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SENSITIVITY ANALYSIS IN MODELLING OF A DRIVING SYSTEM OF A GEARHEAD OF THE KGS 300 SHEARER

Summary. The paper presents the application of a direct differentiation method for the sensitivity analysis of a driving system of a gearhead of the KGS 300 coal shearer. An electromechanical model of the driving system is composed of a model of a discrete mechanical part of the system and of a model of the electrical motor. A functional describing the maximum values of dynamic forces has been assumed for the analysis.

ANALIZA WRAŻLIWOŚCI W MODELOWANIU UKŁADU NAPĘDOWEGO GŁOWICY KOMBAJNU WĘGLOWEGO KGS-300

Streszczenie. W pracy przedstawiono zastosowanie metody bezpośredniego różniczkowania do analizy wrażliwości układu napędowego głowicy kombajnu węglowego KGS 300. Elektromechaniczny model układu napędowego składa się z modelu dyskretnego mechanicznej części układu oraz modelu silnika elektrycznego. Do analizy przyjęto funkcjonał opisujący wartości maksymalne sił dynamicznych.

АНАЛИЗ ЧУВСТВИТЕЛЬНОСТИ В МОДЕЛИРОВАНИИ ПРИВОДНОЙ СИСТЕМЫ ГОЛОВКИ КОМБАЙНА КГС-300

Резюме. В статье приводится применение метода непосредственного дифференцирования к анализу чувствительности приводной системы головки угольного комбайна КГС-300. Электромеханическая система состоит из дискретной модели механической части системы и модели электрического двигателя. Для анализа принято функционал описывающий максимальные значения динамических сил.

1. INTRODUCTION

The dynamic analysis of mechanical systems makes it possible to

identify dynamic phenomena occurring in the system being considered. The obtained magnitudes of dynamic forces may constitute the basis for selecting design features of the driving system providing for its durability and reliability. However, a change in the design features of the system causes a change in character and magnitude of dynamic forces in kinematic pairs. The analysis of sensitivity of this system enables evaluating of the influence of a change in parameters of the physical model of the driving system on magnitudes of dynamic forces in kinematic pairs. Finding of the analysis may serve as the basis for optimization of the system under consideration. Assumption of a proper functional which should constitute an objective function for the optimization is an important stage of the sensitivity analysis.

2. MODELLING

A physical model of the system should describe a real system as precisely as possible under taking into account the factors which have a crucial influence on dynamic properties of the object being modelled. The assumed model is a result of the compromise between the most true possible description of a phenomenon and the realization of intended examinations. When analyzing the design form of the driving system of gearheads of the hearer it can be assumed that physical phenomena will be described precisely enough provided that this is a discrete model with the feedback of mechanical part and the electrical one. In this case dynamic phenomena in the driving system are being modelled as torsional characteristics and depend on a state of load of the system, its mechanical properties and performance characteristic of the electric motor. In this way the driving system is considered as an electromechanical system with feedback. It is a particular importance that the feedback is taken into account when transient states such as start-up or a sudden change in the load state are subject to analysis. In these cases it happens that in kinematic pairs there are dynamic forces generated the maximum magnitudes of which often exceed to a considerable degree the nominal values [2,3]. The assumption of physical relationships existing in kinematic pairs of the driving system is essential for evaluating the factor influencing on magnitudes of dynamic forces of the system. The research which has been carried out in the Department of Engineering Mechanics hitherto indicates that it is necessary to take into

account the nonlinearities resulting from design features of the system and especially the plays. Due to the examination performed a model and an algorithm of calculation with regard to above mentioned nonlinearities have been developed. But because of difficulties of mathematical nature when analyzing the sensitivity of the system with nonlinear parameters of the physical model, a linearized model has been used in the first stage of the sensitivity analysis. This model will be also applied for optimization of the driving system. This is a form of differential equations of motion in this case:

$$M\ddot{q} + C\dot{q} + Kq = Q \quad (1)$$

where: M - inertia matrix,
 C - damping matrix,
 K - stiffness matrix,
 Q - generalized forces,
 q - generalized displacements.

The dynamic characteristic of the driving motor to be found have been determined by modelling the electric motor with a system of differential equation [2, 4]. Parameters of the model of the electric motor have been stated by theoretical study and experiments.

3. SENSITIVITY ANALYSIS

It is a task of the sensitivity analysis to examine the influence of selected parameters of the physical model of the system on maximum values of dynamic forces in kinematic pairs. The functional of the following form has been assumed for this purpose:

$$\Psi = P_{\max}^2 \quad (2)$$

where: P_{\max} - calculated maximum value of a dynamic force in the kinematic pair of the system under consideration.

The algorithm of the sensitivity analysis under applying of the direct differentiation method is as follows [1]:

$b = [b_1, \dots, b_n]^T$ - vector of design variables

$$\dot{x} = f(x, b) \quad (3)$$

$$\mathbf{x}(t_1) = \mathbf{h}(\mathbf{b}) \quad (4)$$

$$t_1 \leq t \leq t_2$$

$$\mathbf{x}(t) = [x_1(t), \dots, x_m(t)]^T \quad (5)$$

where $\mathbf{x}(t)$ - solution of initial value problem (3),
 t_1, t_2 - initial and final time.

In general case the final time is expressed by a function:

$$F(t_2, \mathbf{x}(t_2), \mathbf{b}) = 0 \quad (6)$$

A typical functional describing the state of the system at a definite moment t has a form:

$$\Psi = g(t_2, \mathbf{x}(t_2), \mathbf{b}) \quad (7)$$

The first derivative of the functional with respect to the design variables has a form:

$$\frac{d\Psi}{d\mathbf{b}} = \mathbf{G}^T(t_2, \mathbf{x}(t_2), \mathbf{b}) \frac{\partial \mathbf{x}(t_2)}{\partial \mathbf{b}} + \frac{\partial g}{\partial \mathbf{b}} - \left[\frac{\frac{\partial g}{\partial t_2} + \frac{\partial g}{\partial \mathbf{x}} \cdot \mathbf{f}(t_2)}{F(t_2)} \right] \frac{\partial F}{\partial \mathbf{b}} \quad (8)$$

$$\mathbf{G}^T(t_2, \mathbf{x}(t_2), \mathbf{b}) = \left[\frac{\partial g(t_2)}{\partial \mathbf{x}} \right]^T - \left[\frac{\frac{\partial g}{\partial t_2} + \frac{\partial g}{\partial \mathbf{x}} \mathbf{f}(t_2)}{F(t_2)} \right] \left[\frac{\partial F}{\partial \mathbf{x}} \right]^T \quad (9)$$

$\frac{\partial \mathbf{x}(t_2)}{\partial \mathbf{b}}$ is the only expression in the equation (8) which cannot be determinated analytically for $\mathbf{x}(t)$ determined numerically. It is possible to determine this expression by introducing an additional equation as a result of differentiation of (3) and (4) with respect to \mathbf{b} :

$$\frac{\partial \mathbf{x}}{\partial \mathbf{b}} = \frac{\partial \mathbf{f}}{\partial \mathbf{x}} \cdot \frac{\partial \mathbf{x}}{\partial \mathbf{b}} + \frac{\partial \mathbf{f}}{\partial \mathbf{b}} \quad (10)$$

$$\frac{\partial \mathbf{x}(t_1)}{\partial \mathbf{b}} = \frac{\partial \mathbf{h}}{\partial \mathbf{b}} \quad (11)$$

$t_1 \leq t \leq t_2$

The solution of the initial value problem (10) and (11) brings about $\frac{\partial \mathbf{x}(t)}{\partial \mathbf{b}}$. A very simple algorithm and possibility of simultaneous

calculation of $\partial x(t)/\partial b$ and $x(t)$ are advantages of this method. Its disadvantage consists in the necessity of solving a great number of additional differential equations. Results of the sensitivity analysis as carried out may be checked by means of the method based on calculus of variations. If we add a small perturbation δb to the design variables

$$b^* = b + \delta b \quad (12)$$

and then solve the perturbed model, the actual change in the functional value is expressed by:

$$\delta \psi = \psi(b^*) - \psi(b) \quad (13)$$

In the case of correctly calculated gradients the actual change in the functional value should be approximate to:

$$\delta \psi = v \psi^T \cdot \delta b \quad (14)$$

4. NUMERICAL CALCULATIONS

A driving system of gearheads of the KGS 300 coal shearer incorporates as asynchronous double squirrel cage motor, six stages of series gear and planetary gear. The system has been modelled as an electromechanical system with 16 degrees of freedom (Fig.1). Assuming that the changes in the system under consideration should be as little as possible seven design variables have been selected:

- moment of inertia I_3 ,
- stiffness coefficient k_1 ,
- damping coefficient c_1 ,
- stiffness coefficient k_{15} ,
- damping coefficient c_{15} ,
- moment of inertia I_{14} ,
- moment of inertia I_{15} .

Calculations have been made for the following variants of external load:

1. $M_r = \text{const} > 0$ for $t \geq 0$,
2. $M_r = 0$.

As a result of numerical calculations the same signs of gradients in relations to corresponding design variables have been obtained but the values of the variant 1 are much higher than those of the variant 2. Values of gradients as well as the time of calculation for the work station APOLLO

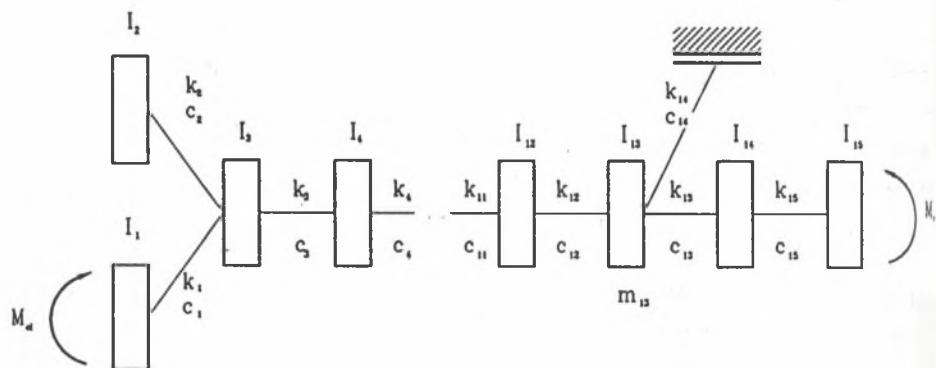


Figure 1. Dynamic model of a driving system of the KGS 300 shearer

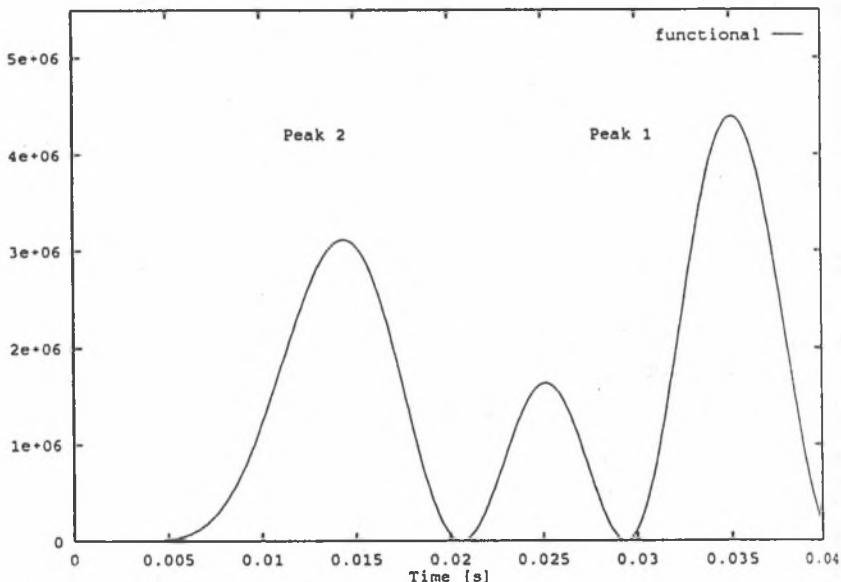


Figure 2. Functional $\psi(t)$

DN 5500 at the second variant of calculations are shown in the table 1. The functional for the variant 2 is shown in Fig. 2.

5. CONCLUSIONS

Basing oneself on results of the sensitivity analysis it is possible to evaluate a quantitative and qualitative influence of selected parameters

Table 1

Gradients of the functional obtained using the direct differentiation method

peak		1	2
value		4395415.	3116474.
gradient with respect to	k_1	$-0.4237 \cdot 10^0$	$-0.1189 \cdot 10^0$
	C_1	$-0.8272 \cdot 10^1$	$-0.3141 \cdot 10^2$
	I_3	$0.3664 \cdot 10^7$	$0.3195 \cdot 10^7$
	K_{15}	$-0.4438 \cdot 10^0$	$0.7764 \cdot 10^{-1}$
	C_{15}	$-0.2072 \cdot 10^3$	$-0.1281 \cdot 10^2$
	I_{14}	$0.1205 \cdot 10^7$	$0.2219 \cdot 10^6$
	I_{15}	$0.1715 \cdot 10^7$	$0.1574 \cdot 10^6$
Predicted change of peak value		1221.68	223.85
Actual change of peak value		1221.64	223.87
CPU time [s]		586.3	242.7

of the model on values of the assumed functional. The results obtained indicate a predominant influence of magnitudes of moments of inertia and of damping coefficients on maximum magnitudes of dynamic forces in kinematic pairs. The accuracy of the results obtained proves that the assumed algorithm of the sensitivity analysis is applicable to the solution of the optimization problems of mechanisms of this class. The results of the sensitivity analysis point to the direction of changes in design features of the leading to its optimization.

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Streszczenie

Analizując postać konstrukcyjną układu napędowego głowic kombajnowych przyjęto, że zjawiska dynamiczne będą wystarczająco dokładnie opisane przy założeniu modelu dyskretnego ze sprzężeniem zwrotnym części mechanicznej i elektrycznej. W ten sposób układ napędowy rozpatrywano jako układ elektro-mechaniczny ze sprzężeniem zwrotnym. Ze względu jednak na trudności matematyczne przy analizie wrażliwości układu z nieliniowymi parametrami modelu fizycznego przyjęto model zlinearyzowany. Poszukiwane charakterystyki dynamiczne silnika napędowego wyznaczono z jego modelu matematycznego. Celem analizy wrażliwości jest zbadanie wpływu wybranych parametrów modelu fizycznego układu na wartości maksymalne sił dynamicznych w parach kinematycznych. Dla tak postawionego zadania przyjęto funkcjonal w postaci opisującej wartości maksymalne sił dynamicznych w parach kinematycznych. Do analizy wrażliwości zastosowano metodę bezpośredniego różniczkowania. Wyniki przeprowadzonej analizy wrażliwości zostały sprawdzone metodą opartą na rachunku wariacyjnym.

Układ napędowy głowicy kombajnu węglowego KGS 300 składający się z dwuklatkowego silnika asynchronicznego, sześciu stopni przekładni szeregowej oraz przekładni planetarnej zamodelowano jako elektromechaniczny o 16. stopniach swobody. Zakładając, że zmiany w rozpatrywanym układzie powinny być jak najmniejsze, wybrano siedem projektowanych parametrów modelu układu (momenty bezwładności, współczynniki sztywności, współczynniki tłumienia). Przedstawiono wartości gradientów oraz czas obliczeń dla stacji roboczej APOLLO DNS500. Otrzymane wyniki wskazują na dominujący wpływ wielkości momentów bezwładności i współczynników tłumienia na wielkości maksymalne sił dynamicznych w parach kinematycznych. Dokładność otrzymanych wyników potwierdza, że przyjęty algorytm analizy wrażliwości może być stosowany do rozwiązywania zadań optymalizacji tej klasy mechanizmów.