting the discharge plunger and the height of the plunger, at which the edge of the plunger begins to shut-off the suction or to open the shut-off windows.

The current volume of the discharge chamber and the fitting is determined from the following expressions:

$$V_{pv} = V'_{pv} + f_p\cdot\left(h_p^{\max} - h_p\right) + f_v\cdot h_v, \quad (19)$$

$$V_{fc} = V'_{fc} - f_v\cdot h_v, \quad (20)$$

where V'_p is the total volume of the discharge chamber at the position of the plunger in the upper dead point and the volumes of fuel ducts of the fuel section, along which fuel flows from the over-plunger volume to the fitting; V'_{fc} is the volume of the pump fitting at the fixed discharge valve.

The boundary conditions with the step functions in the nozzle of the closed type and the inlet section of the II-II discharge pipeline are determined from the following system of equations [21]:

$$\begin{cases} \alpha V_{inj}\cdot\dfrac{dp_f}{dt}= \\ = f_{pp}\cdot c_2-\sigma_6\cdot(\mu f)_{inj}\sqrt{\dfrac{2}{\rho}}\cdot\sqrt{p'_{inj}-p_z}-\sigma_7\cdot f_{nd}\cdot c_{nd}; \\ \sigma_7\cdot M\dfrac{d^2h_{nd}}{dt^2}=\left(f_{nd}-f_{pv}\right)\cdot\left(p_{inj}-p_{inj0}\right)+f_{pv}\cdot p'_{inj}-\delta\cdot y, \end{cases} \quad (21)$$

where V_{inj} is the fuel volume of the nozzle; p_z is the pressure of gases in the cylinder of the diesel engine; p_{inj} is the fuel pressure in the volume of the injector; p_{inj0} is the initial pressure of lifting the shut-off needle of the nozzle; p'_{inj} is the current fuel pressure in the chamber before nozzle holes of the injector sprayer; c_{nd}, c_2 are the rate of motion of the needle and fuel flow in the outlet section of the pipeline; f_{nd}, f_{pv} are the area of the section of the needle and its tip; δ is the rigidity of the spring of the sprayer needle; y is the magnitude of deformation of the needle spring; M is the total weight of movable parts of the injector; $(\mu f)_{inj}$ is the effective flow section of the nozzle holes of the sprayer.

The first equation of system (21) describes the fuel balance between what arrived from the outlet section of the pipeline and totality of fuel flows, between which this amount of fuel is distributed. The first component of this totality characterizes the volume of fuel injected to the cylinders of the diesel engine through the nozzle holes of the sprayer. The second is spent on filling the volume during the motion of the sprayer needle. The third is subjected to compression in the sprayer chamber.

The second equation of the system (21) describes the dynamic balance of the sprayer needle. The force of inertia of the needle (the left part of the equation) is equaled to the sum of forces of fuel (the second and third term of the equation) and elasticity of the spring (the fourth term of the equation).

In accordance with the work [21], the pressure at the nozzle holes of the sprayer is determined from the following expression:

$$p'_{inj}=\frac{k^2}{1+k^2}\left(p_{inj}-p_z\right)+p_z. \quad (22)$$

Magnitude k in this expression characterizes the ratio of the flow section between the cut-off cone of the needle and the seat and nozzle holes:

$$k=\frac{\mu_{nd}\cdot f'_{nd}}{(\mu f)_{inj}}. \quad (23)$$

To calculate the flow area f'_{nd} from the geometric parameters of the sprayer, expression [21] is used:

$$f'_{nd}=\pi\sin\frac{\varphi}{2}\left(d_{pch}-0.5\cdot y\sin\varphi\right)\cdot y, \quad (24)$$

where φ is the angle of the shut-off cone of the sprayer needle; d_{pch} is the diameter of the sprayer chamber before the nozzle holes.

Coefficient of consumption μ_{nd} is determined from experimental data presented in paper [21].

An efficient flow section of the nozzle holes of the sprayer is the functional dependence on the motion of the shut-off needle and is determined from expression [24]:

$$(\mu f)_{inj}=(\mu f)_{inj}^{\max}\cdot\left[\begin{gathered}2.64\frac{y}{y_{\max}}-2.37\left(\frac{y}{y_{\max}}\right)^2+ \\ +0.73\left(\frac{y}{y_{\max}}\right)^3\end{gathered}\right], \quad (25)$$

where $(\mu f)_{inj}^{\max}$ is the maximum flow section of the nozzle holes of the sprayer; y, $y_{\max}$ are the current and the maximum transposition of the shut-off needle of the sprayer.

For the volume of the hydraulic corrector, boundary conditions take the following form:

$$\alpha V_{hc}\cdot\frac{dp_{hc}}{dt}=f_t\cdot c_{hcin}-f_{hc}\left(h_{hc}\right)\cdot c_{hco}, \quad (26)$$

where V_{hc} is the fuel volume of the hydraulic corrector; p_{hc} is the current fuel pressure in the fuel volume of the hydraulic corrector.

The first component of this equation characterizes the volume amount of fuel, compressed in the fuel volume at evident disbalance of the fuel amount in it. The appearance of disbalance is caused by the difference between the amount of fuel supplied to the volume (the second component), and the fuel (the third component), which is drained from it when the flow duct of the hydraulic corrector of open. In the case of complete shut-off of the corrector rod by the edge of the ring groove, all fuel that arrives to the volume of the hydraulic corrector is exposed to compression, that is, it is accumulated.

To demonstrate visually the effectiveness of using the proposed technical solution and the method of the fuel supply intensification, the comparative research was performed. The characteristics of the change of the injection pressure for the standard fuel supply system and the system with the proposed electromechanical intensification device were compared by the calculation method. Calculation was performed at the fixed rotation rate of the cam shaft of the high-pressure fuel pump and the magnitude of cyclic supply, which was selected by the external speed characteristic of the diesel engine, according to the selected rotation rate. Kinematic, volume-dimensional, and fuel-volume parameters of each hydraulic nodes of the calculation model were borrowed from the fuel system with the distributing double-piston fuel high-pressure ND-22/6B4 of the turbo diesel engine 6CN13/11.5. The calculation of the single fuel supply process was performed for the high-speed operation mode of the diesel engine at the highest rotation rate of the crankshaft of 1,600 min^{-1}.

To obtain a solution to the formed systems of differential equations, we used the interpolation method by Adams, according to which the sought-for function can be found:

$$y_{n+1}=y_n+h\sum_{i=0}^{k}\beta_i f\left(t_{n+1-i},y_{n+1-i}\right), \quad (27)$$

where β_1, β_2, β_3...β_i are the numeric coefficients (for implicit methods $\beta_3\neq 0$); h is the integration pitch.

The choice of this method was predetermined by obtaining stable solutions despite the selected integration pitch, which is chosen based on the need to achieve the maximum degree of approximation of the obtained result to the true value.

It should be noted that when modeling the motion of pressure waves along the fuel line, we took into account the fact that during the propagation of waves, there occurs a certain loss of motion energy with the attenuation of the amplitude. This aspect is taken into consideration in the system of equations by d'Alembert by a special multiplier – attenuation decrement – that is a variable magnitude, dependent on pressure and fuel flow rate.

Fig. 2 shows combined results of the calculation research into the characteristics of a change in the fuel injection pressure for the standard and the studied fuel system depending on the angle of rotation of the cam shaft of the pump.

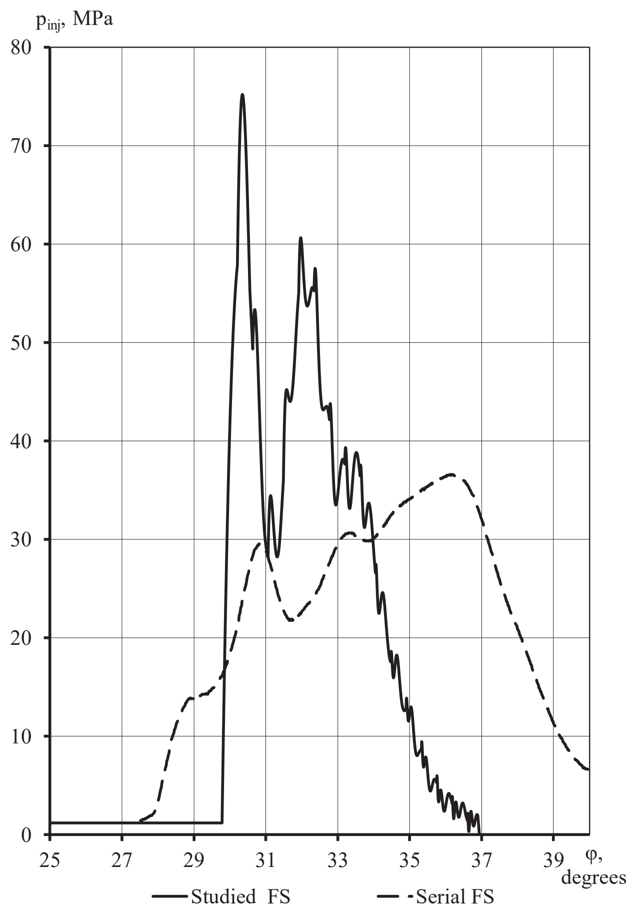

Fig. 2. Dependence of fuel injection pressure on angle of rotation of the cam shaft of the high-pressure pump at $n_{c.sh.}$ = 1,600 min^{-1}

6. Discussion of results of comparative estimation study into the fuel systems of diesel engines that are compared

The specific feature of the given characteristics of fuel injection for the fuel supply system with the studied intensification devices is a rather abrupt front and rear front of a change of the injector pressure. In particular, the rate of pressure increase at the rotation rate of 800 min^{-1} is about 170 MPa/degree, which significantly exceeds the rates of pressure increase in the comparative characteristic of the standard fuel system. The main reason for such intense pressure increase is too compressed fuel in the internal volume of the hydraulic corrector. In this case, after reaching the maximum value of pressure of 75 MPa, the pressure curve abruptly drops to the level of 27...28 MPa, which is caused by the beginning of the motion of the sprayer needle and its release of the additional volume, which lowers the pressure in the nozzle volume. After the needle coming to the rest, the pressure in the nozzle starts to grow and reaches its maximum peak at the level of 60 MPa. The existence of certain instability in changing the injection pressure with the amplitude of 5 MPa can be explained by the active movement of pressure waves between the fuel volume of the injector and the hydraulic corrector. The obtained peak values of injection pressure for the proposed method of the fuel supply intensification exceed the identical indicators of a single injection process for the standard fuel system by 30...40 MPa.

After a rather considerable increase in the injection pressure, not less rapid drop of pressure at the average rate of 48 MPa/degree occurs. This drop occurs due to the expansion wave, that was formed in the inlet section of the fuel discharge main line of the fitting of the fuel pump, due to opening cut-off windows of the plunger, and arrived at the fuel volume of the nozzle after the previous intensification in the volume of the hydraulic corrector. Thanks to such intensive fuel supply, the injection duration decreased from 14 to 7 degrees of the rotation of the fuel pump shaft with an increase in value of average injection pressure from 27 to 30 MPa.

The outlined direction of the improvement of separate fuel systems and the proposed electromechanical device for the fuel supply intensification demonstrated their unconditional advantage over existing hydraulic fuel system by quality indicators of the injection process and control flexibility. In addition, the represented fuel system is superior to its prototypes by its functional capabilities [13]. This is especially true of adaptivity, efficient control of the injection process and the absence of the functional of the intensification device for the separated speed-load operation mode of the diesel engine.This proposed technical solution makes it possible to some extent to solve the issues of economic efficiency and environmental friendliness of already existing diesel vehicles with hydraulic fuel supply systems and to bring in alternative solutions to the set of modern directions of development and improvement of fuel systems of vehicles.

The practical implementation of the outlined approaches to the improvement of fuel injection processes requires the use of high-performance executing devices with a minimum period of relaxation and small inertia manifestation in their work. It should also be noted that the effectiveness of the implementation of technical novelties, given the sharp increase in hydrodynamic components of the fuel injection process, requires a decrease in the degree of technological efficiency of the production of all components the fuel discharge tract. It is meant to prevent the occurrence of manifestations of the loss of the selected benefits because of possible existing fuel losses through small gaps between the precision tangent surfaces, low mechanical and hydraulic strength of the structural components, etc.

Within the framework of the subsequent solution of the outlined problems of the scientific research, it is necessary to expand the shaped model by inclusion to it of the additional complementary calculation points, specifically, the dynamic calculation of fuel lines, spring elements of the fuel system and kinematically-structural calculation of the drive of the high-pressure fuel pump. This created symbiosis will make it possible to fully and systematically approach the comprehensive principle of calculation and design of fuel systems and to approximate the results of calculation research to the experimental results as close as possible.

7. Conclusions

1. To improve the fuel supply process for fuel systems of the separate type, the technical device with the electromechanical control that is mounted to the discharge pipeline was developed. Its action implies the influence on the wave processes of the propagation of fuel supply pulse in the fuel discharge tract from the fuel pump to the hydro-controlled nozzle, causing significant changes in the amplitude and temporal characteristic of the process.

2. Based on the proposed principle of fuel supply intensification, the hydraulic circuit was developed, the practical implementation of which does not require any fundamental changes in the design of fuel equipment. This is due to the fact that the electromechanical devices, based on the proposed intensification principle, are mounted in the fuel line as a separate hydraulic node with the movable plunger element and internal fuel volume.

3. Construction of the calculation model of the fuel system with the new electromechanical device of the fuel supply intensification can be carried out with the involvement of the mathematical foundations of hydrodynamic calculation based on the wave theory of hydraulic impact. To obtain the solution of the constructed systems of differential equations, it is necessary the use implicit methods of numerical integration, specifically, the interpolation method by Adams.

4. Based on the obtained results of comparative research, it can be argued about a rather enough improvement of the fuel supply parameters for the fuel system with a new device of fuel supply intensification. In particular, there is an increase in the maximum and medium injection pressure, reducing the total duration of fuel supply through the nozzle. The existence of steep fronts of an increase and a decrease in hydraulic pressure will contribute to avoiding occurrence of additional injections and slowly developing character of fuel leakage through the sprayer nozzles.

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ANALYSIS AND SELECTION OF THE PARAMETRIC PROFILE OF A POWERPLANT ENGINE FOR A LIGHT TRAINER AIRCRAFT

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Актуальність проведених досліджень обумовлена поліпшенням паливної ефективності літака і, як наслідок, зменшенням вартості життєвого циклу авіаційного двигуна у складі силової установки навчально-тренувального літака типу DART-450. Теоретично обґрунтовано льотно-технічні та економічні характеристики сучасного легкого літака для навчання льотного складу. В основі методів дослідження використовується набір параметрів, характеристик і показників, що в цілому відображають техніко-економічну досконалість двигуна силової установки технічної системи «силова установка – планер» легкого навчально-тренувального літака.

Наукова новизна одержаних результатів полягає у формуванні нового параметричного обрису турбогвинтових двигунів силової установки для легкого навчально-тренувального літака типу DART-450 з урахуванням моделювання заданого польотного циклу літака та життєвого циклу двигуна.

Чисельними дослідженнями встановлено, що максимальна дальність польоту літака з різними двигунами при однаковій злітній масі визначається, в основному, запасом палива, а не економічністю витрати палива. Тому двигун найменшої потужності має перевагу у всіх характеристиках, крім злітної дистанції, яка найменша у літака з двигуном найбільшої потужності.

Результатами обгрунтовано, що для виконання задач по навчальному тренуванню льотного складу доцільно встановлення двигуна АІ-450СР, який має найменшу вартість життєвого циклу. Очевидно, що даний літак із встановленим двигуном буде мати найнижчу вартість льотної години. Однак для виконання розвідувальних та ударних задач на літаку типу DART-450 доцільно встановлення двигуна АІ-450СР-2. Для виконання тільки ударних задач на літаку типу DART-450 доцільно встановлення двигуна МС-500В-С, який має більшу потужність, ніж розглянуті двигуни

Ключові слова: навчально-тренувальний літак, життєвий цикл, льотно-технічні характеристики, турбогвинтовий двигун

1. Introduction

At present, much attention is paid to the creation of light aircraft, designed to perform training tasks, to monitor the Earth's surface, for business flights, etc. The capability to land at an aerodrome of any class, easy flight operation and maintenance, elegance of interior design, make these light aircraft a reasonable choice for business meetings, recreation,