OPTIMAL DESIGN OF THE DRIVING SYSTEM OF THE 630 T MAXIPRESS

Abstract: *In this paper the optimal (re)design of the driving system of the 630 t Maxipress is presented. We focused on the flywheel-clutch assembly that actually weights 9.458 tones. The main goal of the optimization is to reduce this mass, and keep the moment of inertia at least at the actual level. After optimization a mass of 8.410 tones was obtained, that represents a diminution of 11%. The moment of inertia varied only about 2%.*
The optimization method used for the optimal design was the genetic algorithms.

1. INTRODUCTION

The 630 t Maxipress is a high capacity hammering press. 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.
We focused 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 clutch pressure plate (4), the taper-roller bearings (5), the key (6), the shaft (7), the hub (8) and the sealing plate (9).
Obviously, in the specification of this assembly the moment of inertia and the mass are very important. It is worthy to mention that the actual moment of inertia of the flywheel-clutch assembly is of $6.266 \cdot 10^9$ kg·m^2, and its mass is of 9.458 tones.

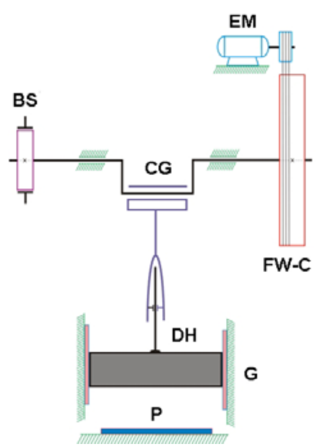


Fig.1. Functional scheme of the 630 t Maxipress

To keep up the moment of inertia at least at the same level and to reduce this enormous mass becomes a provocative challenge.

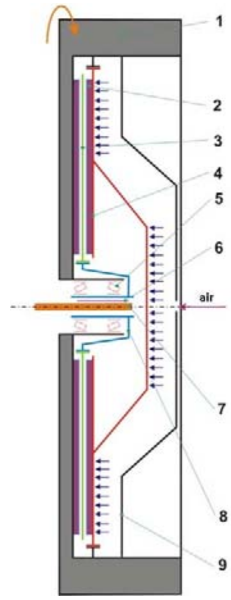


Fig.2. Sketch of the flywheel-clutch assembly

In addition, we propose to retain the core of the assembly (in order to be able to assemble the redesigned parts on the same bearing) and the shape of all these parts.
In Figure 3 the SolidWork model of the flywheel-clutch assembly is presented. In this figure the numbering represents: the flywheel rim (1), the inner toothed crown wheel (2), the clutch pressure plate (3) with wearing plate (4), the clutch disk (5), the hub (6), the taper-roller bearings (7), the sealing plate (8), the shaft, the threaded fasteners (10-11), the clutch disengaging springs (12), and the stud-nut assemblies of the springs (13-14).
In Figure 4 the drawing of the flywheel-clutch assembly is shown. In this figure one could follow the main geometrical dimensions used in the following equations.

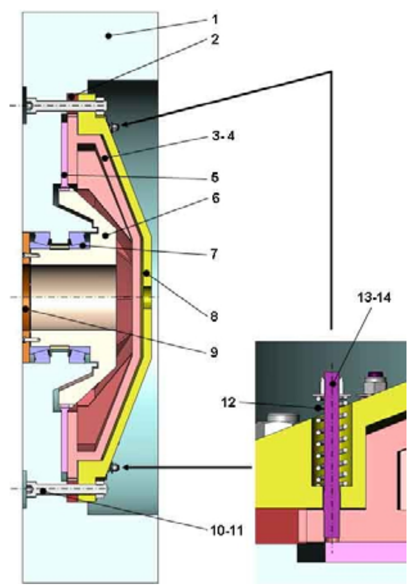


Fig.3. Model of the flywheel-clutch assembly

2. OPTIMAL DESIGN

In order to perform the optimal design of the flywheel-clutch assembly is necessary to set up:
- the objective function;
- the constraints of the problem;
- the variables (genes) that uniquely describe the problem (both the objective function and the constraints).

2.1. Objective function

The objective function chosen for this application is the mass of the whole flywheel-clutch assembly. Obviously, we want to obtain a minimal mass (as reduced as possible).
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.

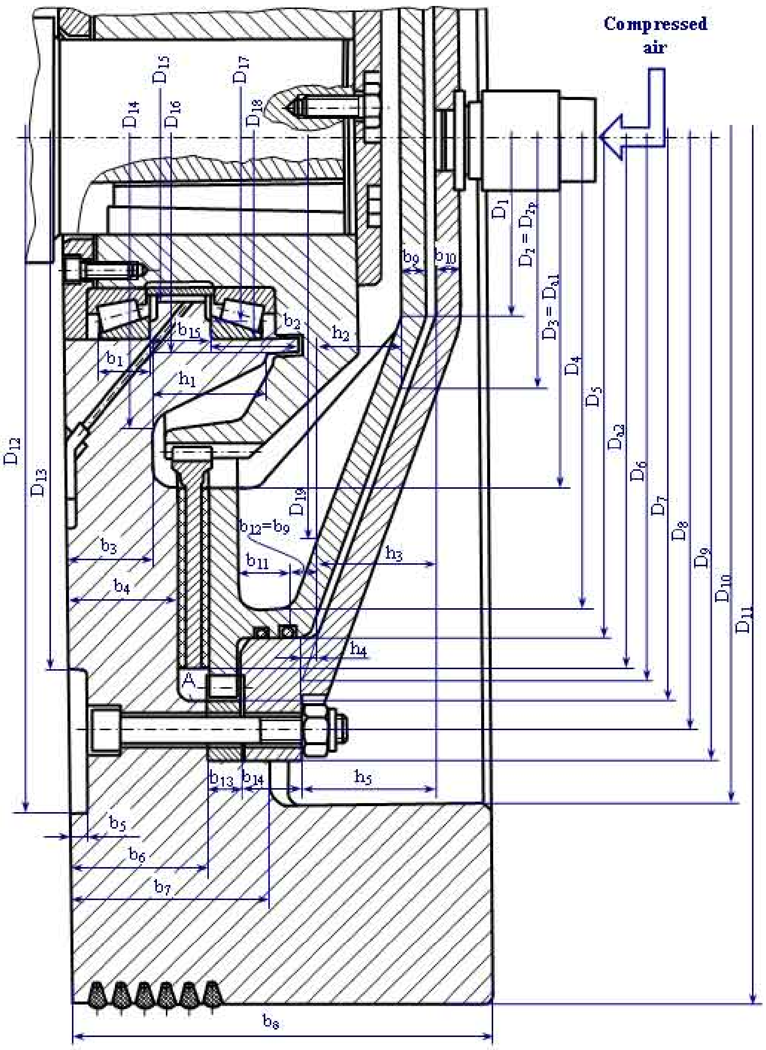


Fig.4. Drawing of the flywheel-clutch assembly

These simple bodies are cylinders or tapers with or without cylindrical or taper holes. In Figure 5 the decomposition of the assembly into simplified geometrical bodies is presented.

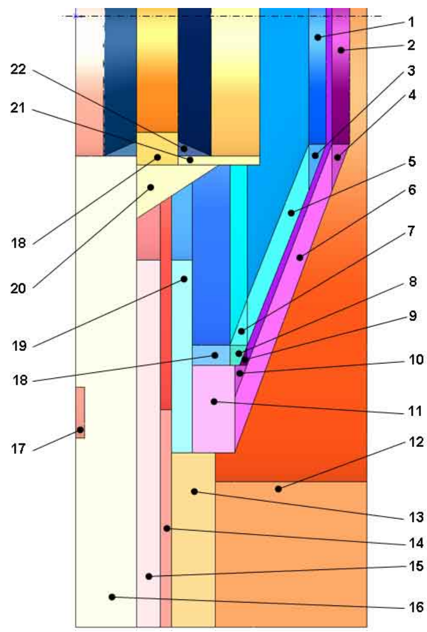


Fig.5. Decomposition of the flywheel assembly

As one can observe there was identified 22 such as simplified bodies. Consequently, the mass of the assembly is given by the following equation:

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

where:
k – index of the simplified body;
m_k – mass of the body k [kg].

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

In the case of the calculus of the moment of the inertia, the same convention was used too.

2.2. 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.

The imposed constraints are as follows:

Geometrical constraints:

R 1. 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:

$$g_1 = \frac{D_3 + 40}{D_4} - 1 \le 0 \qquad (3)$$

R 2. 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:

$$g_2 = \frac{D_9 + 20}{D_{10}} - 1 \le 0 \qquad (4)$$

R 3. The diameter D_{10} of the circular recess of the flywheel must be less than the maximum diameter D_{11} of the flywheel:

$$g_3 = \frac{D_{10} + 20}{D_{11}} - 1 \leq 0 \qquad (5)$$

R 4. 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:

$$g_4 = \frac{D_{12} + 20}{D_{11}} - 1 \leq 0 \qquad (5)$$

R 5. 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:

$$g_5 = \frac{D_{13} + 40}{D_7} - 1 \leq 0 \qquad (6)$$

R 6. 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):

$$g_6 = \frac{D_5 + 40}{D_{a2}} - 1 \leq 0 \qquad (7)$$

R 7. It is mandatory to use standardized bolts (see Notes beneath Table 8):

$$g_7 = \frac{70}{l_s - 9.5 \cdot P_s - b_{13} - b_{14} - (b_6 - b_4)} - 1 \leq 0 \qquad (8)$$

R 8. 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:

$$g_8 = \frac{b_7}{b_8 - 1} \leq 0 \qquad (9)$$

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

R9. The clutch engagement time t_a has to be at least 4 seconds:

$$g_9 = \frac{4}{t_a} - 1 \leq 0 \qquad (10)$$

R10. The clutch engagement time t_a should be 8 seconds at the very most:

$$g_{10} = \frac{t_a}{8} - 1 \leq 0 \qquad (11)$$

where:

$$t_a = \frac{3 \cdot \omega \cdot J_a \cdot \left(D_{a2}^2 - D_{a1}^2\right)}{z_a \cdot \mu_a \cdot F \cdot \left(D_{a2}^3 - D_{a1}^3\right)} \qquad (12)$$

ω – angular speed ($\omega = 10.472\ s^{-1}$);
J_a – moment of inertia of the parts of the press that have to be set on ($J_a = 1.567 \cdot 10^9$ kg·mm^2);
μ_a – coefficient of friction ($\mu_a = 0.35$);
F – axial force [N]:

$$F = \frac{\pi \cdot D_5^2}{4} \cdot p_{air} \qquad (13)$$

p_{air} – compressed air pressure ($p_{air} = 0.5$ MPa);

Constraints related to the strength of material:

R11. The disk (or the disks) of the clutch has to withstand to the crush:

$$g_{11} = \frac{\sigma_{sa}}{\sigma_{saA}} - 1 \leq 0 \qquad (14)$$

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

$$\sigma_{sa} = \frac{4 \cdot F}{\pi \cdot \left(D_{a2}^2 - D_{a1}^2\right)} \quad (15)$$

σ_{saA} – allowable crushing stress (σ_{saA} = 3 MPa).

R12. The acting faces of the splines must withstand to the crush:

$$g_{12} = \frac{\sigma_{sc}}{\sigma_{scA}} - 1 \le 0 \quad (16)$$

where:

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

$$\sigma_{sc} = \frac{J_a \cdot \omega}{0.4 \cdot k_f \cdot t_a \cdot m^2 \cdot z^2 \cdot (z_a - 2) \cdot (b_{13} - 2)} \quad (17)$$

where:

k_f – coefficient related to the technology (k_f = 0.8) ;
σ_{scA} – allowable crushing stress (σ_{scA} = 30 MPa).

R13. The rods of the bolts must withstand to the traction (it was taken into account the preload too):

$$g_{13} = \frac{\sigma_{ts}}{\sigma_{tsA}} - 1 \le 0 \quad (18)$$

where:

$$\sigma_{ts} = \frac{4 \cdot 1.1 \cdot F}{z_s \cdot \pi \cdot d_{s1}^2} \quad (19)$$

σ_{tsA} – allowable tensile stress (σ_{tsA} = 80 MPa).

R14. The centrifugal stress arose in the flywheel should not exceed a certain limit:

$$g_{14} = \frac{\sigma_{cf}}{\sigma_{cfA}} - 1 \le 0 \quad (20)$$

σ_{tvA} – allowable tensile stress of the flywheel material (σ_{tvA} = 80 MPa).

Constraint related to the moment of inertia:

R15. The moment of inertia J of the re-designed assembly should be at least J_i (the actual level):

$$g_{15} = \frac{J_i}{J} - 1 \le 0 \quad (21)$$

where:

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

k – index of the simplified body;
J_k – moment of inertia of the body k [kg·mm^2];
J_i – actual moment of inertia of the flywheel-clutch assembly ($J_i = 6.266 \cdot 10^9$ kg·mm^2).

2.3. Genes

The final 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).

We consider that there are 12 genes that can describe completely the optimization problem. These genes are listed in the Table 1. All the genes can have only integer values.
As one can observe the values of the gene *TS* are integer numbers that code several lists that contain all the standardized values related to a bolt (nominal diameter of the thread, pitch, bolt length, head width and root diameter of the thread).
Taking into account the huge dimension of the flywheel (and consequently of the clutch) the module *m* of the involute splines of the inner toothed crown wheel can takes only two standardized values: 10 or 12.5 mm.
The other genes do not ask for standardized value, but we consider only the integer values in the mentioned range (see Table 1). Any integer in the range could be a possible value of the respective gene.
Eventually, we mention that only the alternative of single or two clutch disk was taken into consideration because, as we already noted, the core of the assembly must remain unmodified.

Table 1. Genes of the optimization program

No	Denotation		Range	MU
0	Number of bolts	z_s	8 ... 36	-
1	Bolt type*	*TS*	0 ... 31	-
2	Number of acting faces of the clutch	z_a	2, 4**	-
3	Module of the involute splines	m	10, 12.5	mm
4	Number of splines	z	100 ... 250	-
5	Inner diameter of the clutch disk	$D_3 = D_{a1}$	800 ... 1100	mm
6	Fitting diameter of the clutch pressure plate	D_5	1000 ... 1600	mm
7	Inner diameter of the flywheel	D_{10}	1000 ... 2000	mm
8	Outer diameter of the flywheel	D_{11}	1500 ... 2500	mm
9	Outer diameter of the circular recess of the flywheel	D_{12}	1000 ... 2000	mm
10	Inner diameter of the circular recess of the flywheel	D_{13}	800 ... 1500	mm
11	Width of the flywheel	b_8	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.

3. OPTIMIZATION AND RESULTS

3.1. Kreator v. 1.0

We used Kreator v.1.0 software in order to perform the optimization and to finalize on this basis the new design of the driving system of the 630 t Maxipress.
The main specification and the settings of the *Kreator* software are presented in Table 2.

Table 2. Settings of the Kreator software

Specification	**Type or Value**
Representation of genes	Real
Method of fitness assignment	Dynamic
Method of selection	Roulette wheel
Number of individuals	100
Crossover rate	0.75
Mutation rate	0.05

3.2. Optimization results

The values of all the considered genes, after optimization, are given in Table 3 and in Table 4 the specifications of the actual and optimal solution are shown.

Table 3. Results of the optimization

<table>
<tr><th>Gene</th><th colspan="3">Denotation</th><th>Value</th><th>MU</th></tr>
<tr><td>0</td><td colspan="2">Number of bolts</td><td>z_s</td><td>32</td><td>-</td></tr>
<tr><td rowspan="5">1</td><td rowspan="5">Bolt type (M30)</td><td>Nominal diameter of the thread</td><td>d_s</td><td>30</td><td>mm</td></tr>
<tr><td>Pitch of the thread</td><td>P_s</td><td>3.5</td><td>mm</td></tr>
<tr><td>Bolt length</td><td>l_s</td><td>240</td><td>mm</td></tr>
<tr><td>Head width</td><td>K_s</td><td>30</td><td>mm</td></tr>
<tr><td>Root diameter of the thread</td><td>d_{s1}</td><td>26.211</td><td>mm</td></tr>
<tr><td>2</td><td colspan="2">Number of acting faces of the clutch</td><td>z_a</td><td>2</td><td>-</td></tr>
<tr><td>3</td><td colspan="2">Module of the involute splines</td><td>m</td><td>10</td><td>mm</td></tr>
<tr><td>4</td><td colspan="2">Number of splines</td><td>z</td><td>169</td><td>-</td></tr>
<tr><td>5</td><td colspan="2">Inner diameter of the clutch disk</td><td>$D_3 = D_{a1}$</td><td>900</td><td>mm</td></tr>
<tr><td>6</td><td colspan="2">Fitting diameter of the clutch pressure plate</td><td>D_5</td><td>1510</td><td>mm</td></tr>
<tr><td>7</td><td colspan="2">Inner diameter of the flywheel</td><td>D_{10}</td><td>1890</td><td>mm</td></tr>
<tr><td>8</td><td colspan="2">Outer diameter of the flywheel</td><td>D_{11}</td><td>2357</td><td>mm</td></tr>
<tr><td>9</td><td colspan="2">Outer diameter of the circular recess of the flywheel</td><td>D_{12}</td><td>1828</td><td>mm</td></tr>
<tr><td>10</td><td colspan="2">Inner diameter of the circular recess of the flywheel</td><td>D_{13}</td><td>1000</td><td>mm</td></tr>
<tr><td>11</td><td colspan="2">Width of the flywheel</td><td>b_8</td><td>293</td><td>mm</td></tr>
</table>

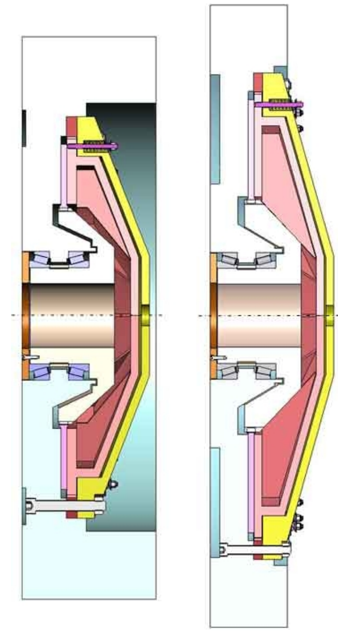
Table 4. Specifications of the actual and optimal solution

Denotation		**Value**		**MU**
		Actual	**Optimal**	
Number of bolts	z_s	16	32	-
Nominal diameter of the thread	d_s	36	30	mm
Number of acting faces of the clutch	z_a	2	2	-
Module of the involute splines	m	10	10	mm
Number of splines	z	134	169	-
Inner diameter of the clutch disk	$D_3 = D_{a1}$	838	900	mm
Fitting diameter of the clutch pressure plate	D_5	1200	1510	mm
Inner diameter of the flywheel	D_{10}	1600	1890	mm
Outer diameter of the flywheel	D_{11}	2100	2357	mm
Outer diameter of the circular recess of the flywheel	D_{12}	1550	1828	mm
Inner diameter of the circular recess of the flywheel	D_{13}	1275	1000	mm
Width of the flywheel	b_8	500	293	mm
Moment of inertia of the assembly	J	6,266	6,391	$kg\cdot m^2$
Mass of the assembly	M	9.458	8.410	tones

In Figure 6 the comparison side-by-side of the actual construction (a) and of the optimal solution (b) is presented. The real proportions of the constructions were respected.

4. CONCLUSION

After optimization the mass of the assembly was reduced with 11% that represents an excellent result, even that the outer diameter of the flywheel increased with 257 mm (from 2100 to 2357). Note that the press is 6 m high. In the same time the moment of inertia was maintained about at the same level. In fact, the moment of inertia even increased with about 2%.



a) b)

Fig.6. Comparison side-to-side of the design solutions