268. Concept and Methods of Adaptive Vibration Protection and Stabilization

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Abstract

The concept, the general formulation of problems and the algorithms of adaptive real-time control of multivariate vibration protection and stabilization of multi-axis chassis body position in space, which in the general case provide in general case the solution of the problems of oscillations moderation and motion stability, are considered.

Keywords: Adaptive control, Vibration protection, Adaptive stabilization, Adaptive regulator, Stability, Multi-axis machines

1. Introduction

Theoretical and applied fundamentals of the active vibration protection and stabilization are considered by K.M. Ragulskis [1], M.D. Genkin [3], M.Z. Kolovsky [4], V.B. Larin [6], D. Rujechca [7], L.A. Ribak [8], A.V. Sinev [8], K.V. Frolov [9, 10], F.A. Furman [10], R.I. Fourounjiev [11-25], A.F. Opejko [15-20], A.L. Homich [22-25], V.D. Sharapov [26], J.T. Whether, D.L. Mejry, V.G. Rosler and many others.

The further essential improvement of quality of these systems is obtained based on concepts of the inverse problems of controlled system’s dynamics [2, 5] and adaptive regulators of a new generation [12-14], which combine accuracy and high speed without overshoot. The original approach and methods provide in a general case adaptive vibration protection and stabilization in real time, taking into account the motion stability.

2. Mathematical models

The position in space of the case of the multi-axis machine with an essentially nonlinear passive suspension, functioning together with an active suspension, is characterized by the phase coordinates z, φ, ψ (the vertical displacement of the center of masses, the angle of pitch and the angle of roll of the case of the multi-axis (?) machine), for which the command values z_S, φ_S, ψ_S are assigned. The motion equations of the controlled system are supposed to be known with accuracy to the control vector $u = (u_1, ..., u_m)^T$:

$$\ddot{x} = F(\dot{x}, x, u), \tag{1}$$

$$t \geq t_o : x(t_o) = x_o, \ \dot{x}(t_o) = \dot{x}_o ,$$

where $F = (F_1, ..., F_n)^T$ is a vector of the right parts of the system; $x = (x_1, ..., x_n)^T$ is a vector of the generalized coordinates; $\dot{x} = (\dot{x}_1, ..., \dot{x}_n)^T$ is a vector of the generalized speeds; $x_o = (x_{1o}, ..., x_{no})^T$ is a vector of an initial condition; $\dot{x}_0 = (\dot{x}_{01}, ..., \dot{x}_{0n})^T$ is a vector of the initial speeds; T is the symbol of transposing.

In the system (1) the motion equations of the electro hydraulic drives are considered in the following form:

$$\dot{q}_5 = \varphi_5(q_5, u, q_6);$$

$$\dot{q}_6 = \varphi_6(q_6, q_5, q_2);$$

$$\dot{q}_7 = \varphi_7(q_7, \dot{q}_2);$$

$$\dot{q}_8 = \varphi_8(q_8, \dot{q}_4);$$

$$t \geq t_0 : q_i(t_0) = q_{0i}, \ i = 5, ..., 8,$$

where $\varphi_5 - \varphi_8$ are known vector operators.

Servo-drives can be installed on all the supports, or only on the required ones (for example, only on the side supports).

There are unified quality criteria for all the drives, which are presented in the form of desirable motion properties, of formulation of the problems of control and restriction, as well as in the form of adaptive control algorithms, which parameters, however, vary for the drives of the various supports. For the chassis with pneumatic elastic elements it is natural to control the position of the plunger of the pneumatic elastic element by controlling the submission of the liquid in the cavities above and under the plunger.

In the mathematical model of the spatial oscillations of the machine (1), the virtual elastic elements (led to the supports) are examined. At this, for an active suspension the deformation of the $il-$th support, which is led to the wheel/roller, is defined by the expression:

$$\Delta_{il}(t)=y_{il}(t)-z_{il}(t)+q_{6il}(t) \quad (i=\overline{1,n};\ l=1,2), \qquad (2)$$

Where $z_{il}(t)=z(t)+l_i\varphi(t)+b_{il}\psi(t)$; y_{il} are the displacements of the non-amortized masses; q_{6il} are plungers' displacements; l_i, b_{il} are geometrical parameters.

The virtual displacements of the plungers of the actuating servo-cylinders have the following meaning:

$$q_{6il}=\beta_{il}q_{6fil},$$

where β_{il} is a transfer function of reduction of the physical characteristic to the virtual: $\beta_{il}=\beta_{il}(t)$; q_{6fil} are the displacements of the plungers of the physical servo drives. The virtual variables are recalculated into the physical variables (displacement, speeds and accelerations on the real drive), taking into account the factors of reduction, while designing.

3. Observable variables

The accelerations of the amortized and non-amortized masses and the actuating mechanisms plungers' displacements can be considered as the observable variables. In the other variant, the accelerations of the amortized mass, actuating mechanisms plungers' displacements and dynamic loadings on the road (or their derivatives) can be observed. All the other variables, used by the intellectual regulators, can be identified.

Let the accelerations of the amortized mass above the corresponding supports $\ddot{z}_{il}$ $(i=\overline{1,n};\ l=1,2)$, accelerations of the non-amortized masses $\ddot{y}_{il}$ $(i=\overline{1,n};\ l=1,2)$ and actuating mechanisms plungers displacements q_{6il} $(i=\overline{1,n};\ l=1,2)$ be observable. In this case, $4n$ accelerations sensors and $2n$ displacements sensors are required. Values, measured by sensors, are estimations of the observable variables. The variables, missing for control formation, can be identified later on.

The desirable condition of the case of the machine is assigned by the command values of the variables z_s, φ_s, ψ_s. The command values of the control variables for each channel of control are calculated according to the formula:

$$\bar{z}_{il}=z_s+l_i\varphi_s+b_{il}\psi_s \quad (i=\overline{1,n};\ l=1,2). \qquad (3)$$

Apparently, the command values of the control variables for each support in a multivariate control system represent function of the assigned parameters of stabilization of the case of the machine z_s, φ_s, ψ_s and also of the parameters l_i, b_{il}, which describe the servo drives position, referring to the center of masses of the vehicle.

The quality criteria for each of the drives differ by the command values (3) for each of them, by the current values of the phase variables, and also by the control parameters. Regulators generate control signals for each servo-drive while functioning:

$$u_{il}=\Phi(z_s,\varphi_s,\psi_s,\ddot{z}_{il},f) \quad (i=\overline{1,n};\ l=1,2), \qquad (4)$$

where $\Phi(.)$ is a known operator [14-16]. Each servo-drive controls the condition in space of the virtual mass m_{il}, which is a part of the general amortized mass, connected to the il-th support.

An active suspension of the chassis at each moment of time must provide the following:

- minimization of the oscillations of the amortized mass,
- stabilization of the dynamic loadings on the controlled wheels, in relation to the corresponding static loadings.

4. Conditions of stability

Besides the general conditions of motion stability, according to Lyapunov, which are provided at synthesis of the control functions, the requirement of stabilization of dynamic loadings on control wheels, in relation to the corresponding static loadings, adequate to the requirement of minimization of the take off of the controlled wheels of the machine from the ground, corresponds to the supplying of the machine with the motion stability.

In the problem of active moderation of the oscillations, the output variable is the vertical displacement of the amortized mass $x(t)$, and in the problem of stability the output variable is the vertical loadings on controlled wheels of the machine $P_{il}(t)=P_{4il}(t)+P_{5il}(t)$.

The take off of the tires from the ground occurs at the performance of the following condition:

$$P_{4il}(t)\le-\bar{P}_{4i}=-(m_{1il}+m_{2il})g \qquad (5)$$

or

$$\delta_{il}(t)\le\delta_{si}=(m_{1il}+m_{2il})g/C_{si},$$

where $\delta_{il}(t)=q_{il}(t)-y_{il}(t)$; δ_{si} is a static deformation of the $i-$th tires; $q_{il}(t)$ is the kinematic perturbation; C_s is the rigidity of the $i-$th tire, which corresponds to the small oscillations.

As fluctuations are viewed concerning the position of static equilibrium of system, both problems belong to a class of vibration protection tasks, when the command value of an output variable is assigned equal to zero, and to the stabilization tasks if the output variable is not equal to zero.

The algorithm of the solution of a stability problem of controlled wheels is defined by the properties of the used sensors. We will consider two alternatives.

1. The accelerations of the amortized and non-amortized masses are observed (measured) and the current values of dynamic loadings $P_{il}(t) = m_{1il}\ddot{x}_{il} + m_{2il}\ddot{y}_{il}$ are calculated. Then the derivative is defined by differentiation: $\dot{P}(t)=m_1\dddot{x}+m_2\dddot{y}$

2. The dynamic loadings within the contact of a tire with the road are observed: $P_{il}(t)=P_{4il}(t)+P_{5il}(t)$. After differentiation we receive its derivative $\dot{P}_{il}(t)=dP_{il}(t)/dt$. If the derivatives of dynamic loadings $\dot{P}_{il}(t) = \dot{P}_{4il}(t) + \dot{P}_{5il}(t)$ are observed, we can find the force by integration: $P_{il}(t)=\int \dot{P}_{il}(t)dt$ (at zero initial conditions).

5. Formulation of the problem

The problem of adaptive control of an active suspension envisages the solution of two problems at each moment of time:

- minimization of the oscillations of the amortized mass;
- maintenance of the motion stability (minimization of the take off of the controlled wheels from the ground).

Let's consider formulations of these problems, leaving out the indexes of supports and boards of the machine for simplicity.

Formulation of the vibration protection problem

For object (1) for the output variable $x(t)$ the desirable motion properties are set in the following form:

$$\ddot{x}=f(\bar{x}, \dot{x}, x), \tag{6}$$

$$t\geq t_o: x(t_o)=x_o,\ \dot{x}(t_o)=\dot{x}_o,$$

where $f(.)$ is the set operator, generally nonlinear, defined on fuzzy sets; $\bar{x}$ is the command value (in the problem of vibration protection $\bar{x}=0$)

and the condition of asymptotical stability looks as follows:

$$x(t)\to\bar{x}(t),\ \dot{x}(t)\to 0 \text{ at } t\to\infty . \tag{7}$$

It is required to construct a regulator, which provides fulfillment of the conditions of the motion optimality (6) and asymptotical stability (7).

In a specific case, the desirable motion properties of an output variable are assigned by a linear equation of the type:

$$\ddot{x} + 2\psi\omega_0\dot{x} + \omega_0^2 x = \omega_0^2\bar{x} \tag{8}$$

with the initial conditions, which correspond to the system (1).

Here ψ, ω_0 are the assigned factors of aperiodicity and frequency of the non-dissipated oscillations, which define the desirable motion properties of the object giving constant influence on its input.

Equation (8) can be presented in the form of (6), making a replacement:

$$f(.) = \beta_0(\bar{x} - x) - \beta_1\dot{x}. \tag{9}$$

Here $\beta_0 = 1/\omega_0^2$; $\beta_1=2\psi\omega_0$.

Apparently, factors β_0 also β_1 are unequivocally connected with the parameters ψ and ω_0, which define the property of the reference motion of the amortized mass. Later on, the parameters β_0 and β_1 will be included into the control algorithms (regulators).

Formulation of the problem of stability of motion

The dynamic loading on a wheel $P(t)$ is considered as the output variable. For the output variable $P(t)$ the desirable motion properties are assigned in the following form:

$$\ddot{P} = f_P(\bar{P}, \dot{P}, P), \tag{10}$$

$$t\geq t_o: P(t_o)=P_o,\ \dot{P}(t_o)=\dot{P}_o .$$

Here $f_P(.)$ is the assigned operator, generally nonlinear, defined on fuzzy sets; $\bar{P}$ is the static loading.

And a condition asymptotic stability of system:

$$P(t)\to\bar{P}(t),\ \dot{P}(t)\to 0 \text{ at } t\to\infty . \tag{11}$$

It is required to construct a regulator, which provides performance of the conditions (10) and (11).

In particular case, the desirable motion properties of the output variable are assigned by a linear equation of the following kind:

$$\ddot{P} + 2\psi\omega_0\dot{P} + \omega_0^2 P = \omega_0^2\bar{P}$$

with the initial conditions, corresponding to the system (1).

Here ψ, ω_0 are specified values of factors of aperiodicity and frequency of the not dissipated oscillations of the transient for $P(t)$.

For the formation of control for each of the mentioned problems it is necessary to define the corresponding phase coordinates. Hence, it is required to define the observable and identified variables for each problem. Both problems are realized on the same actuating mechanism.

6. Control Algorithms

Control for each of servo-drives is calculated using the following formula (indexes of supports and boards are left out):

$$u(\bar{x}, x, \dot{x}, \ddot{x}, \dddot{x}, x^{(4)}\ \ f, \dot{f}, \ddot{f},\ z, \dot{z}, \ddot{z}) = \\ = \Phi_0\left\{k_0 z + \left[\Phi_1(f, \dot{f}, \ddot{f}) + \Phi_2(\ddot{x}, \dddot{x}, x^{(4)}) + \Phi_3(z, \dot{z}, \ddot{z})\right]\right\} \tag{12}$$

Here $\Phi_j(.)$ are the known operators ($j = 0,...,3$); z is the output variable of the power actuating mechanism; k_0 is the constant, which describes the efficiency of the negative feedback on the output variable of the power actuating mechanism.

The spectrum of adaptive regulators can be obtained by variation of the functionals Φ_j. As follows from the expression (12), from the point of view of calculations it is convenient to observe the accelerations of the controlled variable. Not observable, but necessary for the purposes of control variables can be obtained by using integration or differentiation.

The role of the functional Φ_1 consists in giving desirable motion properties to the controlled system. The functional Φ_2 reflects the actual condition of the object at each moment of time. The role of the functional Φ_3 is reduced to the compensation of the imperfections of the actuating mechanism. The first member in the expression (12) compensates the weakening of the control signal, if the feedback for the restriction of the output variable of the actuating mechanism is chosen a priori.

There are patents [14, 15] on the algorithm (12). In case of a special interest, the presented algorithm can be described completely. In the work [16], in addition, a method of control of the vehicle dynamics based on a new approach is described.

Control is calculated at the appeal to the procedure Regulator, in which $\bar{x}, x, \dot{x}, \ddot{x}, \dddot{x}, x^{(4)}, f, \dot{f}, \ddot{f}, z, \dot{z}, \ddot{z}$ are formal parameters: $\bar{x}, x, \dot{x}, \ddot{x}, \dddot{x}, x^{(4)}$ are accordingly the output variable and its derivatives for the corresponding channel of control (the channel of adaptive vibration protection and the channel of stability regarding the take off of the tires); $z, \dot{z}, \ddot{z}$ is the output variable of the actuating mechanism and its derivatives; $f, \dot{f}, \ddot{f}$ is the right part of the equations of reference motion for the output variable and its derivatives.

The control on every of the servo-cylinders contains two components:

$$u = B_1 u_1 + B_2 u_2 , \tag{13}$$

where u_1 is the control by vibration protection criteria; u_2 is the control by controllability criteria; B_1, B_2 are the constants: $B_1 + B_2 = 1$. The control components at each moment of time are calculated according to the procedure, which corresponds to the formulas (12) and (13), at various input variables and parameters. At calculation of the component u_1 the values $\bar{q}_1, q_1, \dot{q}_1, \ddot{q}_1, q_5, \dot{q}_5, \ddot{q}_5$ are transferred into the formula (13), and at calculation of the component u_2 the values $\bar{P}, P, \dot{P}, \ddot{P}, q_5, \dot{q}_5, \ddot{q}_5$, and also corresponding to them parameters of the algorithm.

The control components are calculated at each moment of time using the formula (12) at various input variables and parameters. Algorithms for the presented problems of control can be identically accurate within parameters or various on accuracy and speed.

Thus, parameters of resiliency are observed in an active suspension as well as parameters of dissipation. Energy for the accelerated adjustments of the functioning of the hydraulic cylinder is generated by means of an external source. The sensors of the wheel loading, of the displacement and of the acceleration transfer the signals to the electronic block of control within milliseconds. The control system allows reaching a constant loading on a wheel with maintenance of a constant average height of the vehicle. Steel springs or hydro pneumatic elements of the suspension are used to maintain the static loading on a wheel.

7. Example of computer modelling

Mathematical models and program modules are realized for the multivariate systems of stabilization and active vibration protection. Computer modeling was made for the existing multi-wheel and caterpillar machines with an adaptive suspension. Here, for the simplicity of formulation of the general approach, we will consider the two-mass systems with an active suspension at stochastic kinematic perturbation.

The motion equations for a non-linear two-mass system with an active suspension are presented in the following form:

$$\begin{aligned}
&\dot{q}_1 = q_2 , \\
&\dot{q}_2 = -\Sigma_1 / m_1 , \\
&\dot{q}_3 = q_4 , \\
&\dot{q}_4 = (\Sigma_1 - \Sigma_2) / m_2 , \\
&\dot{q}_5 = (-q_5 + k_x (u - k_{oz} q_6)) / T_x , \\
&\dot{q}_6 = (-q_6 + k_z q_5 - k_z k_e q_2) / T_z , \\
&\dot{q}_7 = (-q_7 + k_s \dot{q}_2) / T_s , \\
&\dot{q}_8 = (-q_8 + k_s \dot{q}_4) / T_s , \\
&u(\bar{x}, x, \dot{x}, \ddot{x}, \dddot{x},\ f, \dot{f}, \ddot{f},\ z, \dot{z}) = \\
&= \Phi_0 \left\{ k_0 z + \left[\Phi_1(f, \dot{f}, \ddot{f}) + \Phi_2(x, \dot{x}, \ddot{x}, \dddot{x}) + \Phi_3(z, \dot{z}) \right] \right\} \\
&t \ge t_0 : q_i(t_0) = q_{0i},\ i = 1,...,8 .
\end{aligned} \tag{14}$$

Here q_i are the phase coordinates ($i = 1,...,8$); Σ_1, Σ_2 are the "sheafs" of forces, generally non-linear, influencing accordingly on the masses m_1 and m_2:

$$\Sigma_1 = P_{11} + P_{21} + P_{31} ;$$

$$\Sigma_2 = P_{12} + P_{22} + P_{32} ;$$

$$P_{11} = P_{11}(\Delta_1) ; P_{21} = P_{21}(\dot{\Delta}_1) ; P_{31} = \bar{P}_{31} \operatorname{sgn} \dot{\Delta}_1 ;$$

$$P_{12} = P_{12}(\delta_2) ; P_{22} = P_{22}(\dot{\delta}_2) ; P_{32} = \bar{P}_{32} \operatorname{sgn} \dot{\delta}_2 ;$$

$\Delta_1 = q_3 - q_1 + q_6$; $\dot{\Delta}_1 = q_4 - q_2 + \dot{q}_6$;

$\delta_2 = Q(t) - q_3$; $\dot{\delta}_2 = \dot{Q}(t) - q_4$,

where $P_{ji}(.)$ are the known functionals, generally non-linear; $\bar{P}_{31}$, $\bar{P}_{32}$ are the parameters of the dry friction in a static state in the suspension and in the tire, accordingly; $Q(t)$ kinematic perturbation; $\dot{Q}(t) = dQ(t)/dt$. The displacement of the plunger of the actuating mechanism q_6 is included into the deformation of the elastic element Δ_1; k_x, T_x are accordingly the factor of strengthening and the constant of time of the electro hydraulic converter; k_z, T_z are accordingly the factor of strengthening and the constant of time of the power actuating mechanism (hydro-cylinder); k_e is the parameter of the hydraulic engine, which describes outflows of the working body at its normal functioning; k_o is the factor of the feedback by the position of the plunger of the actuating mechanism; k_s, T_s are accordingly the factor of strengthening and the constant of time of the measuring device.

In a linear variant the elastic-dissipative forces look as follows:

$$P_{11}=C_1\Delta_1;\quad P_{21} = K_1\dot{\Delta}_1;\quad P_{31} = 0;$$

$$P_{12}=C_2\delta_2;\quad P_{22} = K_2\dot{\delta}_2;\quad P_{32} = 0.$$

Here K_1, K_2 are accordingly the factors of viscous dissipation of energy in the shock absorber and in the tire; C_1, C_2 are accordingly rigidities of the elastic element and the tire.

The first two equations in the system (14) are equations of the control object (of the amortized mass m_1); the third and the fourth equations are the motion equations of the non-amortized mass m_2, the fifth is the equation of the electro-hydraulic converter (spool operated valve), the sixth is the equation of the power actuating mechanism (hydraulic cylinder); the seventh and the eighth are the motion equations of the sensors of accelerations of the masses m_1 and m_2. The other variables, which are used in the control algorithm, can be identified. The position sensor of the actuating mechanism is considered as an inertialess static part and, consequently, its equation is not examined.

At modeling the following restrictions are taken into account:

- on the displacement of the spool operated valve $q_5(t)$:

$q_{5\min} \le q_5(t) \le q_{5\max}$,

where $q_{5\min}$, $q_{5\max}$ are the bottom and top legitimate values;

- on the displacement of the variable $q_6(t)$ (the displacement of the plunger of the power hydro cylinder):

$q_{6\min} \le q_6(t) \le q_{6\max}$,

where $q_{6\min}$, $q_{6\max}$ are the bottom and top legitimate values;

- on the control functions:

$u_- \le u(t) \le u_+$,

where u_-, u_+ are accordingly the bottom and top legitimate values. In case if the variables q_5, q_6 and u overrun the legitimate limits, they get boundary values.

The results of the computer modeling of oscillations of the multi-wheel and caterpillar machines have shown high efficiency of the considered methods of the adaptive vibration protection and stabilization in a wide range of conditions of functioning.

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