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Minimum cost design of a welded punch press for light industry

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Abstract: A punch press table consists of a box beam the upper flange of which is an orthogonally stiffened plate. In order to guarantee the exact punching the structure should fulfil the requirements of global and local stiffness. All welds should be checked for fatigue stress range. The verification of the original structural version shows that cost savings can be achieved decreasing the plate thicknesses and the number of stiffeners. The calculation shows that the bending deflection of stiffeners can be neglected compared to the shear one. Thus, in the analysis of the grid of stiffeners the force method can be used considering only the shear deflections. The cost comparison of the original and modified version shows a significant cost saving.

Keywords: minimum cost design, structural optimization, welded structures, press table, stiffened plates

1. Introduction

Punch presses are used in light industry for punching of various nonferrous materials such as leather, plastics, paper, textile etc. The main problem for the supporting structure is to guarantee a very stiff plane surface, deflections of which caused by high pressing load are very small. A press consists of two main parts as follows: the table is a box beam with an upper flange constructed as a stiffened plate, and the mobile bridge, which is a similar box beam with a stiffened lower flange. The box beams produce the high stiffness for the whole structure and the stiffened flanges guarantee the local stiffness of the plane surfaces.

Punching presses are fabricated in Hungary by the company Schön Engineering Ltd. The original welded structure is a well designed version, the prescribed deflections of which have been verified by measurements on a real machine.

Our aim is to show that, by using our minimum cost design procedure, it is possible to find a cheaper structural version. We treat here the optimum design of the table beam only, since the design of the mobile bridge can be calculated similarly.

Optimum design of welded machine structures appears in the literature very rarely. Farkas [1,2] has treated the optimum design of a closed press frame constructed from welded box beams and columns. The cost calculation method of welded structures developed by Farkas and Jármai [3,4] and used for various types of structures (stiffened plates [5], silos [6], bunkers [7], conical roofs [8], tubular Vierendeel trusses [9], bridge decks [10]) enables us to compare the original and the optimized table beams and show the achievable cost saving.

2. Constraints on stiffness and fatigue stress range

The maximum prescribed beam deflection due to pressing force of $F = 1600$ kN is

$$w_{max} = 0.50 \text{ mm} \quad (1)$$

The maximum prescribed local deflection is

$$w_L = 0.12 \text{ mm} \quad (2)$$

The pressing force is distributed in a square surface with sides of 400x400 mm, thus, the intensity of the uniformly distributed normal pressing load is $p = 10$ MPa.

The welded joints should fulfil the constraint on fatigue stress range for the number of cycles $N_f = 10^7$.

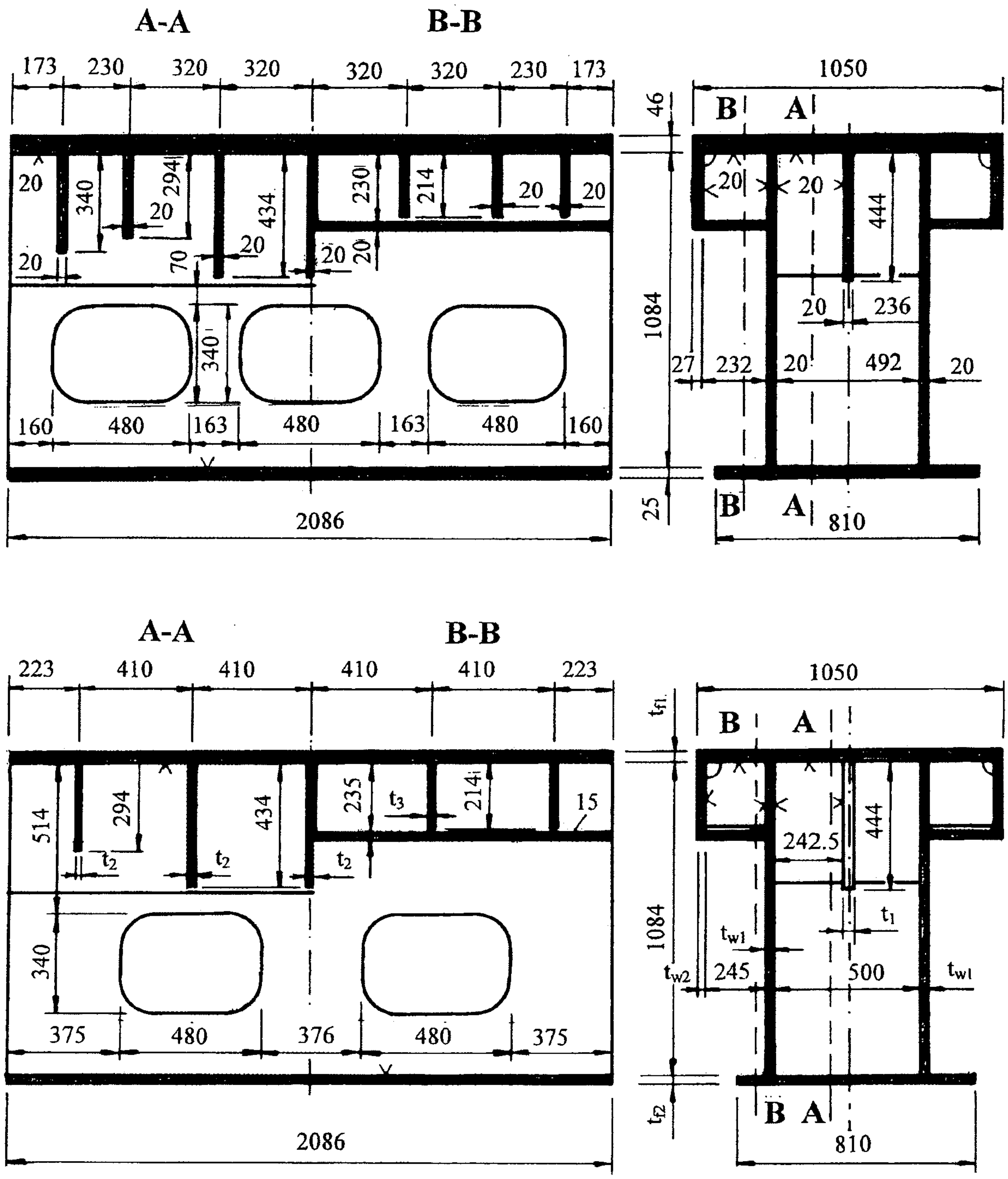

Fig. 1. Welded table beam of a punch press. (a) The original structure, (b) the optimized structure

3. Verification of the original table beam

3.1 Local deflection of the upper flange plate

The flange thickness is $t_F = 46$ mm (Fig. 1a). A flange field of side dimensions 320x236 mm is subjected to bending by $p = 10$ MPa. It is assumed that its edges are clamped. The maximum local deflection can be calculated according to [11] with values of $a = 236$ mm and the elastic modulus $E = 2.1\text{x}10^5$ MPa

$$w_L = \frac{0.0214pa^4}{Et_F^3} = 0.0325 < 0.12 \text{ mm} \qquad (3)$$

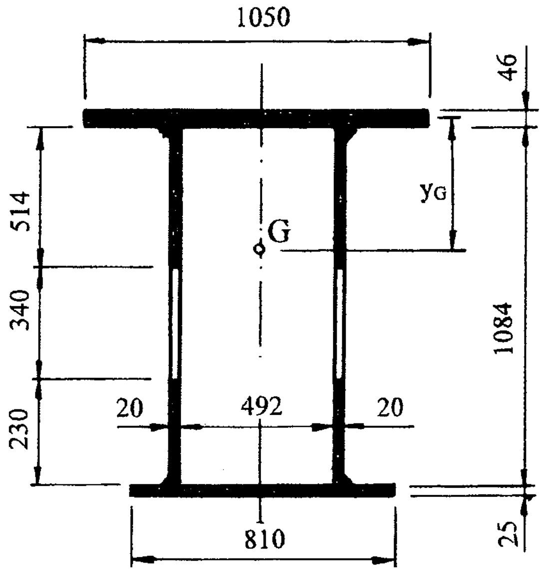

Fig.2. Cross-section of the original box beam through holes

3.2 Deflection of the whole box beam

For the calculation of deflection it is sufficient to consider a simply supported beam of cross-section shown in Fig.2. For the sake of assembly and inside welding the three holes have been constructed in webs (Fig.1a and 2). It is on the safety side when we consider the cross-section through holes.

The distance of the center of gravity is

$$y_G = 384 \text{ mm} \qquad (4)$$

The moment of inertia is

$$I_x = 1.8751\text{x}10^{10} \text{ mm}^4 \qquad (5)$$

The maximum deflection due to bending, with a span length of $L = 2086$ mm, is

$$w_b = \frac{FL^3}{48EI_x} = 0.0768 \text{ mm} \qquad (6)$$

The maximum deflection due to shear, using the shear modulus of G = $0.8077\text{x}10^5$ can be calculated as

$$w_S = \frac{800x10^3 x323}{0.8077x10^5 x40x1084} + \frac{800x10^3 x720}{0.8077x10^5 x40x744} = 0.3134 \text{ mm} \tag{7}$$

The total deflection is

$$w_{\max} = w_b + w_S = 0.3902 < 0.5 \text{ mm, OK.} \tag{8}$$

3.3 Fatigue of welded joints

The longitudinal K-butt welds should be checked for fatigue due to normal and shear stresses. According to Eurocode 3 (EC3) [12] the fatigue stress ranges for $N_f = 2x10^6$ are as follows:

$$\Delta\sigma_C = 100, \Delta\tau_C = 80 \text{ MPa} \tag{9}$$

For $N_f=10^7$ one obtains $\Delta\sigma_N = 64, \Delta\tau_N = 58$ MPa. The safety factor is $\gamma_{Mf} = 1.25$.

The maximum bending moment is

$$M_{max} = FL/4 = 8.344x10^8 \text{ Nmm} \tag{10}$$

The maximum normal stress in longitudinal welds at lower flange is

$$\sigma_{\max} = M(1084 - y_G)/I_x = 31.1 \text{ MPa} \tag{11}$$

The average shear stress in webs is

$$\tau = \frac{800x10^3}{40x744} = 26.9 \text{ MPa} \tag{12}$$

According to EC3

$$\left(\frac{\sigma_{\max}}{\Delta\sigma_N / \gamma_{Mf}}\right)^3 + \left(\frac{\tau}{\Delta\tau_N / \gamma_{Mf}}\right)^5 = 0.2896 < 1, \text{ OK.} \tag{13}$$

The K-butt welds at upper flange should be checked for normal compression stress due to pressing load. For this case EC3 prescribes a category of $\Delta\sigma_C = 71$ MPa. For $N_f = 10^7$ we obtain $\Delta\sigma_N / \gamma_{Mf} = 45.5/1.35 = 33.7$ MPa. The actual pressing stress is

$$10 < 33.7 \text{ MPa, OK.} \tag{14}$$

3.4 Local deformation of inner stiffeners in the upper flange

It has been already shown [1,2] that the shear deformation is governing when the beams are very short and the concentrated load is very high. To illustrate this fact consider first a stiffener shown in Fig.3. Defining the ratio of

$$\omega = \frac{320x46}{434x20} = 1.6958 \tag{15}$$

one obtains for the moment of inertia

$$I_x = \frac{434^3 x20}{12}\frac{1+4\omega}{1+\omega} = 3.9336x10^8 \text{ mm}^4 \tag{16}$$

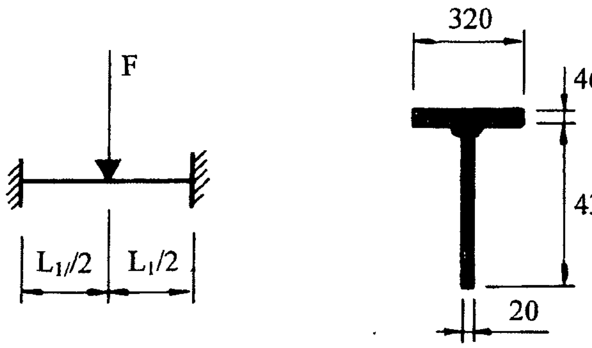

Fig.3. A transverse stiffener

Considering the stiffener as a beam with clamped supports, the maximum deflection due to bending is ($L_1 = 492$ mm)

$$w_b = \frac{F L_1^3}{192EI_x} = 0.0120 \text{ mm} \tag{17}$$

and the shear deflection is ($A_w = 8680$ mm^2)

$$w_S = \frac{FL_1}{4GA_w} = 0.2807 \text{ mm} \tag{18}$$

It can be seen that the bending deflection is only 4% of the shear one, thus, it can be neglected.

The resulting deflection of a grid of stiffeners can be calculated considering the shear deformations only. Using the force method the unknown forces X_1 and X_2 acting at points A and B (Fig.4) can be determined from the deflection equations expressing that the deflections w_A and w_B are the same for the connecting two beams as follows:

$$w_A = \frac{(F - X_1)B}{4GA_1} - \frac{X_2 b}{GA_1} = \frac{X_1 L_1}{4GA_2} \tag{19}$$

$$w_B = \frac{(F - X_1)b}{2GA_1} - \frac{X_2 b}{GA_1} = \frac{X_2 L_1}{4GA_2} \tag{20}$$

where A_1 and A_2 are the cross-sectional areas of stiffener webs.

The solution of Equations (19) and (20) is

$$X_1 = \frac{F}{3 + \frac{L_1 A_1}{2bA_2}}; X_2 = \frac{F}{2} - X_1\left(\frac{1}{2} + \frac{L_1 A_1}{4bA_2}\right) \tag{21}$$

In our case it is $F = 1600$ kN, $L_1 = 492$, $b = 320$ mm, $A_1 = 444x20$, $A_2 = 434x20$, thus $X_1 = X_2 = 4.225x10^5$ N. The maximum deflection is

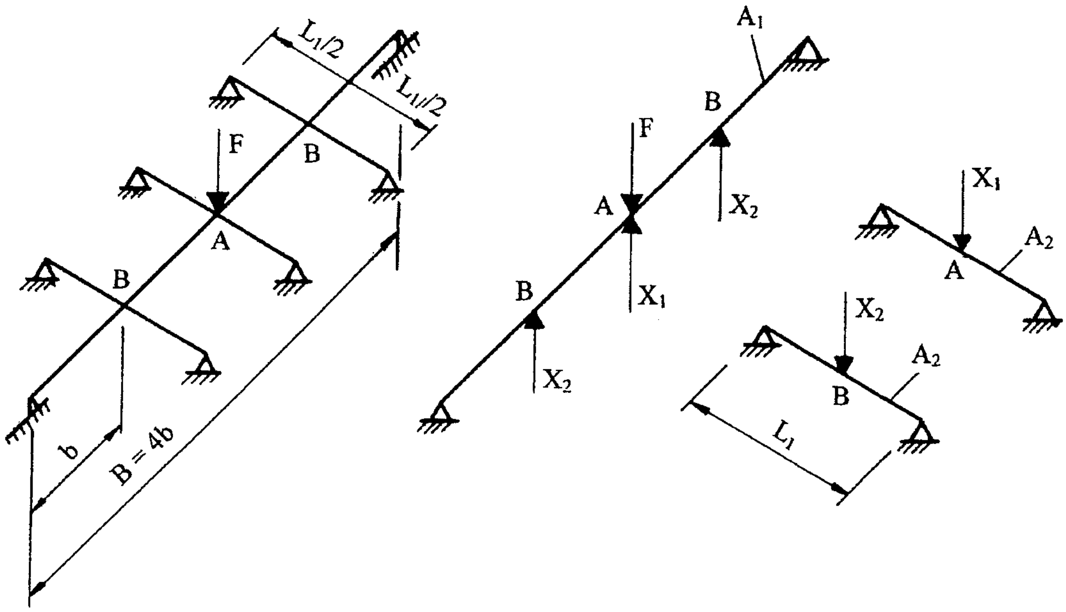

Fig.4. Grid model of inner stiffeners

$w_A = 0.041 < 0.12$ mm, OK. (22)

and the shear stress, considering a safety factor of 1.35 instead of 1.25, is

$$\tau = \frac{X_1}{2A_2} = 24.3 < 43.0 \text{ MPa, OK.} \tag{23}$$

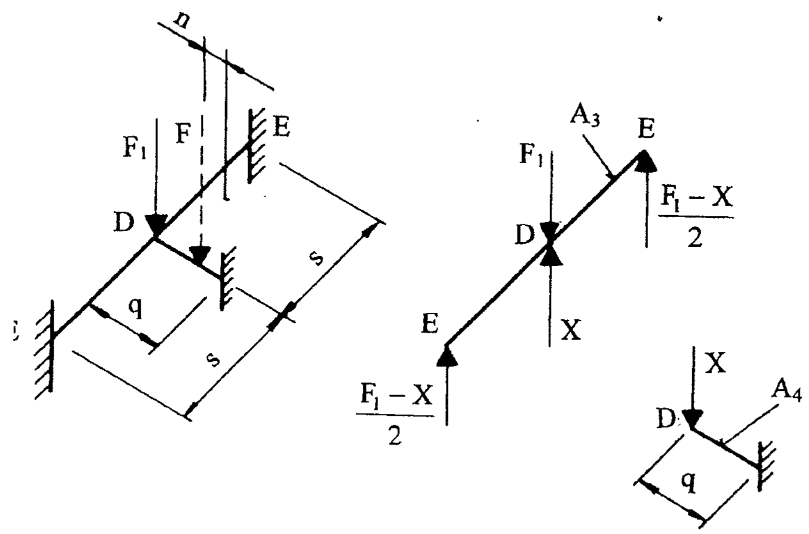

Fig.5. Grid model of outer stiffeners

3.5 Outer stiffeners of the upper flange

The pressing load F is acting on a transverse stiffener in a distance of $n=q-200=32$ mm. From this action a force of $F_1 = 32F/232 = 220.7\text{x}10^3$ N arises at point D. Similarly to Section 3.4, we calculate the shear deformation using the deflection equation of a beam system shown in Fig.5.The shear deflection equation for point D is

$$w_D = \frac{Xq}{GA_4} = \frac{(F_1 - X)s}{2GA_3} \tag{24}$$

The solution of (24) is

$$X = \frac{F_1}{\frac{2A_3 q}{A_4 s} + 1} \tag{25}$$

With q = 232, s = 320 mm, A_3 = 6318, A_4 = 4280 mm^2 one obtains X = 70.3x10^3 N. According to (24) the maximum deflection is

$$w_D = 0.047 < 0.12 \text{ mm, OK.} \tag{26}$$

The shear stress in the transverse stiffener is

$$\tau = \frac{70.3x10^3}{214x20} = 16 < 43.0 \text{ MPa, OK.}$$

(27)

and in the longitudinal stiffener is

$$\tau = \frac{(220.7 - 70.3)10^3}{27x234} = 24.0 < 43.0 \text{ MPa, OK.} \tag{28}$$

3.6 Cost calculation

The cost is calculated disregarding the fabrication sequence. It is assumed that the welding method for all the butt welds is GMAW-C (Gas Metal Arc Welding with CO_2). The material cost factor is k_m = 1.0 \$/kg, the fabrication cost factor is k_f = 25 \$/h = 0.417 \$/min, thus k_f/k_m = 0.417 kg/min. According to [4], for K-butt welds

$$C_2 a_w^n = 0.152 a_w^{1.9358} \tag{29a}$$

and for ½ V butt welds

$$C_2 a_w^2 = 0.2245 a_w^2 \tag{29b}$$

where a_w is the weld size in mm, the weld lengths should be calculated in m.

The volume is as follows:

upper flange plate	1050x2086x46	100.75x10^6
lower flange plate	810x2086x25	42.24
two webs	2x20x1084x2086	90.45
inner longitudinal stiffener	444x20x2086	18.42

outer longitudinal stiffeners	2x250x27x2086	28,16
outer horizontal plates	2x232x20x2086	19.36
inner transverse stiffeners	2x472x20x1295	24.45
outer transverse stiffeners	14x232x20x214	13.90
The total volume is		$V = 337.73x10^6$
The total mass is	$(\rho = 7.85x10^{-6}$ kg/mm$^3)$	$\rho V = 2651$ kg

Total number of structural elements to be assembled is $\kappa = 37$.

The cost is as follows:

$$K = k_m \rho V + k_f \left(C_1 \Theta_d (\kappa \rho V)^{0.5} + 1.3 \sum_i C_{2i} a_{wi}^n L_{wi} \right) = 2651 +$$

$$+ 0.417 \left[3(37x2651)^{0.5} + 1.3 \left(0.152x20^{1.9358} x28.838 + 0.152x27^{1.9358} x4172 + 0.2245x20^2 x8344 \right) \right] =$$
$$= 4431 \ \$ \qquad (30)$$

4. Optimum design of a new structural version

The verification of the original version shows that some dimensions can be decreased. Since the main table dimensions should remain, the number of upper transverse stiffeners and the thicknesses of plates can be decreased. In an optimum design procedure the thicknesses of $t_{w1}, t_{f1}, t_{f2}, t_1, t_2, t_{w2}, t_3$ are determined (Fig.1b).

4.1 *Thickness of the upper flange plate t_{f1}*

The number of transverse stiffeners is changed from 7 to 5 and their distance is changed from 320 to 410 mm. Thus a flange field has the side dimensions of 410x242.5 mm. The required plate thickness can be calculated according to [11]

$$t_{f1} = \left(\frac{0.026x10x242.5^4}{2.1x10^5 x0.12} \right)^{1/3} = 32.9 \text{ mm} \qquad (31)$$

The value of t_{f1} = 36 mm is selected, since this table plate should be cut to 36 mm from the original thickness of 40 mm.

4.2 *Design of thicknesses t_{w1} and t_{f2}*

We take t_{w1} = 15 and t_{f2} = 20 mm and check the box beam for deflection and fatigue.

Fig.1b shows that the number of web holes is decreased to 2. and we calculate the I_x neglecting the holes.

The distance of the center of gravity is

$$y_G = 419 \text{ mm} \tag{32}$$

The moment of inertia is

$$I_x = 1.8253 \times 10^{10} \text{ mm}^4 \tag{33}$$

The deflection of the table beam due to bending, according to (6), is

$$w_b = 0.0789 \text{ mm} \tag{34}$$

The deflection due to shear, according to (7), is

$$w_s = 0.3845 \text{ mm} \tag{35}$$

The total deflection is

$$w_b + w_s = 0.4634 < 0.50 \text{ mm, OK.} \tag{36}$$

The maximum normal stress in longitudinal welds at lower flange, according to (11), is

$$\sigma_{\max} = 31.2 \text{ MPa} \tag{37}$$

The shear stress, similarly to (12), is

$$\tau = 24.6 \text{ MPa} \tag{38}$$

Check for fatigue stress range, similarly to (13), is expressed as

$$0.2263 + 0.0419 = 0.2682 < 1, \text{ OK.} \tag{39}$$

4.3 *Thicknesses of inner stiffeners t_1 and t_2*

In order to determine the required thicknesses we use the minimum cost design procedure for the simplified grid model shown in Fig.4. The cost function is expressed by the unknown thicknesses t_1 and t_2 as follows:

$$\frac{K}{k_m} = \rho(4bA_1 + 3L_1A_2) + \frac{k_f}{k_m}\left[3(7\rho V)^{0.5} + 1.3x0.152\left(2.528t_1^{1.9358} + 6.663t_2^{1.9358}\right)\right] \tag{40}$$

where $A_1 = 444t_1$ and $A_2 = 434t_2$, $F = 1600$ kN, $L_1 = 500$, $b = 410$ mm, $k_f/k_m = 0.417$ kg/min. Deflection constraint for the point A is expressed as

$$w_A = \frac{X_1 L_1}{4GA_2} \le 0.12 \text{ mm} \tag{41}$$

The shear stress constraint is given by

$$\tau = \frac{X_1}{2A_2} \leq 43.0 \text{ MPa} \tag{42}$$

X_1 is calculated with (21).

In the minimum cost design the optimum thicknesses are sought, which minimize the cost function and fulfil the design constraints.

The optimization procedure is performed by using the Rosenbrock's hillclimb method complemented with a discretization considering the following discrete thicknesses: 10, 12, 15, 20, 25, 30 mm. The result is $t_1 = t_2 = 15$ mm. $X_1 = 441.5 \times 10^3$ N

Check of the constraints:

$$w_A = 0.1050 < 0.12 \text{ mm, OK.} \tag{43}$$

$$\tau = 33.9 < 43.0 \text{MPa, OK.} \tag{44}$$

4.4 *Thicknesses of outer stiffeners t_{w2} and t_3*

In the minimum cost design procedure we use the grid model shown in Fig.5 with the solution given by (25). $F_1 = 45F/245$, $q = 245$, $s = 410$ mm, $A_3 = 250t_{w2}$, $A_4 = 214t_3$.

The cost function is expressed as

$$\frac{K}{k_m} = \rho V + \frac{k_f}{k_m}\left[3(12\rho V)^{0.5} + 1.3x0.152\left(2x2.086t_{w2}^{1.9358} + 10x0.676t_3^{1.9358}\right)\right] \tag{45}$$

where $V = 2x250x2086t_{w2} + 10x245x214t_3$.

Deflection constraint is defined by

$$w_D = \frac{Xq}{GA_4} \leq 0.12 \text{ mm} \tag{46}$$

The shear stress constraint is

$$\tau = \frac{X}{A_4} \leq 43.0 \text{ MPa} \tag{47}$$

The result is $t_{w2} = t_3 = 15$ mm. $X = 122.7 \times 10^3$ N.

Check of the constraints:

$$w_D = 0.1159 < 0.12 \text{ mm, OK.} \tag{48}$$

$$\tau = 38.2 < 43.0 \text{MPa , OK.} \tag{49}$$

4.5 *Cost calculation*

The volume is as follows:

upper flange plate	1050x2086x36	78.8508x10^6
lower flange plate	810x2086x20	33.7932
two webs	2.15x1084x2086	67.8367
inner longitudinal stiffener	444x15x2086	13.8928
outer longitudinal stiffeners	2x250x15x2086	15.6450
outer horizontal plates	2x245x15x2086	15.3321
inner transverse stiffeners	2x242.5x15(2x294+3x434)	13.7498
outer transverse stiffeners	10x245x15x214	5.9578
The total volume is		245.0582x10^6 mm^3.

The total mass of the modified structure is $\rho V = 1924$ kg, this means that the mass saving achieved by optimization is (2651-1924)100/1924 = 38%.

The number of elements to be assembled is $\kappa = 29$.

The cost is as follows:

$K = 1924 + 0.417\left[3(29x1924)^{0.5} + 1.3\left(0.152x15^{1.9358}x31.317 + 0.2245x15^2x8.344\right)\right] = 2936\$.$

The cost savings is (4431-2936) 100/2936 = 51%.

It should be mentioned that the cost savings is high, since the fabrication cost is 1012x100/2936 = 34% of the total cost.

Conclusions

The table beam of a punch press is a special welded structure consisting of a box beam with an orthogonally stiffened upper flange. Design constraints on beam deflection, local deformation of stiffened upper flange and on the fatigue of welds should be fulfilled. The bending deformation of the grid of stiffeners can be neglected, since the shear deformations are governing. The local deflection and the shear stresses in the stiffeners can be determined by a simplified grid model using the force method.

The verification of the original structural version shows that the number of transverse stiffeners and the plate thicknesses can be decreased. The optimum thicknesses of stiffeners are determined by using minimum cost design procedures. The comparison of masses and costs of the original and the optimized versions shows significant savings.

Acknowledgements

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