Optimization of Insensitivity Rate of Speed Control Systems

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Abstract. The present work is aimed at developing a method for estimating the optimal insensitivity parameters of the automatic speed control systems. When operating vehicles, unsteady loads applied on the engine are perceptible, which leads to the decrease in power and efficiency of the engine also reasoned by the existence of dead zone of the speed control system. Reducing the degree of insensitivity of the regulator positively influences the characteristics of regulatory processes, yet the significant reduction causes a decrease in the stability of the system. There is a problem of assessing the need to respond not only to external influences, but also to intra-cycle changes in internal combustion engines, leading to a change in rotational speed. In case of using modern microprocessor systems, the discreteness of receiving and processing initial information greatly influences the regulatory process. In order to optimize the insensitivity rate of regulators, it is proposed to estimate the allowable reduction of the dead zone until self-induced vibrations and resonance phenomena occur. An expression for determining the frequency of self-induced vibrations in control systems causing resonance is defined. The implementation of the proposed method will improve the efficiency of operation and increase the service life of transport-and-technological machines.

Keywords: Internal combustion engine · Speed controller · Insensitivity rate

1 Introduction

Rolling stock of railway transport is operated under conditions of continuously changing load on engine due to constant changes in the track grading and speed. The constant change in load characterizes not only land transport and technological machines, but also ship power plants. The fluctuations of the angular velocity caused by change in the moment of resistance of the external load are summed with the oscillations arising due to the cyclical nature of the internal combustion engine's working process. As a result, the operation of an engine under unsteady modes leads to a decrease in its power and efficiency. Firstly, the filling ratio of engine cylinders, the indicator efficiency and the excess air ratio are reduced. Secondly, in case of using diesel engines equipped with all-frequency speed controllers as a power source, phase shifts occur between the output and input coordinates of the control system [1].

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These phase shifts consist in the lag of the rotational speed in phase from the change in the moment of resistance and, accordingly, in the lag of the engine torque from the movement of the executive body of the change in fuel supply. The movement of the fuel control body begins with a certain shift in time when the resistance moment changes due to the presence of the regulator's dead zone. Consequently, the beginning of the change in the effective torque of the engine in relation to the beginning of the change in the moment of resistance occurs with a delay that is in some cases a significant value. These factors lead to an incomplete use of power and increase in the specific fuel consumption of a power plant.

The size of the dead zone of an automatic control system is influenced by frictional and backlash forces in kinematic pairs of mechanical elements of regulators, gaps in contacts of electrical control systems, the viscosity of working fluid in hydraulic control systems, the settings of electronic control systems and a plenty of other reasons. The engine itself is also an element of dead time between the moment of fuel flash in the cylinder and the production of net torque. The duration of dead time is determined, among other things, by the angular velocity of a crankshaft and a number of cylinders [2]. It is obvious that reducing the insensitivity rate of the controller positively affects the dynamic and static characteristics of regulatory processes due to the decrease in the amplitude and period of self-oscillations in the system. At the same time, the significant insensitivity rate reduction leads to the probability of arising of high-frequency oscillations and self-oscillations that cause the decrease in stability of control system. For each particular regulation object under given boundary conditions, there is the only duration of the transition process determined by speed of the regulation system that implies the minimal fuel consumption of a power plant [3].

Therefore, in order to improve the efficiency of vehicle operation, it is necessary to optimize the insensitivity parameters of speed controllers and develop measures to implement an additional effect on the controlled value depending on the value of the derivative of the input signal [4]. In modern conditions, the management of fuel supply and speed control is carried out by microprocessor devices, the main advantage of which is the possibility of implementing complex control laws. However, in such control systems, there is also a delay in the receiving and processing of control signal, which significantly affects the quality of control. At the same time, the noises of electronic devices of various origins that occur during operation introduce some signs of randomness into the control process and inconsistency in the operation of the system links. Moreover, in this case, one of the major problems is the problem of assessing the need for the automatic speed control system's responding not only to external influences, but also to cyclic changes in the angular velocity of a crankshaft, caused, for example, by ignition of fuel in the engine cylinders. The accuracy of regulation depends on the frequency and way of processing the initial information on the current value of rotation speed.

One of the modern types of control of fuel supply processes in automatic speed control systems is the process of controlling the position of the fuel pump rail via pulse-width modulation (PWM). The signal generated by the PWM controller is a discrete sequence of rectangular pulses characterized by amplitude, frequency, and duration of the signal. Figures 1 and 2 show the processes of regulating the rotational speed of the crankshaft n for a time t during a load shedding transition process and when the engine is running in stationary mode.

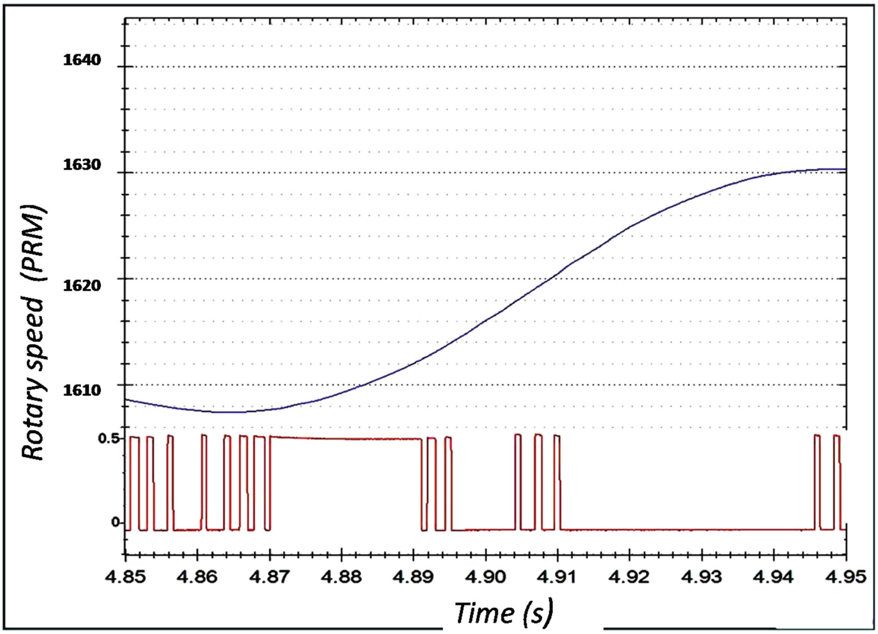

Fig. 1. The process of regulating the rotational speed of a diesel engine equipped with a PWM speed controller in the transition process of load shedding

In order to implement all capabilities of such systems, it is necessary to ensure optimal discretization in signal processing, which determines the accuracy of restoring the original analog signal on the rotational speed value at a specific point in time. The shorter the sampling interval, the more accurate the measurement process is, however, the complexity and cost of the recording equipment increase significantly. Digital signal processing methods fully allow evaluating the technical condition and interaction of individual elements of complex mechanical systems [6]. However, since they are based on restoration of a continuous function from its discrete values, the problem of objectivity and timeliness of obtaining information arises in case of using electronic control systems.

According to one of the interpretations of the Kotelnikov theorem, random signal $u(t)$ with an oscillation spectrum limited by a certain frequency value F_B can be fully restored only by following sample values with a time interval determined by the following expression:

$$\Delta t = \frac{1}{2F_B} \tag{1}$$

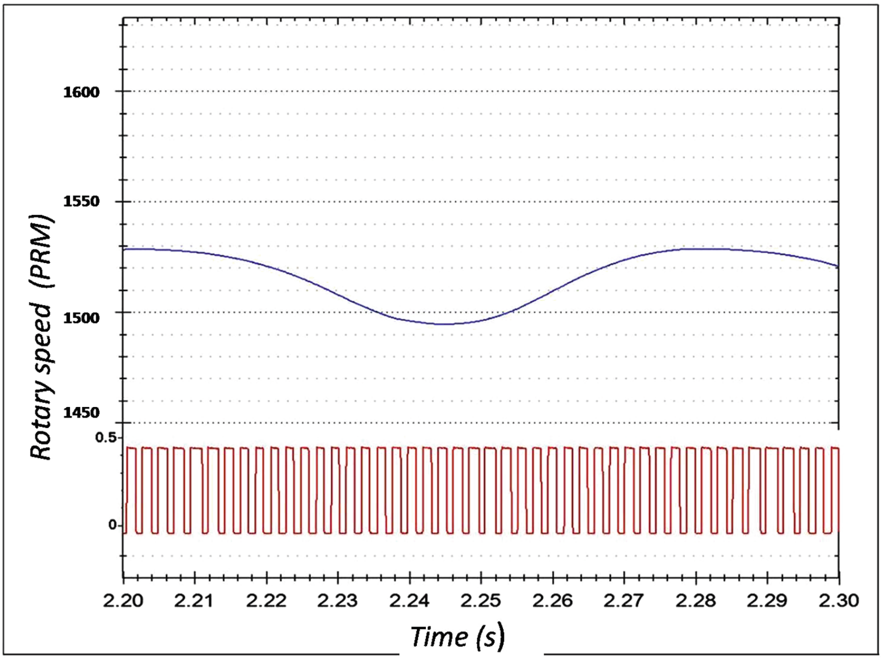

Fig. 2. The process of regulating the speed of diesel engine equipped with PWM speed controller, when working in steady state mode

Proceeding from the above, the wrong conclusion can be made that any harmonic signal with frequency F_B can be represented as a sequence of indications with frequency corresponding to obtaining two measurements of the signal for a period. However, it is impossible even theoretically to restore the original harmonic signals by two indications over a period. Therefore, in practice the restoration of signals with a wide range of oscillations even with the use of Fourier transform algorithms is a difficult task, becoming almost unmanageable in case when the maximum frequency in the signal exceeds half the sampling frequency.

Solving the problem of actually objective signal processing is possible when using the Hilbert transform, the physical meaning of which is to rotate the initial phases of all components of the spectrum of the initial signal $s(t)$ by the same angle $-0,5\pi$. This turn equals to the arising of a common factor $\exp(-i\pi/2)$. Instrumental processing of discrete signals using a phase shifter could help solve the problem of optimizing the degree of insensitivity of the regulator, but the use of this method is hard to implement in mass production and operation of vehicles. Consequently, there is a need for methods to optimize parameters of control systems, that would not depend on the discretization of control signals, if possible.

2 Research Methods

To solve the problem of optimizing the value of the insensitivity rate of rotational speed controllers, it is necessary to determine the criteria for ensuring the highest efficiency of the regulatory system, which means adaptation to achieving the set goal. In this case, it seems preferable to use multiplicative criteria for which the objective function takes the following form, where the plus sign implies restrictions that require the maximum increase of the function, and the minus sign means restrictions that require minimization of the function:

$$f(x) = \frac{\prod_{j=1}^{g} y_j^+(x)}{\prod_{i=j+1}^{m} y_i^-(x)} \tag{2}$$

In this case, as a positive limitation, one can take a response that provides the required gain of the system, that is, the ratio of the amplitude of oscillations of the controlled parameter φ and the amplitude of the disturbing action $f(t) = A \sin \omega t$.

As proved by studies in the field of theory of oscillations and vibration insulation of heat engines [8, 9], friction forces do not significantly affect the amplitudes of nonlinear oscillations, therefore, for example, when analyzing the characteristics of control systems with mechanical speed controllers, friction forces in the controller mechanism and in the rack drive The fuel pump can be neglected. This is all the more likely in the case of microprocessor-based automatic control systems where there are no mechanical moving parts and friction units.

However, it should be noted that external influences on the regulator and the fuel supply control unit of a diesel internal combustion engine contain several harmonic and possibly subharmonic components of a complex spectral composition. With such polyharmonic excitation, the zones of instability increase substantially as the individual elements of the system move. Therefore, as negative constraints in expression (2), which describes the essence of the multiplicative criterion, one can use friction forces that are still objectively present in the regulator. If the rotational speed is controlled by electronic systems, which, like mechanical devices, have their own oscillation frequency, then the control signal lag can be represented as an analogue of the friction force. Thus, as a criterion for optimizing the magnitude of the degree of insensitivity of the regulatory system, it is proposed to use the permissible degree of reduction of the dead zone until the moment of possible self-oscillation and resonance phenomena occurs. The effect of minimizing value ε_P on the behavior of speed control systems is currently not well understood. Based on the well-known theoretical principles of the theory of mechanical oscillations, we present the mechanism of the fuel supply control body of a power plant in the form of an elastic oscillatory system with a natural frequency of oscillations consisting of a regulator with its own conventional stiffness Cr and a drive of the executive control body of the fuel supply with mass m and stiffness cd. Hardness in this case should be considered an analogue of the level of resistance to movement of the controls of the fuel supply due to the presence of internal friction, that is, an analogue of the degree of insensitivity. If, in order to simplify the calculations, we assume the law of the steady-state motion of the mechanism of the fuel supply control organ linear, then we can assume that it is sinusoidal in nature:

$$f(t) = Z_0 \cdot \sin \omega t \tag{3}$$

Where z –oscillation amplitude of the fuel supply control drive; ω –circular oscillation speed.

3 Results of the Research

Differentiating the law of change of the disturbing force on the coordinate of displacement z, we obtain the following expression to describe the oscillatory process of the rail of the high-pressure fuel pump:

$$m\ddot{z} + (C_d - C_r) \cdot z = Z_0 \sin \omega t \tag{4}$$

Introduce the designation of natural oscillations of the vibrating object:

$$v = \sqrt{\frac{C_d - C_r}{m}} \tag{5}$$

Then the expression (4) takes the following form:

$$\ddot{z} + v^2 z = \frac{Z_0}{m} \sin \omega t \tag{6}$$

Using known methods for solving such equations, the following expression is obtained for the amplitude of the steady state of forced oscillations:

$$z = \frac{Z_0}{C_d - C_r} \cdot \frac{1}{1 - \omega^2/v^2} \tag{7}$$

The natural oscillations of a vibrating object can be represented as:

$$v^2 = \frac{C_d - C_r}{m} = v_d^2 (1 - \frac{C_r}{C_d}) \tag{8}$$

where v_d^2 –natural oscillations of the fuel control drive.

Taking into account the expression (8), the amplitude of the forced oscillations of the drive of the fuel control body without friction forces will be determined as follows:

$$z = \frac{Z_0}{C_d} \cdot \frac{1}{1 - \frac{C_r}{C_d}} \cdot \frac{1}{1 - \left(\frac{\omega^2}{v_d^2}\right) \cdot \left(1 - \left[\frac{C_r}{C_d}\right]\right)} \tag{9}$$

In this expression, the multiplier $\frac{Z_0}{C_d}$ is the movement caused by the maximum perturbing force, when it is applied statically, while the other two factors determine the dynamic effect of this force. Their absolute values can be considered the dynamic coefficient of the system:

$$K_g = \frac{z}{Z_0} = \frac{1}{\left(1 - \frac{C_r}{C_d}\right) - \frac{\omega^2}{\nu_n^2}} \tag{10}$$

Figure 3 shows the change in the dynamic coefficient K_g depending on the circular oscillation frequency ω for different values of the conditional stiffness coefficient of the $C_r$controller. When $C_r = 0$, (when conditional elasticity is absent (resistance to movement of the regulator)), the forced oscillations in the system are similar to the oscillations of the load on the spring. If the ratio C_r/C_d reaches1, the natural oscillation frequency decreases, leading to a shift of the amplitude-frequency characteristic of the oscillations to the left along the axis of the abscissas and, consequently to a decrease in the range of those frequency values ω at which $K_g \rangle 1$. When the frequency of the perturbing force approaches the frequency of free oscillations of the system, then the value of the dynamic coefficient rapidly increases.

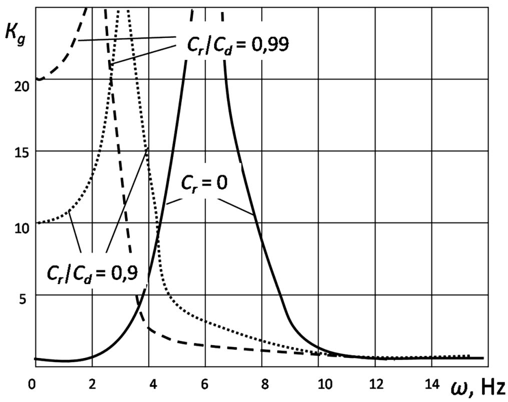

Fig. 3. Influence of conditional stiffness of the regulator on the value of the dynamic coefficient

The value of the dynamic coefficient K_g reaches its maximum under the following conditions:

$$\frac{\omega}{\nu_d} = \sqrt{1 - \frac{C_r}{C_d}} \tag{11}$$

The minimum values of ω, at which K_g values are less than 1, always lie in the super-resonance area, or in other words in the area where resonance is no longer possible. These values are smaller, when the total system stiffness $C_\Sigma = C_d - C_r$ gets smaller values. When the system has zero stiffness, $|C_d| = |C_r|$, then Eq. (10) takes the following form:

$$K_g = \frac{v_d^2}{\omega^2} \tag{12}$$

The obtained expressions show that when the condition $|C_d| = |C_r|$ is met, the system does not have a natural frequency of oscillation, and the resonant peak is located on the ordinate axis. Therefore, in this case oscillations always occur in the super-resonance area.

Now suppose that in the oscillatory system considered above there are forces of dry friction. In this case, the expression (4) looks as follows:

$$m\ddot{z} + 2fsCsign\ddot{z} + (C_d - C_r)z = Z_0 \sin \omega t \tag{13}$$

Where f – sliding friction coefficient; s – contact surface area.

Equation (15) is the usual equation of forced vibrations with dry friction and allows determining the values of amplitude Z_0 and phase shiftφ:

$$z = \frac{Z_0}{C_d - C_r} \cdot \frac{\sqrt{1 - \frac{4 \cdot 2frC_r}{\pi \cdot Z_0}}}{1 - \frac{m\omega^2}{C_d - C_r}} \tag{14}$$

$$tg\varphi = \frac{1}{\sqrt{\left(\frac{\pi \cdot Z_0}{4 \cdot 2frC_r}\right)^2 - 2}} sign\left(1 - \frac{m\omega^2}{C_d - C_r}\right) \tag{15}$$

Using expression (14) and introducing the concept of relative friction coefficient $\gamma = \frac{2fs}{Z_0}$, the expression is obtained for determining the amplitude of oscillations in the control system, taking into account the presence of friction forces:

$$z = Z_0 \cdot \frac{\sqrt{1 - \frac{16}{\pi^2}\gamma^2 \cdot \left(\frac{C_r}{C_d}\right)^2 \cdot \left(\frac{v_d^2}{\omega^2}\right)^2}}{\frac{v_d^2}{\omega^2}\left(1 - \frac{C_r}{C_d}\right) - 1} \tag{16}$$

The dynamic coefficient in this case equals to:

$$K_g = \frac{z}{Z_0} = \frac{\sqrt{1 - \frac{16}{\pi^2}\gamma^2 \cdot \left(\frac{C_r}{C_d}\right)^2 \cdot \left(\frac{v_d^2}{\omega^2}\right)^2}}{\frac{v_d^2}{\omega^2}\left(1 - \frac{C_r}{C_d}\right) - 1} \tag{17}$$

When $f = 0$, then the expression for determination of K_g will take the form:

$$K_g = \frac{1}{\frac{v_d^2}{\omega^2}\left(1 - \frac{C_r}{C_d}\right) - 1} \tag{18}$$

Comparing expressions (18) and (10), it can be concluded that they are identical, but having the difference consisting in the referring of Eq. (18) to a different base value. This expression takes into account the change in the amplitude of the perturbing force with a change in frequency. As the ratio C_r/C_d approaches 1, the amplitude-frequency characteristic of the control system shifts towards smaller values of ω, thus the resonance zone expands. In addition, at the initial values of frequency of the perturbing force, a transition through resonance occurs, but only minor amplitudes of oscillations develop, since there is not enough energy available to significantly shake the system. If the conventional values of the controller stiffness and drive rigidity are equal in values, then the dynamic coefficient equals 1 and there are no resonance phenomena in the system. At the same time, the oscillations remain constant throughout the entire frequency range and will not exceed the amplitude values of Z_0.

Analysis of the curves presented in Fig. 4 and characterizing the expression (18) for different values of ω, C_r/C_d and *f*, shows that friction forces corresponding to the presence of a dead zone in a controller reduces the resonant peaks of oscillations and allows the system to remain in a "locked" state when a certain frequency of resonant oscillations is reached.

Non-stop self-oscillations begin with a frequency, called the breakaway frequency (the beginning of the resonance), and is determined by the formula:

$$\omega = \sqrt{\frac{4}{\pi}\gamma \frac{C_r}{C_d} v_d^2} \tag{19}$$

Therefore, the limiting value of the relative coefficient of friction is the value determined by the expression:

$$\gamma = \frac{\omega^2 \pi C_d}{4 C_r v_d^2} \tag{20}$$

4 Discussion

The results of calculations and theoretical studies show that for the stability of a control system and the absence of resonant oscillations, the resistance to movement of the regulator's mechanisms or, in other words, the conventional stiffness of the control action should be more than or equal to the conditional movement stiffness of the fuel delivery control (regulatory action). In particular, it is shown that self-oscillations in such systems leading to resonance phenomena, begin with a frequency determined by the formula $\omega = \sqrt{(4/\pi)\cdot\gamma \cdot (C_p/C_n) \cdot v_n^2}$. The analysis of frequency characteristics allows optimizing or changing the sensitivity parameters of automatic control systems, which, in case of using digital control systems, is preferable compared to control of the sampling processes of monitored values.

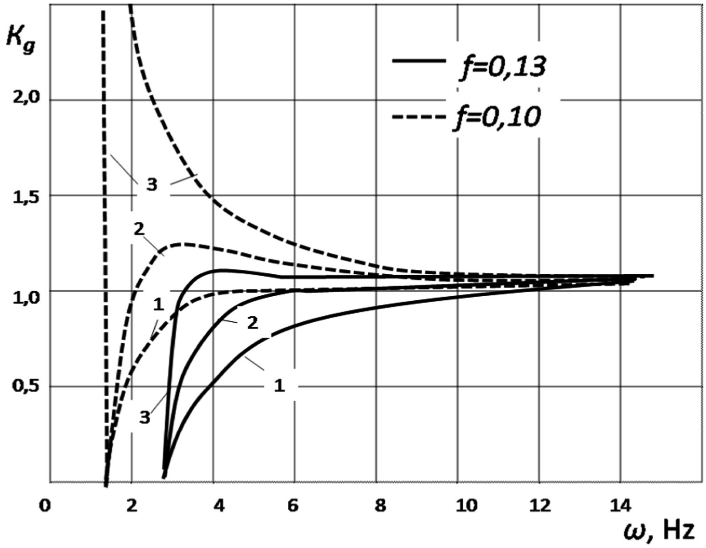

Fig. 4. The influence of friction on the dynamic coefficient for different values of ratio C_r/C_d: 1,0 - curves 1; 0.95 - curves 2; 0.9 - curves 3

In case of applying the proposed method of estimating the appropriate parameters of insensitivity during the design and adjustment of the controllers, the opportunity arises to apply of the most reasonable response to a change in rotational speed during the movement of vehicles and the performance of technological operations. As a result, operational efficiency can be significantly increased and the service life of transport and technological machines can be extended.

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