Modelling of the diesel engine in researches of dynamics of machine tractor units

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Summary. The article presents the mathematical description of diesel engines of machine tractor units of different function in researches of their dynamics stated by working out of new technics. It considers the methodical positions of modelling of diesel engines in various problems of dynamics of tractor units. The developed variants of mathematical descriptions of the diesel engine are presented, and the variant choice is defined by problems of concrete research of dynamics of the machine tractor unit.

Key words: machine tractor unit, modelling dynamics, mathematical description of the diesel engine, dynamics research.

INTRODUCTION

Perfection of machine tractor units (MTU) has led to an essential intensification of their working processes, increase of individual capacities, frequently, speeds of movement, load-carrying capacity or tonnage, the main technological parameters, for example, width of capture, and also to a combination of technological process by combination within the limits of one unit of several technological operations, for example, pre-seeding processing of soil, etc. [2, 7, 8, 18, 19, 20, 21]. However all it also has resulted and to high dynamism of such tractor units. Has caused application of the branched out mechanical and hydro-mechanical drives wheels, active working bodies and made active driving wheels on the technological module aggregated with a tractor, essential change of their constructive concept, use of diesel engines with more progressive systems of giving of fuel, wide application of electronics in automatic control systems high-speed and power modes of such diesel engines. The listed changes demand the further consideration [2, 5, 7, 9, 13, 17], for the purpose of development of the theory of perspective tractor units for the proved choice of rational constructive and regime parameters, both the unit as a whole, and its functional subsystems, for example, the engine, a transmission, communications between bridges and communications between transmission wheels, etc.

Creation of tractor technological units on the basis of a wheel tractor with inevitability leads to necessity of the account of dynamic loadings for transmission. It is essential, as engine parameters, inertial and geometrical parameters of the unit, rigidity, a friction and energy dispersion, rotating weights in transmission cause character and levels of fluctuations of moving weights and loadings. Such loadings can surpass many times over static, and therefore they are necessary for knowing correctly to choose settlement values at durability and resource calculations of the projected unit, and also for a correct estimation of a spectrum of defining operational properties of the created machine tractor unit.

In the article the methodical positions of modelling of diesel engines in various problems of dynamics of tractor units are considered.

THE MATHEMATICAL DESCRIPTION OF THE DIESEL ENGINE AT THE DECISION OF PROBLEMS OF DYNAMICS OF MACHINE TRACTOR UNITS

At designing of self-propelled cars it is necessary to model the characteristic of the diesel engine, that, frequently, at a stage of predesign researches, represents some difficulty. We will consider an order of construction of such characteristic on a concrete example of an atmospheric diesel engine with multiphase a regulator of all modes.

Let us accept the following data, for example:

- demanded rated power engine, $P_{e_{nom}} = 58,5$ [kW];
- nominal frequency of rotation of cranked shaft of the engine, $n_{nom} = 2100$ [rpm];
- specific expense of fuel on a nominal mode, $g_e = 222$ [g·(kW·h)$^{-1}$].

Calculations are carried out in the following sequence. The nominal torque of the engine:

$$M_{e_{nom}} = \frac{60P_{e_{nom}}}{2\pi \cdot n_{nom}} = \frac{P_{e_{nom}}}{\omega_{nom}}[\text{kN·m}];$$

$$M_{e_{nom}} = \frac{60 \cdot 58.5 \cdot 1000}{2 \cdot 3.14 \cdot 2100} = 266.2[\text{N·m}].$$

Accepting the factor of torque set μ = 15% that is characteristic for atmospheric diesel engines, we find the maximum torque of the engine:

$$M_{\max} = \frac{100+\mu}{100} \cdot M_{e_{nom}} = \frac{100+15}{100} \cdot 266.2 = 306[\text{N·m}].$$

Factor adaptations on the torque :

$$\chi_P = \frac{M_{\max}}{M_{e_{nom}}} = \frac{306}{266.2} = 1.15.$$

Accepting degree of non-uniformity of a regulator of the tractor engine - δ = 6%, we define the maximum frequency of rotation of a cranked shaft of the engine:

$$n_{\max} = n_{nom} \cdot \frac{200+\delta}{200-\delta} = 2100 \cdot \frac{200+6}{200-6} \approx 2300[\text{rpm}].$$

Let us accept factor adaptations on frequency as

$$\chi_B = \frac{n_{nom}}{n_{\min}} = 1.45.$$

where: $n_{\min}$ - the minimum frequency of steady work of the engine, often accepted at $n_{\min} = n_{M_{\max}}$.

Then values of factors [5] are equal to, in Lejderman's S.R. formulas:

$$a = \frac{\chi_P \cdot \chi_B \cdot (2-\chi_B)-1}{\chi_B \cdot (2-\chi_B)-1}, \text{ a=0,41};$$

$$b = -\frac{2 \cdot \chi_B \cdot (\chi_P - 1)}{\chi_B \cdot (2-\chi_B)-1}, \text{ b=2,15};$$

$$c = \frac{\chi_B^2 \cdot (\chi_P - 1)}{\chi_B \cdot (2-\chi_B)-1}, \text{ with =-1,56}.$$

As check converges: $a+b+c = 1$ factors of the formula of Lejderman's S.R. are defined correctly. Then capacity of the engine at a preset value of frequency of rotation of a cranked shaft is defined from the expression:

$$P_e = P_{e_{nom}} \times \left[a \cdot \frac{n}{n_{\max}} + b \cdot \left(\frac{n}{n_{\max}}\right)^2 - c \cdot \left(\frac{n}{n_{\max}}\right)^3 \right].$$

Using the calculated values of capacity of the engine, we calculate values of the current torque of the engine under the formula:

$$M_e = \frac{60P_e}{2\pi \cdot n}.$$

The hourly expense of fuel at rated power [1]:

$$G_{h_{nom}} = g_{e_{nom}} \cdot P_{e_{nom}} / 1000 = 222 \cdot 58.5 / 1000 = 13[\text{kg·h}^{-1}].$$

We count the hour expense of fuel at the maximum frequency of rotation of a cranked shaft, accepting average for diesel engines of concrete type factor of communication of hourly expenses of fuel of a diesel engine with all regulator modes at rated power and at the maximum turns of the idling, equal k=0,25, then:

$$G_{h_x} = k \cdot G_{h_{nom}} = 0.25 \cdot 13 = 3.25[\text{kg·h}^{-1}].$$

If an argument accepts torque frequency of rotation of a cranked shaft of the engine, hourly and specific effective fuel expenses, at change of the torque from zero to rating value (for a working branch of a regulator), it is possible to define branches of the high-speed characteristic of the engine under formulas:

$$n = n_{\max} - (n_{\max} - n_{nom})\frac{M_e}{M_{e_{nom}}},\ G_h = \left(G_{h_{nom}} - G_{h_x}\right)\frac{M_e n}{M_{e_{nom}} n_{nom}},$$

$$g_h = (g_{h_{nom}} - g_{h_x}) \cdot \frac{M_e \cdot n}{M_{e_{nom}} \cdot n_{nom}},$$

and at change of the torque from nominal to the maximum value (for an external branch) can be defined numerical values of the same sizes on expressions:

$$n = n_{nom}\left(\alpha + (1-\alpha)\sqrt{\frac{M_{e_{\max}} - M_e}{M_{e_{\max}} - M_{e_{nom}}}}\right),$$

$$G_h = G_{h_{nom}}\left[\frac{\gamma - \alpha^2}{1-\alpha}\left(1 - \frac{n}{n_{nom}}\right) + \left(\frac{n}{n_{nom}}\right)^2\right]\frac{M_e n}{M_{e_{nom}} n_{nom}},$$

where: $\alpha = \frac{1}{\chi_B} = \frac{n_{M_{\max}}}{n_{nom}}$ - degree of decrease in frequency of rotation of a cranked shaft of the engine in the field of use of a stock of the torque , we accept 0,7; γ - degree of change of the specific expense of fuel on an external branch of the high-speed characteristic, we accept 1,05.

Capacity of the engine and the specific expense on external characteristic branches are counted under formulas:

Let us carry out calculations on the resulted algorithm at various values of the torque on an engine shaft: from a mode of the maximum torque and to an idling mode. By results of calculations it is possible to construct a working branch of a regulator the loading characteristic - P_e (M_e), n (M_e), G_h (M_e), g_e (M_e) (Fig. 1).

By results of the same calculations it is possible to construct as frequency of rotation of a shaft of the engine the high-speed characteristic of a diesel engine - schedules of dependences $P_e(n)$, $M_e(n)$, $G_h(n)$, $g_e(n)$ (Fig. 2).

Capacity of the engine and the specific expense on an external branch of the characteristic are counted under formulas:

$$P_e = \frac{\pi \cdot n \cdot M_e}{30} \text{ и } g_e = \frac{1000 \cdot G_h}{P_e},\ [\text{g·(kW·h)}^{-1}].$$

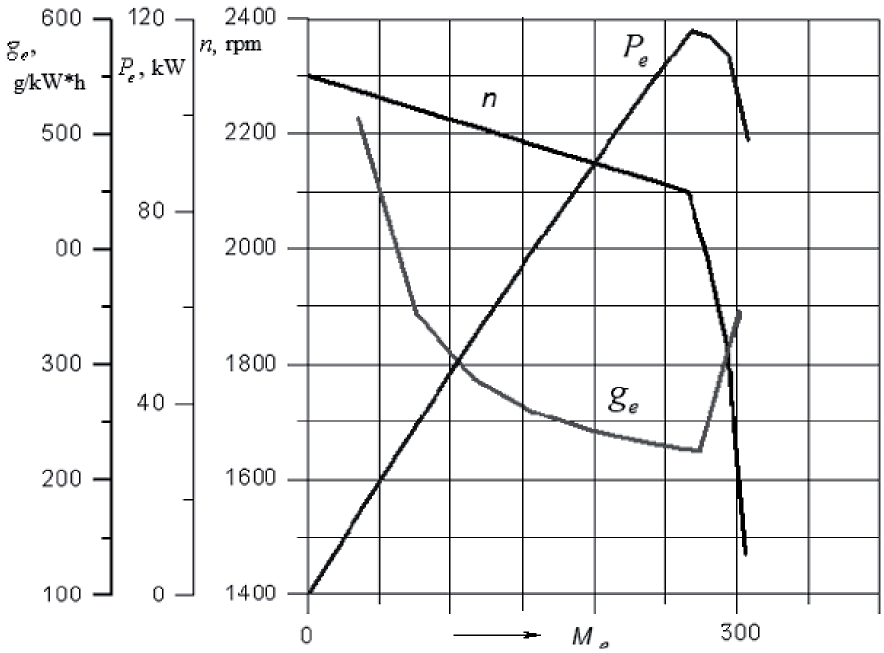

Fig. 1. Settlement of regulatory characteristic of the diesel engine capacity 58,5 kW

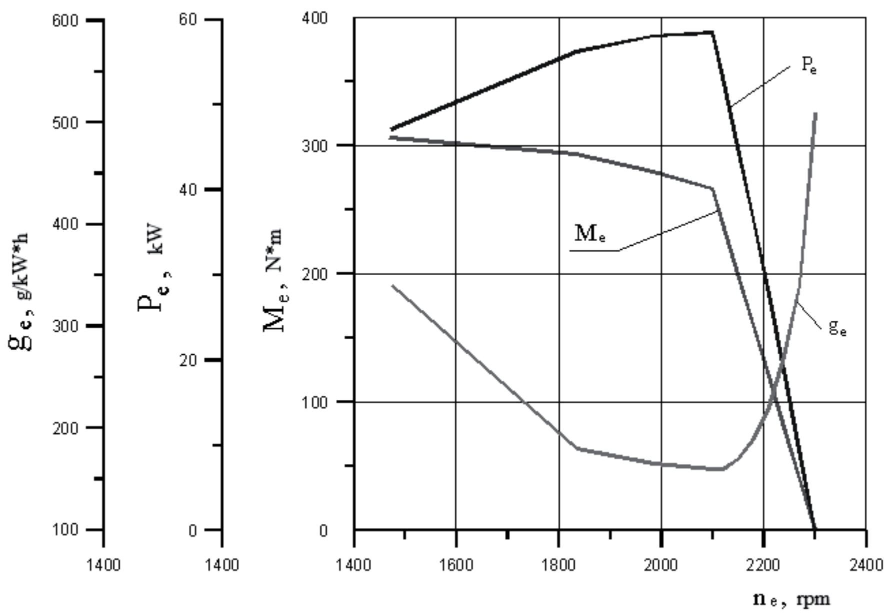

Fig. 2. Settlement of external high-speed characteristic of the diesel engine capacity 58,5 kW

Figures 1 and 2 present characteristics of the diesel engine use at the design static analysis of traction and traction-dynamic properties of a developed tractor or the machine tractor unit [1, 2, 3, 4, 6, 7, 8, 9].

For research of transitive modes of projected tractor units on the developed mathematical models it is also necessary to define the current torque of the engine.

The mathematical model of a tractor diesel engine in this case can be made at an assumption that in transitive modes character of dependence of the moment of the engine from frequency of rotation, positions of a pedal of "gas" and position of the lever of a regulator does not differ essentially from those at equilibrium high-speed modes. Though in the known monograph of academician V.N.Boltinskij "Work of the tractor engine at unsteady loading" data about presence of such influence are cited.

Considering the diesel engine as adjustable system, it represent two dynamic links:

– The first - actually the engine in the form of inertial weight which has as an input angular speed ω and a feedback with a regulator of the fuel pump on its course lever of giving of fuel h_p, and also has an exit - twisting moment M_e;
– The second - all modes a regulator of the fuel pump of a high pressure which has one input - angular speed ω, and the second input - lever position x_p fuel supply whereas an exit is the course lever of giving of fuel h_p.

From the above it follows, that the torque of the engine is defined as function of two variables in the form of $M_e=f(\omega, h_p)$. And though in transitive modes mutual change of arguments of this function leads to instant values of the moment of the engine, differing from static, but, taking into account the accepted assumption, the model of the moment of a diesel engine is accepted not the inertial factorial, describing static high-speed characteristics of the engine at the various fixed positions lever of giving of fuel the fuel pump [7, 8, 10, 11, 12, 15, 20].

As the analytical description of dependence of moment M_e only from angular speed n (Fig. 2), use resulted before expression.

By working out of programs of management by engines, it is useful to use the complex characteristic of the diesel engine (Fig. 3) [14, 16, 18, 20].

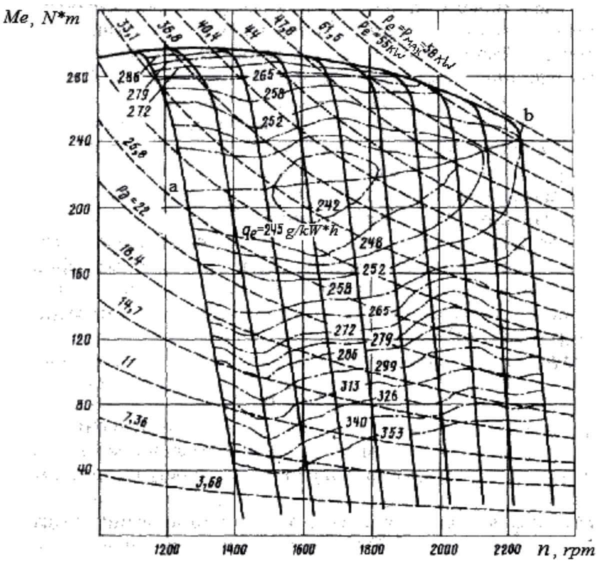

Fig. 3. The complex high-speed characteristic of a tractor diesel engine.

Continuous lines in (Fig. 3) it a working branch of a regulator and external branches of the high-speed characteristic of the diesel engine; shaped - auxiliary lines of constant capacity; strokes-dashed lines - isolines-characteristics of the specific expense of fuel; ab - the characteristic of the minimum specific effective expense of fuel.

Apparently from the analysis of (Fig. 3) resulted expressions do not change and for partial characteristics, only values of parameters $M_{e_{nom}}, \omega_{nom}, \omega$ for partial characteristics (Fig. 3) are functions of position of the lever x_p supply of fuel and are easily defined on the basis of the experimental characteristic whose example for diesel without pressuring the engine is resulted in the specified figure. The given characteristic is convenient at development of strategy of engine management which then is pawned in the form of the program in the microprocessor module of automatic control by the engine.

In (Fig. 4) the model of the tractor diesel engine with two inputs is shown. It is necessary at research few the studied transitive modes of tractor units. For example, the processes connected with incorporated joint automatic control engine-transmission-wheels by installation of a tractor with volume hydro-mechanical transfer. In such a system of engine-transmission-wheels simultaneous influences on the lever of giving of fuel, on corners of turn of cradles of hydro-cars volume hydro-mechanical transfer and on structural transfer of wheels take place, for example at automatic connection of the additional leading bridge.

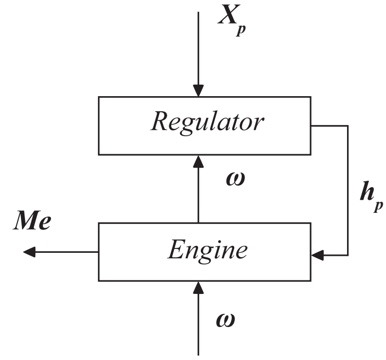

Fig. 4. Model of a tractor diesel engine with all regulator modes

To key parameters of a regulator of the fuel pump, subject to definition as initial data, the following concerns: led at connecting knot weight of moving parts of the fuel pump and a regulator m_p, inertial factor of cargoes A, factor of an internal friction v_p and restoring force E of connecting knot a regulator on the core and partial high-speed modes. The factor depends on position z of connecting knot a regulator, i.e. $A=f(z)$; the factor v_p for modern high-speed diesel engines practically does not depend on relative speed of displacement of rubbing bodies centrifugal of connecting knot a regulator and can be accepted in constant size; force E according to the accepted scheme of model will depend on two sizes and is described by function $E=f(z, x_p)$ [13, 17, 19, 21, 20].

Static balance of a regulator is defined by equality $E=A \cdot \omega_p^2$ (here ω_p - angular speed of a shaft of a regulator).

In transitive modes the equation of a regulator looks as:

$$m_p \cdot \ddot{z} + v_p \cdot \dot{z} + E = A \cdot \omega_p^2.$$

In a general view the factorial model of a diesel engine with all modes a regulator is expressed by the modelling expressions offered by Solonskij A.S. and Vernigor V. A:

$$Me = -C_1 - C_2 \cdot \omega + C_3 \cdot h_p + C_4 \cdot \omega^2 - \\ -C_5 \cdot h_p^2 - C_6 \cdot h_p \cdot \omega^2 + C_7 \cdot \omega \cdot h_p^2, \text{ [N·m]},$$

$$m_p \cdot \ddot{z} + v_p \cdot \dot{z} + E = A \cdot \omega_p^2; \; h_p = z \cdot \frac{R}{r}; \; A = C_8 + C_9 \cdot z, \text{ [N·s}^2\text{·rad}^{-2}\text{]}.$$

Restoring force E:

– on working branches external and partial high-speed characteristics

$$E = E_p = C_{10} - C_{11} \cdot x_p + C_{12} \cdot z - \\ -C_{13} \cdot z \cdot x_p - C_{14} \cdot x_p^2 + C_{15} \cdot z^2, \text{ [N]},$$

– on external branches of the external high-speed characteristic

$$E = E_p = C_{16} + C_{17} \cdot z, \text{ [N]}.$$

The hour expense of fuel is defined from expression

$$G_h = -C_{18} - C_{19} \cdot \omega + C_{20} \cdot h_p + C_{21} \cdot h_p \cdot \omega, \text{ [kg·h}^{-1}\text{]},$$

where: $C_1 \cdot C_{21}$ - constants of modelling expressions which are defined at identification of a tractor diesel engine by a method of adjusted model by results of natural tests of the engine [3, 5, 6, 20].

At moving of the lever of giving of fuel system the engine-regulator works as watching system of automatic control (SAC), and at invariable position of this lever - as automatic regulation system (SAR), realizing a management principle on a deviation.

As follows from stated, working out of full factorial model of the tractor diesel engine represents enough challenge demanding definition of values of factors of model on the basis of natural experiment.

Dynamic properties of the tractor diesel engine as one of objects of management SAC engine-transmission-wheels can be presented also typical non-periodic dynamic a link of the first order [3, 5, 6, 20]:

$$T\frac{d\varphi}{dt} + \varphi = K \cdot \mu - f(t),$$

Where: T - a constant of time of the engine; φ - relative change of angular speed of a shaft of the engine; μ - relative moving he dosing out lever the fuel pump; *To* - factor of strengthening of the engine on a course of regulating body; $f(t)$ - the dimensionless deviation of angular speed caused by change of variable making external loading.

In the absence of fluctuation of resistance $f(t)$ of last expression in the operational form we will receive:

$$(Tp+1)\varphi = K\mu,$$

and it also is the equation non-periodic a link of the first order with transfer function:

$$W(p) = \frac{\varphi(p)}{\mu(p)} = \frac{K}{Tp+1},$$

which values of parameters define dynamic properties of the engine in transitive operating modes.

At equipment of the diesel engine by modern systems of giving of fuel, for example, «Common rail», «Unit Injection» with electronic control by direct injection of fuel in engine cylinders, modelling of the characteristic of the engine depends on the mode of the stabilization supported in given movement by an automatic regulator of the engine. Most often at the established movement on гоне technological agricultural unit, for example, harvest, the mode of stabilization of capacity of the engine, and for the transport unit - a mode of stabilization of speed, or a mode of minimization of the expense of fuel can be provided. At engine modelling are in that case necessary external and partial high-speed characteristics of the engine or the complex characteristic (Fig. 3) and algorithm of work of the most automatic control mean the engine. And, at a choice for a control system, for example, a mode of minimization of the expense of fuel, the block of engine management and transmission the cars working synchronously, those from possible at current loading and speed of movement choose modes of the engine and the transfer relation of transmission, at which the engine mode will come nearer as much as possible close to a line *аб* on the complex characteristic (Fig. 3), it means, that the control system will change also a high-speed mode of the engine, and the transfer relation of transmission [7, 8, 18, 19, 20].

At creation of the mobile car it is necessary to estimate its indicators of dynamics, quality of transients in system engine-transmission-wheels to installation, dynamic loadings of transmission on which judge level of operational properties of the mobile car and efficiency of accepted constructive decisions.

The choice of structure, parameters, laws and algorithms of a control system of the engine, transmission and wheels of the created mobile car, at mathematical modelling of processes in system engine-transmission-wheels is made in a program application by the task of initial data on the blocks concerning corresponding subsystem of system engine-transmission-wheels.

At the task of parameters of the external high-speed characteristic of the diesel engine are used or S.R Lejderman's formulas, or regression polynomial model to nth order, received by processing of experimentally removed characteristics of the engine. For engines with non-monotonic course of curves of capacity and the twisting moment, in case of turbo-supercharging application in diesel engines, for example with several anti-proof readers or at to system of giving of fuel Common Rail, it is used regression polynomial model to 5th order, received by processing of experimentally removed characteristics. The model can be constructed by means of a program application of the mathematical analysis of graphic diagrams AGrapher.

One of problems of modelling of the engine as a part of general model engine-transmission-wheels consists in necessity on each step of integration of the differential equations describing movement of the mobile car, to define a site of a current working point of the engine on its characteristic. The same problem is actual and at creation SAC by transmission if the engine is not equipped by the loading gauge. Therefore the expressions allowing on current values of positions of the lever or a pedal of management by giving of fuel are necessary, to define the current partial high-speed characteristic and twisting moment of a diesel engine on current angular speed of its shaft.

Initial data which are necessary for setting before the beginning of modelling of the diesel engine with all modes a regulator in fuel pump, include the following:

- I_d – The moment of inertia of the engine,
- $M_{\min}$ – The twisting moment on is minimum steady turns of the engine,
- $n_{\min}$ – Is minimum steady frequency of rotation of a shaft of the engine,
- $M_{\max}$ – The maximum twisting moment of the engine,
- $n_{\max}$ – Frequency of rotation of a shaft of the engine at the maximum moment,
- $P_{\max}$ – Maximum engine rated power,
- $n_{p_{\max}}$ – Frequency of rotation of a shaft of the engine at the maximum capacity,
- β_p – Factor of non-uniformity of regulator fuel pump,
- $n_{xx_{\max}}$ – The maximum frequency of rotation of a shaft of the engine idling,
- y – A share from a full speed of the lever of the task of giving of fuel (from 0 to 1),
- z – A share from a full speed of a pedal of gas at the beginning (from 0 to 1).

At lever installation fuel supply not on a maximum, and in some intermediate position y where $0 \le y < 1$, a working branches of a regulator is in parallel displaced towards smaller frequencies of rotation and it becomes insignificant more flat according to dependence for the concrete engine, (Fig. 5).

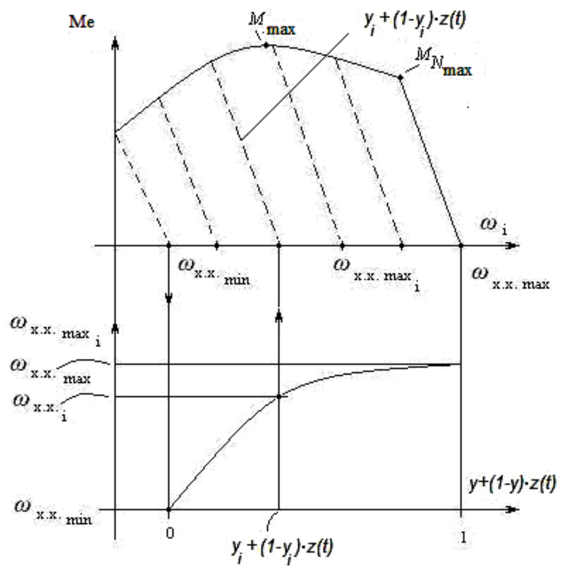

Fig. 5. A working branch of a regulator of the characteristic of a diesel engine at intermediate position of the lever of fuel supply

In (Fig. 6) diesel engine work on various modes at which the minimum expense of fuel (the shaded area) is shown. Here the moments of necessary switching on the higher and lowest transfers after achievement by shaft of the engine of certain angular speed are shown to remain in the shaded zone [1, 18, 20].

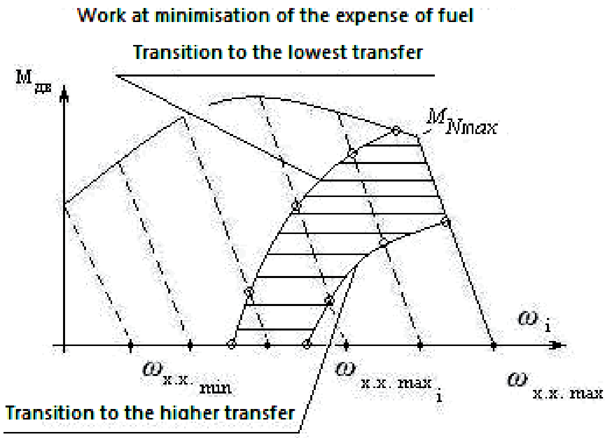

Fig. 6. Area of the minimum expense of fuel

Accepting linear approximation of dependence β_p ($n_{y\max}$), we can write $k_{\beta p}$ - factor of communication of these sizes:

$$k_{\beta p} = \frac{\beta_{Pн} - \beta_P}{n_{xx\max} - n_{\min}} = \frac{\beta_{Py} - \beta_P}{n_{xx\max} - n_{y\min}}. \tag{1}$$

For example, for diesel engines with TNWD with all modes a regulator of universally-agricultural tractors:

$$\beta_P \approx 0{,}12, \quad k_{\beta p} \approx \frac{0{,}18 - 0{,}12}{230 - 150} = \frac{0{,}06}{80} = 0{,}00075.$$

From last expression for factor of non-uniformity of a regulator follows at intermediate position of the lever of the task of giving of fuel y:

$$\beta_{Py} = \beta_P + k_{\beta p} \cdot \left(n_{xx\max} - n_{xx\ y\max}\right). \tag{2}$$

At diesel engine work on transitive modes the effective twisting moment of the engine decreases in comparison with the high-speed characteristic that is considered by factor γ_m then

$$M_{\partial i} = M_{\partial cmi} \cdot \left(1 - \gamma_m \cdot \frac{d\omega_{di}}{dt}\right) = M_{\partial cmi} \cdot \left(1 - \gamma_m \cdot \varepsilon_{di}\right), \tag{3}$$

where: ε_{di} - angular acceleration of a shaft of the engine.

At the set position of the lever of giving of fuel y it is defined corresponding value of frequency of idling:

$n_{xx\ y\max} = n_{\min} + y \cdot \left(n_{xx\max} - n_{\min}\right)$; And angular speed is equal $\omega_{xx\ y\max} = \dfrac{\pi \cdot n_{xx\max}}{30}$. (4)

The factor k_p, a characterizing corner of an inclination flowing working branches, is equal:

$$k_p = \frac{2 \cdot M_{P\max}}{\left[\beta_P + k_{\beta P} \cdot \left(n_{xx\max} - n_{xx\,y\max}\right)\right] \cdot \left(n_{xx\max} + n_{P\max}\right)}. \quad (5)$$

As according to earlier resulted formulas on an external branch of the characteristic

$$M_{y\max} = M_{P\max} \cdot \left[a + b \cdot \frac{n_{y\max}}{n_{P\max}} + c \cdot \left(\frac{n_{y\max}}{n_{P\max}}\right)^2\right],$$

then for partial working branches $M_{di} = k_p \cdot (n_{xy\max} - n_{di})$, and current capacity of engine

$$P_{di} = M_{di} \cdot \omega_{di}. \quad (6)$$

At diesel engine functioned on partial working branches $n_{xymax} \geq n_{di} \geq n_{ymax}$. We will find n_{xymax}, considering, that the point (M_{ymax}, n_{ymax}) simultaneously belongs also to an external branch and partial a working branches of regulator of the high-speed characteristic. Then the following record is fair:

$$a \cdot M_{P\max} + b \cdot M_{P\max} \cdot \frac{n_{y\max}}{n_{P\max}} + c \cdot M_{P\max} \cdot \left(\frac{n_{y\max}}{n_{P\max}}\right)^2 =$$
$$= k_p \cdot n_{xy\max} - k_p \cdot n_{y\max}. \quad (7)$$

If to accept an assumption that $k_p = const$, then, having designated

$$c \cdot \frac{M_{P\max}}{n^2_{P\max}} = A; \quad \left(k_p + b \cdot \frac{M_{P\max}}{n_{p\max}}\right) = B;$$
$$\left(a \cdot M_{P\max} - k_p \cdot n_{xy\max}\right) = C. \quad (8)$$

From expression (7) we will receive:

$$A \cdot n^2_{y\max} + B \cdot n_{y\max} + C = 0 \text{ Or } n^2_{y\max} + \frac{B}{A} \cdot n_{y\max} + \frac{C}{A} = 0. \quad (9)$$

The decision of the received quadratic (9) will look like:

$$n_{y\max} = -\frac{B}{2 \cdot A} - \sqrt{\frac{B^2 - 4 \cdot A \cdot C}{4 \cdot A^2}}. \quad (10)$$

Knowing n_{ymax}, we define M_{di} and P_{di} on corresponding expressions for partial working branches of the high-speed characteristic.

CONLUSIONS

At management of a foot pedal of fuel supply *z (t)* and at the position task *y* a control lever of supply of fuel calculation is conducted in the following sequence. If moving of a pedal of giving of fuel submits to some law *z (t)* in time or z $(\omega_{\partial i})$ as current angular speed of a cranked shaft of a diesel engine, it means, that to value of position of the lever *y* some share from the remained course of the lever *(1-) z*, fuel supply pressed against the stop. In the latter case at = at + (1-).1 = 1, that corresponds to the external high-speed characteristic of a diesel engine. In this case on each step of integration new value *at = at + (1-).z (t)* pays off and earlier received expressions operate.

At modelling of movement of the mobile car on each step of integration of the differential equations describing dynamics system of engine-transmission-wheels, it is necessary to define "site" of a working point of a diesel engine, that is:

– the equation flowing working branches on *at (t)* and *z (t)*,
– value of current torque $M_{\partial i}$ on current frequency of rotation of a shaft of the engine $n_{\partial i}$ or its angular speed $\omega_{\partial i}$.

Thus, the developed variants of mathematical descriptions of the diesel engine are presented, and the variant choice is defined by problems of concrete research of dynamics of the machine tractor unit.

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MODELOWANIE SILNIKA WYSOKOPRĘŻNEGO W BADANIACH DYNAMIKI MASZYNOWYCH AGREGATÓW CIĄGNIKOWYCH

Streszczenie. W publikacji przedstawiono matematyczny opis silnika wysokoprężnego maszynowych agregatów ciągnikowych szerokiego przeznaczenia w badaniach ich dynamiki przy opracowywaniu nowych środków technicznych. W artykule rozpatrzono podstawy metodyczne modelowania silników wysokoprężnych w szerokim aspekcie dynamiki agregatów ciągnikowych. Przedstawiono opracowane warianty matematycznych modeli silnika wysokoprężnego. Wybór wariantu określono zadaniami i celem konkretnych badań dynamiki maszynowego agregatu ciągnikowego.

Słowa kluczowe: maszynowy agregat ciągnikowy, modelowanie dynamiki, matematyczny opis silnika wysokoprężnego, badanie dynamiki.