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MULTI-OBJECTIVE OPTIMIZATION OF THE DRIVING SYSTEM OF THE 630 T MAXIPRESS

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Abstract: In this paper, we propose a population-based evolutionary multi-objective optimization approach based on the concept of Pareto optimality, in order to (re)design the driving system (the flywheel-clutch assembly) of the 630 t Maxipress (fig. 1). The goals of the optimization were to minimize the mass of the driving system and to maximize the moment of inertia. In the actually optimal design problem solved in this work, twelve genes and fifteen constraints were taken into consideration. The Pareto optimal set was obtained by running a new genetic algorithm inspired by Non-Dominated Sorting Genetic Algorithm II (NSGA-II) implemented in Cambrian v.3.09 software which belongs to the Optimal Design Centre of Technical University of Cluj-Napoca.

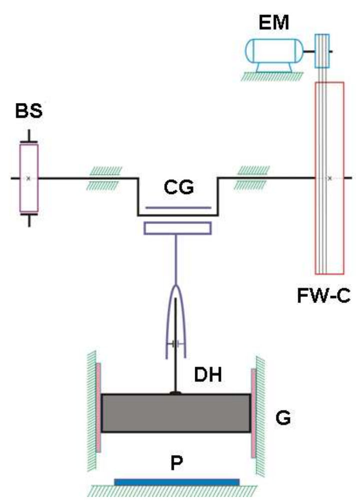

Fig.1. Functional scheme of the 630 t Maxipress

1. INTRODUCTION

The 630 t Maxipress is a high capacity hammering press, manufactured by Fortpres Co. (Cluj-Napoca, Romania). The operating principle of this press (fig. 1) is very simple: the electric motor (EM) actuates through the belt drive the flywheel (FW). Normally the flywheel freely rotates around the shaft and the pneumatic brake system (BS) is on (brakes on). Through the compressed air conducted in the pipe network, the brakes are disengaged and the clutch (C) couples the flywheel and the shaft. Now, the shaft rotates and the crank gear (CG) moves up and down in the guidance (G) the drop-hammer (DH) that hits the part (P). When the compressed air is cut-out, the brakes system freezes the shaft in the upper dead centre and the clutch disengages the shaft of the flywheel.

The authors focus on the optimal re-design of the driving system of the 630 t Maxipress, especially on the assembly mainly consisting in the flywheel and the clutch. The main parts of this assembly (fig. 2) are: the flywheel rim (1), the wearing plate (2), the clutch disk (3), the shaft (4), the hub (5), the tapered rolling bearings (6), the clutch pressure plate (7) and the sealing plate (8).

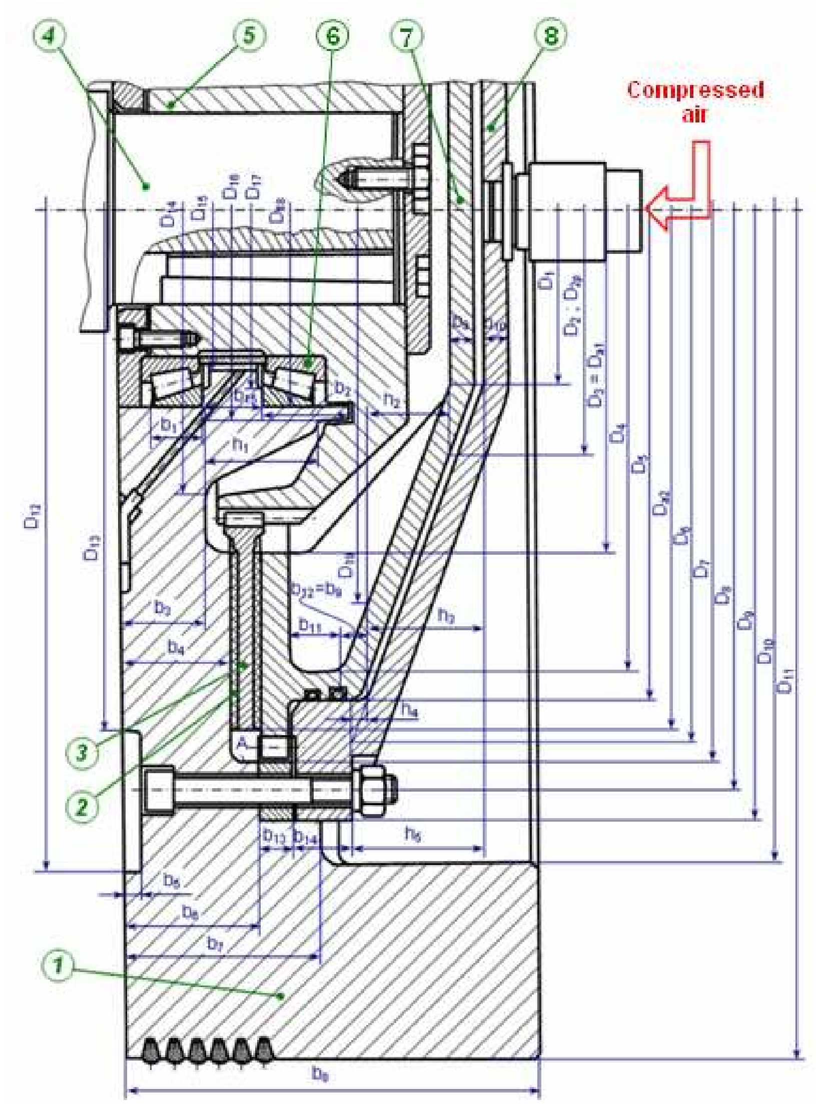

Fig.2. Sketch of the flywheel-clutch assembly

2. DESIGN PROBLEM

The aim of our work is to perform a multi-objective optimization in order to obtain a driving system (fig. 2) with a lower mass and with a moment of inertia as greater as possible. Obviously, these two objectives are in conflict and so it is impossible to reach such a result. Therefore we will use the Pareto front in order to deal with these two goals.

3. DESIGN INPUT DATA

Design data for the actual solution:
Mass of the actual design solution: M_i=9.485 kg;
Moment of inertia of the actual design solution: $J_i=6.266\cdot10^9$ kg·mm^2;
Outer diameter of the flywheel: D_{11}=2100 mm;
Width of the flywheel: b_8=500 mm.

The design input data are listed bellow:
Angular speed $\omega = 10.472\ s^{-1}$;
Moment of inertia of the parts of the press that have to be set on $J_a = 1.567 \cdot 10^8\ kg \cdot mm^2$;
Coefficient of friction: $\mu_a = 0.35$;
Compressed air pressure: $p_{air} = 0.5$ MPa;
Allowable crushing stress: $\sigma_{saA} = 3$ MPa;
Coefficient related to the technology: $k_f = 0.8$;
Allowable crushing stress: $\sigma_{scA} = 30$ MPa;
Allowable tensile stress: $\sigma_{tsA} = 80$ MPa;
Allowable tensile stress of the flywheel material: $\sigma_{tvA} = 80$ MPa;
Flywheel clutch material density: $\rho = 7.85 \cdot 10^{-6}\ kg/mm^3$.

In order to perform the optimal design of the sub-assembly is necessary to set up:

- The variables (genes) that uniquely describe the problem (both the objective function and the constraints);
- The objective functions;
- The constraints of the problem.

4. OPTIMAL DESIGN OF THE SUB-ASSEMBLY

4.1. Genes

The first step of the setup of the optimization program consists in the identification of the variables that are able to uniquely describe the problem. These variables should be involved in the calculus of the objective function and the constraints both. Hereinafter, since the optimization will be performed using genetic algorithms, instead of the notion of *variable* we will use the notion of *gene*.

It is worthy to mention here that the notion of *gene* is rather larger that the usual meaning of a variable. A *gene* could be a real or an integer number, as well as an array, a matrix or a list. The objects of the list could be anything one could imagine and that have a numerical coding (representation).

The authors consider that there are 12 genes that can describe completely the optimization problem. All the genes can have only integer values. These genes are listed as follows:

Gene 1: – number of the bolts, z_s (values between 8…36);
Gene 2: – bolt type*, *TS* (values between 0…31);
Gene 3: – number of acting faces of the clutch, z_a (values between 2, 4**);
Gene 4: – module of the involute splines, *m* (values between 10, 12.5 mm);
Gene 5: – number of splines, *z* (values between 100…250);
Gene 6: – inner diameter of the clutch disk, $D_3 = D_{a1}$ (values between 800…1100 mm);
Gene 7: – fitting diameter of the clutch pressure plate, D_5 (values between 1000…1600 mm);
Gene 8: – inner diameter of the flywheel, D_{10} (values between 1000…2000 mm);
Gene 9: – outer diameter of the flywheel, D_{11} (values between 1500…2500 mm);
Gene 10: – outer diameter of the circular recess of the flywheel, D_{12} (values between 1000…2000 mm);
Gene 11: – inner diameter of the circular recess of the flywheel, D_{13} (values between 800…1500 mm);
Gene 12: – width of the flywheel, b_8, (values between 200…800 mm);

Notes:

* contains all the standardized values: nominal diameter of the thread d_s, pitch P_s, bolt length l_s, head width K_s and root diameter of the thread d_{s1};

** 2 – single clutch disk, 4 – two clutch disks

4.2. Objective functions

The objective functions chosen for this application are the mass of the whole flywheel-clutch assembly and the moment of inertia.

To calculate the mass of this assembly becomes a very difficult task because of the complexity of the construction. There are a lot of parts and their shapes are very complicated. In order to compute the mass of the assembly (as well as of the moment of inertia) the construction was decomposed into simple geometrical bodies. These simple bodies are cylinders or tapers with or without cylindrical or taper holes. In fig. 3 the decomposition of the assembly into simplified geometrical bodies is presented.

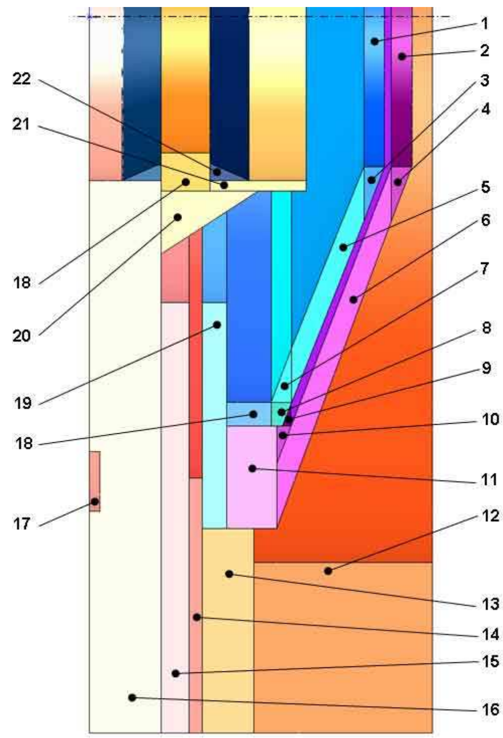

Fig.3. Decomposition of the flywheel assembly

Obj.1 The mass of the flywheel-clutch assembly:

$$M = \sum_{k=1}^{22} (-1)^{\delta(k)} \cdot m_k \to \min \tag{1}$$

where:

k – index of the simplified body;

m_k – mass of the body *k*, [kg].

$$\delta(k) = \begin{cases} 0, \text{ if the body } k \text{ is solid} \\ 1, \text{ if the body } k \text{ is void} \end{cases} \tag{2}$$

Obj.2 The moment of inertia of whole system:

$$J = \sum_{k=1}^{22} (-1)^{\delta(k)} \cdot J_k \to \max \tag{3}$$

where:

J_k – moment of inertia of k part.

4.3. Constraints

After a carefully analysis, there were identified 15 important constraints. These constraints were designed in order to:

- Provide a continuous well operating of the driving system;
- Carry out a correct mounting;
- Satisfy the requirements arose from the strength of materials;
- Use the actual compressed air pipe-line network.

All the values of these constraints have to be negative or zero. The imposed constraints are as follows:

C1. The diameter D_3 of the hole of the clutch pressure plate should be less than the outer diameter D_4 of the taper part of the clutch pressure plate.

$$g_1 = \frac{D_3 + 40}{D_4} - 1 \leq 0 \tag{4}$$

C2. The major diameter D_9 of the inner toothed crown wheel has to be inferior to the diameter D_{10} of the circular recess of the flywheel.

$$g_2 = \frac{D_9 + 20}{D_{10}} - 1 \leq 0 \tag{5}$$

C3. The diameter D_{10} of the circular recess of the flywheel must be less than the maximum diameter D_{11} of the flywheel.

$$g_3 = \frac{D_{10} + 20}{D_{11}} - 1 \leq 0 \tag{6}$$

C4. The outer diameter D_{12} of the circular recess of the flywheel has to be less than the major diameter D_{11} of the flywheel.

$$g_4 = \frac{D_{12} + 20}{D_{11}} - 1 \leq 0 \tag{7}$$

C5. The inner diameter D_{13} of the circular recess of the flywheel should be less than the root diameter D_7 of the involute splines of the inner toothed crown wheel.

$$g_5 = \frac{D_{13} + 40}{D_7} - 1 \leq 0 \tag{8}$$

C6. The fitting diameter D_5 of the clutch pressure plate has to be less than the tip diameter D_{a2} of the involute splines of the inner toothed crown wheel (this diameter is equal to the outer diameter of the clutch disk).

$$g_6 = \frac{D_5 + 40}{D_{a2}} - 1 \leq 0 \tag{9}$$

C7. It is mandatory to use standardized bolts (see notes beneath description of the genes).

$$g_7 = \frac{70}{l_s - 9.5 \cdot P_s - b_{13} - b_{14} - (b_6 - b_4)} - 1 \leq 0 \tag{10}$$

C8. The width b_7 of the flywheel (without the inner recess, and corresponding to the fitting zone of the clutch pressure plate) must be inferior to the width b_8 of the flywheel.

$$g_8 = \frac{b_7}{b_8 - 1} \leq 0 \tag{11}$$

Constraints related to the clutch engagement time: the clutch engagement time t_a must be within a certain range:

C9. The clutch engagement time t_a has to be at least 4 seconds.

$$g_9 = \frac{4}{t_a} - 1 \le 0 \tag{12}$$

C10. The clutch engagement time t_a should be 8 seconds at the very most.

$$g_{10} = \frac{t_a}{8} - 1 \le 0 \tag{13}$$

where:

$$t_a = \frac{3 \cdot \omega \cdot J_a \cdot \left(D_{a2}^2 - D_{a1}^2\right)}{z_a \cdot \mu_a \cdot F \cdot \left(D_{a2}^3 - D_{a1}^3\right)} \tag{14}$$

$$F = \frac{\pi \cdot D_5^2}{4} \cdot p_{air} \tag{15}$$

Constraints related to the strength of material:

C11. The disk (or the disks) of the clutch has to withstand to the crush.

$$g_{11} = \frac{\sigma_{sa}}{\sigma_{saA}} - 1 \le 0 \tag{16}$$

where:

σ_{sa} – crushing stress between the acting faces of the clutch, [MPa]:

$$\sigma_{sa} = \frac{4 \cdot F}{\pi \cdot \left(D_{a2}^2 - D_{a1}^2\right)} \tag{17}$$

C12. The acting faces of the splines must withstand to the crush.

$$g_{12} = \frac{\sigma_{sc}}{\sigma_{scA}} - 1 \le 0 \tag{18}$$

where:

σ_{sc} – crushing stress between the acting faces of the splines, [MPa]:

$$\sigma_{sc} = \frac{J_a \cdot \omega}{0.4 \cdot k_f \cdot t_a \cdot m^2 \cdot z^2 \cdot (z_a - 2) \cdot (b_{13} - 2)} \tag{19}$$

C13. The rods of the bolts must withstand to the traction (it was taken into account the preload too).

$$g_{13} = \frac{\sigma_{ts}}{\sigma_{tsA}} - 1 \le 0 \tag{20}$$

where:

$$\sigma_{ts} = \frac{4 \cdot 1.1 \cdot F}{z_s \cdot \pi \cdot d_{s1}^2} \tag{21}$$

C14. The centrifugal stress arose in the flywheel should not exceed a certain limit.

$$g_{14} = \frac{\sigma_{cf}}{\sigma_{cfA}} - 1 \le 0 \tag{22}$$

Constraint related to the moment of inertia:

C15. The moment of inertia J of the re-designed assembly should be at least J_i (the actual level).

$$g_{15} = \frac{J_i}{J} - 1 \le 0 \tag{23}$$

where:

$$J = \sum_{k=1}^{22} (-1)^{\delta(k)} \cdot J_k \qquad (24)$$

k – index of the simplified body;
J_k – moment of inertia of the body k, [kg·mm^2].

4.4. Results

The optimal Pareto set was obtained using Cambrian v.2.1 software belonging to the Optimal Design Centre of the Technical University of Cluj-Napoca. The resulted Pareto front is presented in fig. 4. For a better interpretation of results in fig. 5 is presented the Pareto front of the actual design solution.

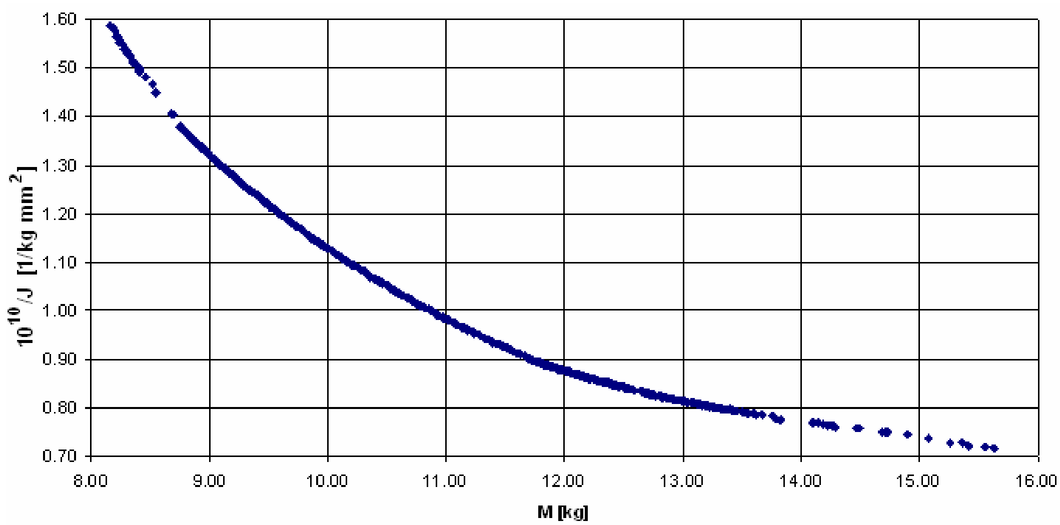

Fig.4. Pareto front (mass vs. 10^{10}/J)

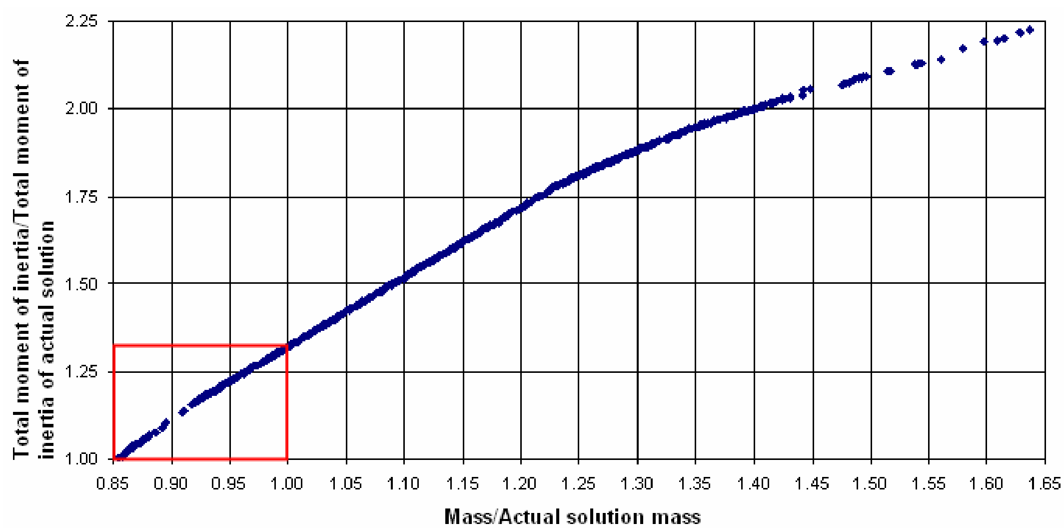

Fig.5. Pareto front (M/M$_i$ vs. J/J$_i$)

4.5. Conclusions

In table 1 is presented the values of the genes for the optimal solution with the lower mass and in the second table is presented a comparison between the values of the objective functions for the actual design solution and for the solution with the lower mass.

All the solutions that are into the red border from the fig. 5 are better then the initial design solution.

Table 1. Values of genes for the solution with the lower mass

No.	Gene	Symbol	Value
1	Number of the bolts	z_s	6
2	Bolt type	M30	d_s = 30 mm, P_s = 3,5 mm, l_s = 240 mm, K_s = 30 mm, d_{s1} = 26,211 mm
3	Number of acting faces of the clutch	z_a	2
4	Module of the involute splines	m	10 mm
5	Number of splines	z	156
6	Inner diameter of the clutch disk	$D_3 = D_{a1}$	989 mm
7	Fitting diameter of the clutch pressure plate	D_5	1466 mm
8	Inner diameter of the flywheel	D_{10}	1962 mm
9	Outer diameter of the flywheel	D_{11}	2423 mm
10	Outer diameter of the circular recess of the flywheel	D_{12}	2023 mm
11	Inner diameter of the circular recess of the flywheel	D_{13}	1000 mm
12	Width of the flywheel	b_8	253 mm

Table 2. Comparison between the solution with the lower mass and the actual design solution

	The lightest design solution	Actual design solution	Variation
Moment of inertia [kg·mm^2]	$J_M = 6.292 \cdot 10^9$	$J_i = 6.266 \cdot 10^9$	0.42 %
Mass [tones]	$M_{\min} = 8{,}162$	$M_i = 9{,}458$	14,5 %

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