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COMPUTER AIDED OPTIMAL DESIGN OF ELASTIC SUSPENDED PRESSES

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1. Introduction

The problem of high precision machine suspension to vibration-isolated fundaments has been under consideration for many years [1,2]. Recently there is a tendency of using elastic suspension for machines with unbalanced moving parts, and for machines with impact loading. In all of these cases a direct elastic suspension is used, instead of using massive and expensive vibroisolated fundaments. Very often metal-rubber tampons as elastic elements for the suspension [5] are used as well.

As it is mentioned in [4] some of the advantages of the elastic suspension are as follows:

– decreasing dynamic loading which are transmitted through the basement to the operating personnel, surroundings, equipment, machines, instruments;

– decreasing the mounting expenditure of machines (speeding up the process of mounting); allowing easy change of the machine's place when it is necessary (due to changes in the production process);

– decreasing the dynamic loading in the machine parts; and decreasing the noise and vibrations.

There are two contraductive tendencies when the elastic suspension is used. From one hand, to minimize the transmitted force down to the basement, soft elastic suspension is required (which leads to greater displacements from the equilibrium); on the other hand, for the precise work of the press it is necessary to have small displacements from the equilibrium, which needs hard elastic suspention. Unfortunately there is no data for the limits of the displacements in the known literature. There are standards for vibro-speed and vibro-acceleration in BDS (Bulgarian State Standard) 16013-64 but there is no standard for the displacement.

2. Problem formulation

The aim of this paper is to solve the problem of optimal setting up of the elastic components and to choose their elastic coefficients such that either

the decreasing of the transmitted force to the basement and the displacement of the press to be in acceptable limits.

Following steps are proposed for solving this problem:

(a) create dynamic model of the press as an elastic suspended rigid body;

(b) create the mathematical model by means of system differential equations of movements;

(c) define the objective functions and carry out the computer aided optimization in order to design the optimal parameters for the elastic suspension;

(d) find the natural frequences of the optimal suspension and check up the resonance.

3. System and optimization description

3.1 A Dynamic Model

The press is considered as a six degree of freedom rigid body. The movement of the elastic suspended press is examined accordingly to the static coordinate axes which origin is in its mass centre. Letters x and y denote the axes in horizontal plane (x axis is pointed to the operator), and z denotes the vertical axis. The three rotating angles around the axes are denoted with φ, ψ and θ respectively.

The system differential equations which describes the movement of the elastic suspended press has the form (1).

$$
\begin{aligned}
&m\ddot{x} + n\beta_2\dot{x} + nz_0\beta_2\dot{\psi} + nc_2x + nz_0c_2\psi = 0\\
&m\ddot{y} + n\beta_2\dot{y} - nz_0\beta_2\dot{\varphi} + nc_2y + nz_0c_2\varphi = 0\\
&m\ddot{z} + n\beta_1\dot{z} + nc_1z = \Phi_\varepsilon\\
&I_x\ddot{\varphi} + \left(\beta_1\textstyle\sum_{i=1}^{n} y_i^2 + nz_0^2\beta_2\right)\dot{\varphi} - nz_0\beta_2\dot{y} +\\
&\qquad\left(c_1\sum_{i=1}^{n} y_i^2 + nz_0^2c_2\right)\varphi - nz_0c_2y = -\Phi_\varepsilon l_y\\
&I_y\ddot{\psi} + \left(\beta_1\sum_{i=1}^{n} x_i^2 + nz_0^2\beta_2\right)\dot{\psi} - nz_0\beta_2\dot{x} +\\
&\qquad\left(c_1\sum_{i=1}^{n} x_i^2 + nz_0^2c_2\right)\psi - nz_0c_2x = -\Phi_\varepsilon l_x - (a - b\dot{\varepsilon}i)
\end{aligned}
\tag{1}
$$

$$I_y\ddot{\theta} + \left(\beta_2 \sum_{i=1}^{n} x_i^2 + \beta_2 \sum_{i=1}^{n} y_i^2\right)\dot{\theta} + \left(c_2 \sum_{i=1}^{n} x_i^2 + c_2 \sum_{i=1}^{n} y_i^2\right)\theta = 0$$

$$\left\{J_r + J_{ec} + m_s\left[r\sin\varepsilon\left(1 + \frac{\lambda\cos\varepsilon}{\sqrt{1-\lambda^2\sin^2\varepsilon}}\right)\right]\right\}\ddot{\varepsilon} =$$

$$(a - b\dot{\varepsilon}i)i - \left[r\sin\varepsilon\left(1 + \frac{\lambda\cos\varepsilon}{\sqrt{1-\lambda^2\sin^2\varepsilon}}\right)\right]R_\varepsilon$$

Assumptions that all elastic elements are the same, all lay in the same plane, and all are symetrically situated accordingly to the mass centre projection on this plane are made.

Following notations are used in (1):

m – mass of the press [kg];

n – number of elastic elements;

β_1, β_2 – linear damping coefficients [Ns/m];

z_0 – distance from the mass centre to the basement [m];

c_1, c_2 – elastic constants of the tampons [N/m];

ε – rotation angle of the electromotor [rad];

Φ_ε – inertial force caused by movement of the press support during the reciprocating movement [N];

R_ε – resistance force of the press (cutting, impact by Cresher) [N];

I_x, I_y, I_z – mass inertial moments of the press [kgm^2];

x_i, y_i – coordinates of the elastic elements;

a, b – characteristic constants of the electromotor [Nms];

i – transmition ratio between the electromotor shaft and flywheel shaft;

J_r – relative mass inertial moment of all rotating parts in front of the flywheel [kgm^2];

J_{ec} – inertial moment of the eccentric shaft [kgm^2];

m_s – mass of the support [kg];

r – eccentricity of the eccentric shaft [m];

λ – eccentricity/length ratio of the connecting rod;

3.2 Objective functions

When presses are elastically suspended there are two groups of parameters which can be optimized. First group includes elastic ratios of the tampons and the second one includes coordinates of tampons' positions. General form of the optimization problem is to minimize (maximize) the objective functions with respect to these parameters. The parameters are bounded by

physical realizability criteria so that the minimization of the objective functions is subjected to the constraints which denote upper and lower bounds of the parameters intervals.

The objective functions under consideration are as follows: The first objective function minimizes displacement of the press from the equilibrium, or more precisely it minimizes amplitudes of the vibrations according z axis.

$$F_1 = \min A_z. \tag{2}$$

The experimental data and numerically evaluated data show that vertical displacement and rotation around Y axis are considerably greater than the others displacements of the press. This is reasonable because the inertial excitation in these two directions is the biggest. The second objective function minimizes amplitudes of torsional vibrations according y axis.

$$F_2 = \min A_\psi. \tag{3}$$

The third objective function minimizes the transmitted force to the basement.

$$F_3 = \min(c_1 A_z). \tag{4}$$

If there are two or more objective functions to be minimized simultaneously the problem is expanded to multi-criteria or multi-objective optimization problem. Some authors [7,8] reduce this problem to single-criterial optimization problem using super criterion which includes all the objective functions. The objective functions usually are multiplied by weight coefficients which indicate the importance of the objectives.

3.3 Parameter intervals

The initial interval for the elastic constant must guarantee eigenvalue of the vertical vibrations of the press in the interval 6-12 [Hz]. It is experimentally proved that if tampons are softer (i.e. the natural frequency of the vertical vibrations is under 6 [Hz]) the displacements from the equilibrium are considerable and the work of the press is getting worse. When the tampons are harder (i.e. the natural frequency of the vertical vibrations is bigger then 12 [Hz]) the transmitted force to the basement is considerable as well. It is important to note that the resonance (i.e. when natural frequency is equal to excitation frequency) is out of consideration.

The intervals for the elastic constants of the tampons (for the family eccentric presses) are given in Table 1.

Table 1. Intervals for the designed elastic constants

[N/m]	PE 25A	PE 40A	PE 63A	PE 100A
$c_{1\min}$	373071	410733	608994	819233
$c_{1\max}$	500000	500000	800000	900000

Following considerations must be taken into account when intervals for the coordinates x and y are evaluated: As a lower bound of the intervals must be taken the coordinates of the supporting points of the press (smaller values than that lead to unstable equilibrium of the press); When the upper bound is determined, the increase of x and y will decrease the natural frequencies k_i $(i = 1, 2, \ldots, 6)$ of the press. The increase of x and y coordinates of the elastic elements increases the size of support plate which leads to difficulties in work with the press.

Taking into account above considerations, the limits for the intervals are chosen as shown in Table 2.

Table 2. Intervals of the design parameters

[m]	PE 25A	PE 40A	PE 63A	PE 100A
$x_{\min}$	−0.785	−0.785	−0.802	−0.780
$x_{\max}$	0.415	0.560	0.528	0.940
$y_{\min}$	−0.500	−0.485	−0.550	−0.600
$y_{\max}$	0.500	0.485	0.550	0.600

3.4 Optimization procedure

The optimization procedure is carried out using method of sounding [6] which searches uniformly distributed testing points from the parameter space, defined from the intervals given in Table 2.

The optimizing algorithm is similar to one suggested in [9]:

(1) give the number of parameters under consideration;

(2) give the parameter intervals (the lower and upper bound of each parameter) which determine the parameter space;

(3) give the number of testing points (these are uniformly distributed sequence of points in the design parameter space);

(4) compute values for the design parameters;

(5) solve system (1) for each point with zero initial values (which represents the steady state vibrations);

(6) choose maximum amplitude (by modulus) from the steady state response;

(7) compute the function values F_i $(i = 1, 2, 3)$;

(8) repeat steps from 3 to 7;

(9) compare values for each function with the previous one and choose the better one, and do this for all points from the uniformly distributed sequence. Using above procedure the optimization is carried out with 1024 testing points on an 386 PC. The obtained optimal results for the family eccentric presses PE25, PE40, PE63 and PE100 are given and discussed below.

4. Results and discussion

Obtained optimal parameters for the family eccentric presses PE25 are given in Table 3.

Table 3. Optimal parameters for family presses PE25

[m]	F_1	F_2	F_3
c_1 [N/m]	499020	496011	374065
$x_1, \ldots, x_5$	0.355	0.411	0.396
$x_6, \ldots, x_8$	−0.591	−0.686	−0.666
y_1, y_8	−0.466	−0.455	−0.482
y_2	−0.260	−0.260	−0.260
y_3, y_7	0	0	0
y_4	0.260	0.260	0.260
y_5, y_6	0.466	0.455	0.482

It is seen from Table 3 that first two criteria determine optimal values for the elastic ratio c_1 which are close to the upper bound of the interval. The other two criteria define close to the lower bound values for the elastic ratio. Obtained optimal values for the coordinates of the elastic elements show that decreasing of the plate size is possible only accordingly y axis.

Table 4. Optimal parameters for family presses PE40

[m]	F_1	F_2	F_3
c_1 [N/m]	499300	421190	411430
$x_1, \ldots, x_5$	0.497	0.482	0.54
$x_6, \ldots, x_{10}$	−0.2	−0.2	−0.2
$x_7, \ldots, x_9$	−0.695	−0.670	−0.767
y_2	−0.210	−0.210	−0.210
y_3, y_8	0	0	0
y_4	0.210	0.210	0.210
$y_5, \ldots y_7$	0.459	0.406	0.471
y_1, y_9, y_{10}	−0.459	−0.406	−0.471

Table 4 shows obtained optimal values for family presses PE63. The result for the elastic ratio c_1, obtained after optimization of the first criterion, is similar to this shown in previous Table 3. The values for c_1 tend to the upper bound of the initial optimization interval. The difference in this case is that optimization after all other criteria leeds to values for c_1 close to the lower bound of the interval (from Table 1). Obtained optimal coordinates for this family presses do not allow considerable decreasing of the plate size.

Table 5. Optimal parameters for family presses PE63

[m]	F_1	F_2	F_3
c_1 [N/m]	798510	635850	798510
$x_1, \ldots, x_5$	0.436	0.476	0.436
$x_6, \ldots, x_{10}$	−0.042	−0.042	−0.042
$x_7, \ldots, x_9$	−0.699	−0.765	−0.699
y_2	−0.185	−0.185	−0.185
y_3, y_8	0	0	0
y_4	0.185	0.185	0.185
$y_5, \ldots y_7$	0.513	0.443	0.513
y_1, y_9, y_{10}	−0.513	−0.443	−0.513

The optimal values for the design parameters of family presses PE63 are shown in Table 5. The results for the second criterion are analogous to that for family PE40 (values for c_1 close to the lower bound of the interval); whether all other criteria leed to optimal values for c_1 close to the upper bound of the interval. Optimal values for the coordinates allow considerable decreasing of the plate size.

Table 6. Optimal parameters for family presses PE100

[m]	F_1	F_2	F_3
c_1 [N/m]	899680	897480	899680
x_1, x_6	0.705	0.705	0.705
$x_2, \ldots, x_5$	0.795	0.935	0.795
x_7, x_{14}	−0.32	−0.32	−0.32
$x_8, \ldots, x_{13}$	−0.659	−0.752	−0.659
y_3, y_2, y_{13}, y_{14}	−0.393	−0.511	−0.393
y_3, y_4, y_{12}	−0.365	−0.365	−0.365
$y_5, \ldots y_8$	0.393	0.511	0.393
y_9	0.365	−0.365	0.393
y_{10}	0.205	0.205	0.205
y_{11}	−0.205	−0.205	−0.205

Finally optimal results for the family presses PE100 are given in Table 6. It is seen from this table that all criteria under consideration define optimal values for c_1 close to the upper, bound of the interval. Obtained optimal values for the coordinates of the elastic elements show that decreasing of the plate size is possible only along y axes.

One can see that obtained optimal values for the design parameters of the elastic suspended family presses are suitable for practical implementation. The designer can choose one or other criterion depending on the specific requirements, i.e. if it is necessary the transmitted to the basement force to be minimal, or if the displacements from the equilibrium of the press to be minimal. If it is necessary all objectives to be satisfied simultaneously then the optimization is considered as multicriteria one, which is not the aim of this work.

Table 7. Natural and excitation frequencies with optimal parameters

Frequences/Presses		PE25A	PE40A	PE63A	PE100A
Excitation freq.		14.95	13.09	12.05	5.51
	k_1	9.43	7.03	3.04	3.18
	k_2	9.93	10.17	8.66	8.87
Natural	k_3	32.71	23.09	19.10	26.22
frequences	k_4	39.71	25.61	24.64	30.28
(F_1)	k_5	43.60	41.57	43.17	39.51
	k_6	133.90	120.08	147.38	290.73
	k_1	9.23	5.83	5.81	3.84
	k_2	11.35	9.1	8.47	10.04
Natural	k_3	32.97	22.81	24.23	26.11
frequences	k_4	39.62	24.26	24.57	30.61
(F_2)	k_5	43.47	38.18	38.52	39.46
	k_6	146.72	111.81	148.50	333.87
	k_1	8.49	6.57	3.04	3.18
	k_2	9.67	9.95	8.66	8.87
Natural	k_3	32.52	22.98	19.10	26.22
frequences	k_4	37.75	25.47	24.64	30.23
(F_3)	k_5	39.34	37.73	43.17	39.51
	k_6	145.79	126.94	147.38	290.73

5. Check up for resonance domain

The resonance domains of the family eccentric presses designed with the obtained optimal values for the parameters c_1, x_i, and y_i, $(i = 1, 2, \ldots, 12)$ are experimentally checked.

Table 7 shows six natural frequencies (any press is considered as six degree of freedom rigid body) of the family presses, according each criteria. The values of excitation frequency are given in this table as well.

Elastic suspended family eccentric presses with elastic constants, and coordinates of the elastic elements chosen accordingly to the numerical results are examined. It is seen in Table 7 that natural frequencies are far enough from the excitation ones, i.e. the nearest natural frequency of PE25 is more than 30% far from the excitation frequency, for PE40, PE63 and PE100 they are more than 20%, 25%, and 30% far from the according excitation frequencies respectively. This check-up assure us that resonance is not possible for all presses designed with optimal parameters according the considered objective functions.

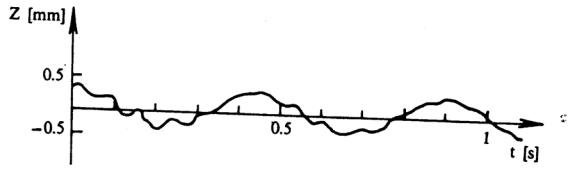


Figure 1(a). Amplitude-frequency characteristic (experimental)

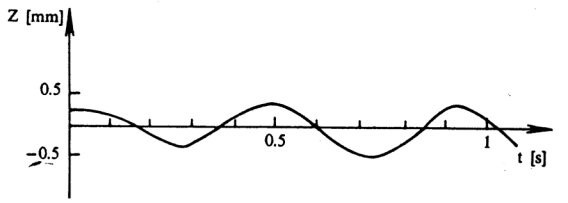


Figure 1(b). Amplitude-frequency characteristic (numerical)

Figure 1(a) illustrates recorded vertical vibrations of the elastic suspended press PE40A (first column in Table 4) and Figure 1(b) shows numerically obtained results with optimal parameters after optimization of F_1, (after solving (1)). Both the numerical and experimental results shown in Figure 1 represent the process of punching. The comparison between two graphics in

Figure 1 shows close agreement between the numerical and experimental results, which proves adequacy of chosen dynamic model to the real press. Shown values in Table 4, Table 5, and Table 6 for parameters c_1, x_i, y_i, $(i = 1, \ldots, 14)$ satisfy completely requirement frequencies of the vertical vibrations to be into the frequency domain 37,68 - 75,36 [rad/s] (see the values for k_5 in Table 7). For obtained parameters all natural frequencies are out of resonance domain which is $0.8 - 1.2$ [s$^-$1] for all presses.

6. Conclusion

The multi-criteria optimization of dynamic load factors for a system with six degree of freedom is solved transferring the problem to a single-criterion optimization.

Designed optimal parameters are easily generated for the cases where analytical solutions hardly can be found.

Numerically obtained results of the optimization are checked experimentally. The comparison shows good agreement between the numerical and experimental results.

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