

MINIMUM COST DESIGN OF A WELDED PUNCH PRESS FOR LIGHT INDUSTRY

K. Jármai*, J. Farkas*, L. Major**

* University of Miskolc, ** Schön Engineering Ltd, Budapest (Hungary)

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 meet the requirements of global and local stiffness. All welds should be checked for fatigue stress range. Verification of the original structural version shows that cost savings can be achieved by decreasing the plate thicknesses and the number of stiffeners. Calculation shows that the bending deflection of stiffeners can be neglected compared to the shear deflection. 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.

IIW Thesaurus keywords: economics; costs; computation; design, optimization; presses; reinforcement; structural members; plates; steel; CO2 welding; comparisons; reference lists

1 INTRODUCTION

Punch presses are used in light industry for punching of various non-ferrous materials such as leather, plastics, paper, textile, etc. The main problem for the supporting structure is to guarantee a very stiff plane surface, with very small deflections caused by high pressing load. 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 a pressing force $F = 1,600$ kN is

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

IIS/IIW-1517-00 (ex. doc. XV-1058-00) Class A, recommended for publication by IIW Commission XV "Fundamentals of design and fabrication for welding".

The maximum prescribed local deflection is

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

The pressing force is distributed over a square surface with sides of 400×400 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 $N_f = 10^7$.

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 320×236 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 $E = 2.1 \times 10^5$ MPa

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

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 as 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

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

The moment of inertia is

$$I_x = 1.8751 \times 10^{10} \text{ mm}^4 \quad (5)$$

The maximum deflection due to bending, with a span length of $L = 2,086$ mm, is

$$w_b = \frac{FL^3}{48E I_x} = 0.0768 \text{ mm} \quad (6)$$

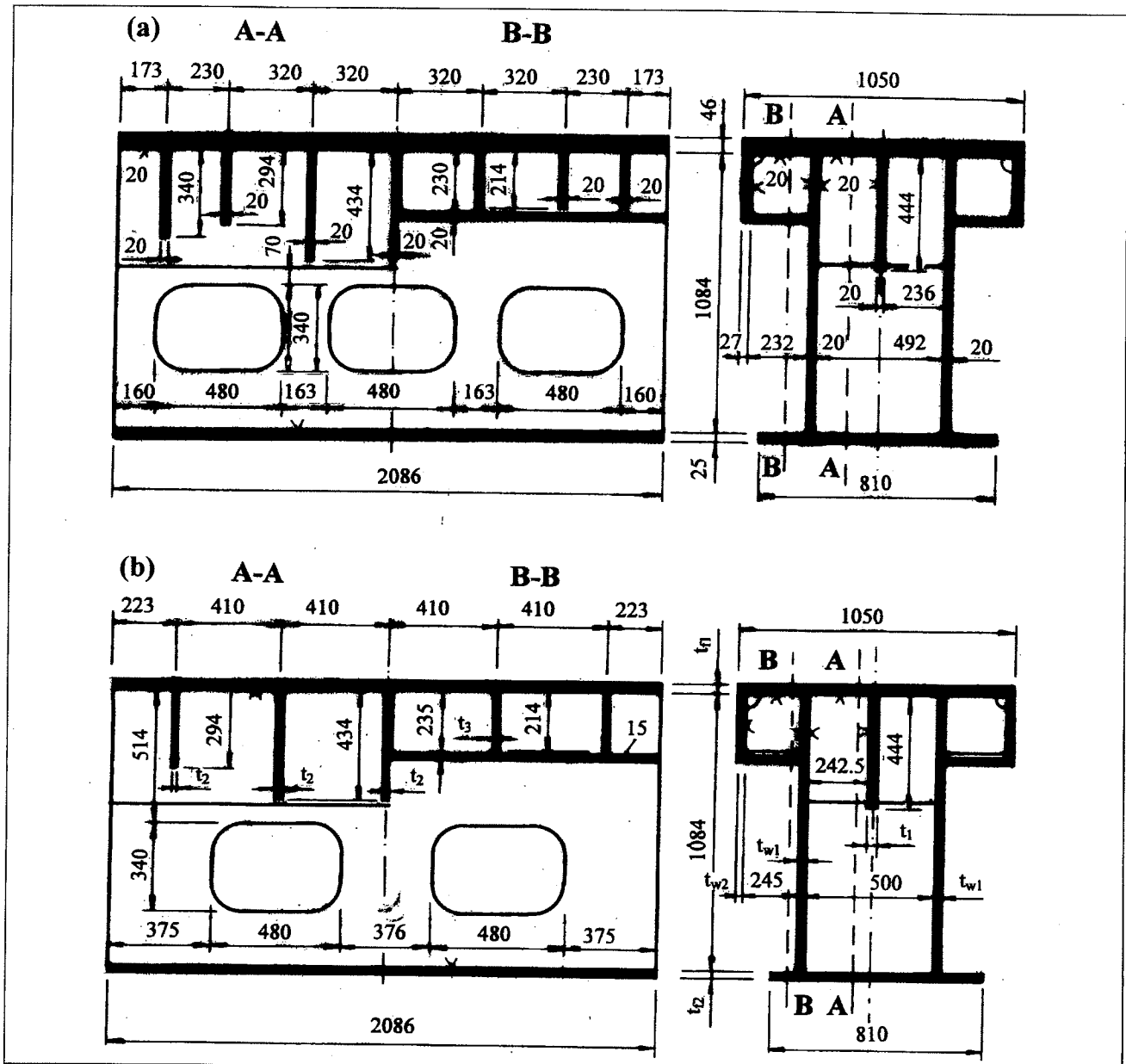


Fig. 1. Welded table beam of a punch press. (a) Original structure, (b) Optimized structure.

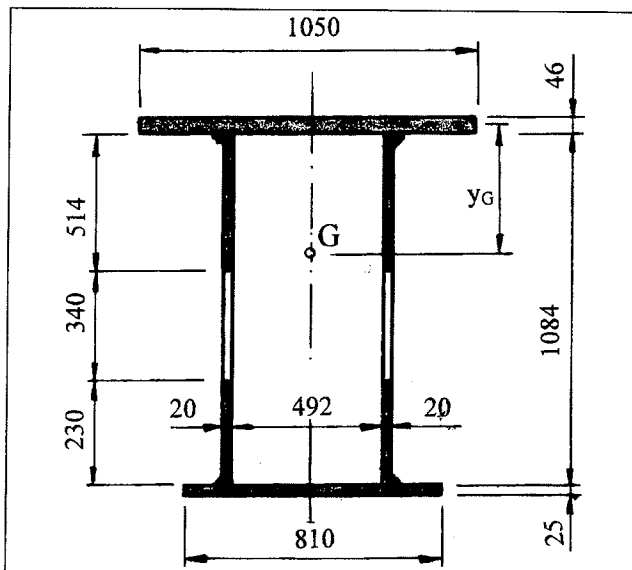


Fig. 2. Cross-section of the original box beam through holes.

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

$$w_s = \frac{800 \times 10^3 \times 323}{0.8077 \times 10^5 \times 40 \times 1084} + \frac{800 \times 10^3 \times 720}{0.8077 \times 10^5 \times 40 \times 744} = 0.3134 \text{ mm} \quad (7)$$

The total deflection is

$$w_{\max} = w_b + w_s = 0.3902 < 0.5 \text{ mm, OK} \quad (8)$$

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 $N_f = 2 \times 10^6$ are as follows:

$$\Delta\sigma_c = 100, \Delta\tau_c = 80 \text{ MPa} \quad (9)$$

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

The maximum bending moment is

$$M_{\max} = FL/4 = 8.344 \times 10^8 \text{ Nmm} \quad (10)$$

The maximum normal stress in longitudinal welds at the lower flange is

$$\sigma_{\max} = M(1,084 - y_G)/I_x = 31.1 \text{ MPa} \quad (11)$$

The average shear stress in webs is

$$\tau = \frac{800 \times 10^3}{40 \times 744} = 26.9 \text{ MPa} \quad (12)$$

According to EC3

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

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

$$10 < 33.7 \text{ MPa, OK.} \quad (14)$$

3.4 Local deformation of inner stiffeners in the upper flange

It has already been 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, let us

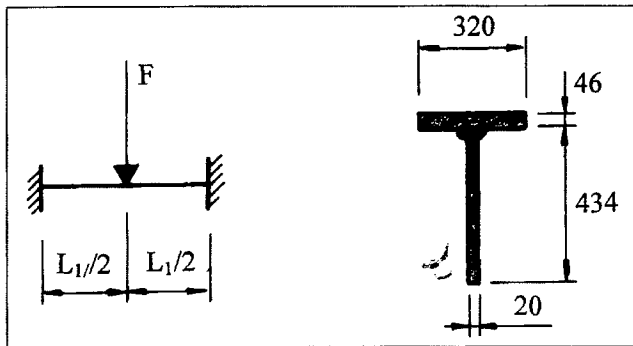


Fig. 3. A transverse stiffener.

consider first a stiffener as shown in Fig. 3. Defining the ratio of

$$\omega = \frac{320 \times 46}{434 \times 20} = 1.6958 \quad (15)$$

We obtain for the moment of inertia

$$I_x = \frac{434^3 \times 20}{12} \frac{1 + 4\omega}{1 + \omega} = 3.9336 \times 10^8 \text{ mm}^4 \quad (16)$$

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

$$w_b = \frac{FL_1^3}{192EI_x} = 0.0120 \text{ mm} \quad (17)$$

and the shear deflection is ($A_w = 8,680 \text{ mm}^2$)

$$w_s = \frac{FL_1}{4GA_w} = 0.2807 \text{ mm} \quad (18)$$

It can be seen that the bending deflection is only 4% of the shear deflection, 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:

$$w_A = \frac{(F - X_1)B}{4GA_1} - \frac{X_2b}{GA_1} = \frac{X_1L_1}{4GA_2} \quad (19)$$

$$w_B = \frac{(F - X_1)b}{2GA_1} - \frac{X_2b}{GA_1} = \frac{X_2L_1}{4GA_2} \quad (20)$$

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

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

$$X_1 = \frac{F}{3 + \frac{L_1A_1}{2bA_2}}; X_2 = \frac{F}{2} - X_1 \left(\frac{1}{2} + \frac{L_1A_1}{4bA_2} \right) \quad (21)$$

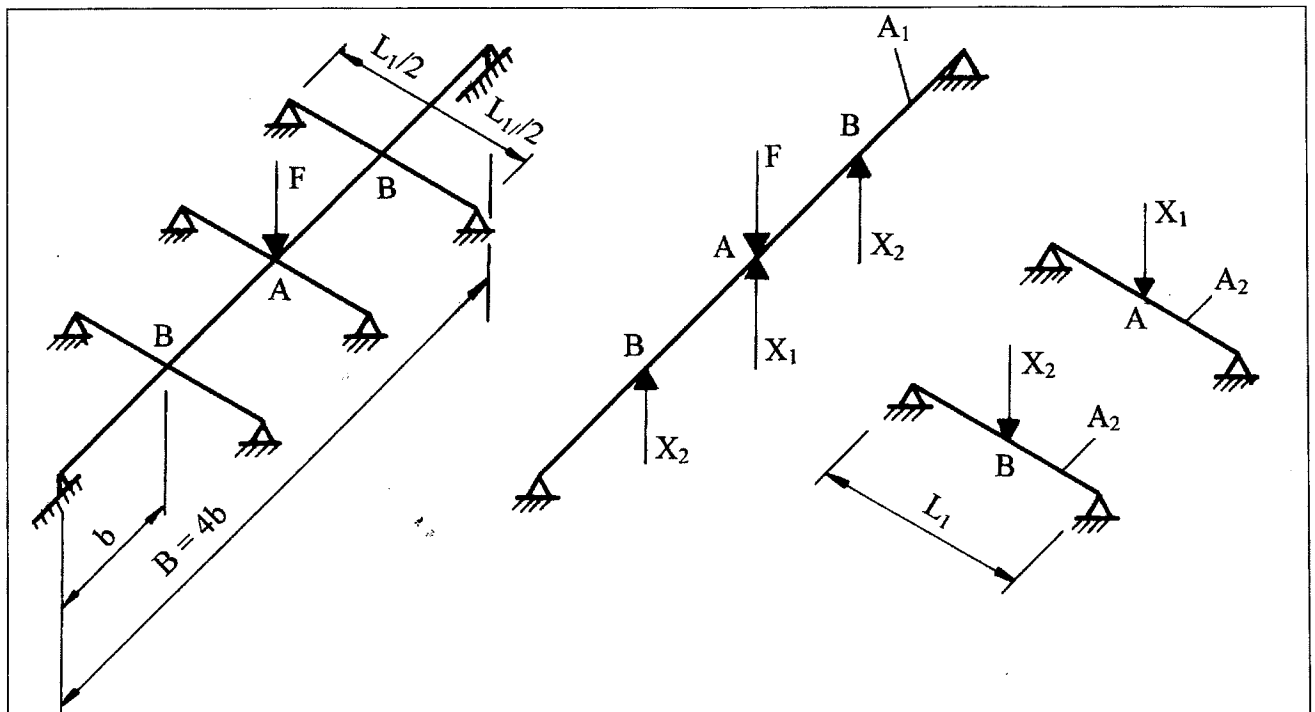


Fig. 4. Grid model of inner stiffeners.

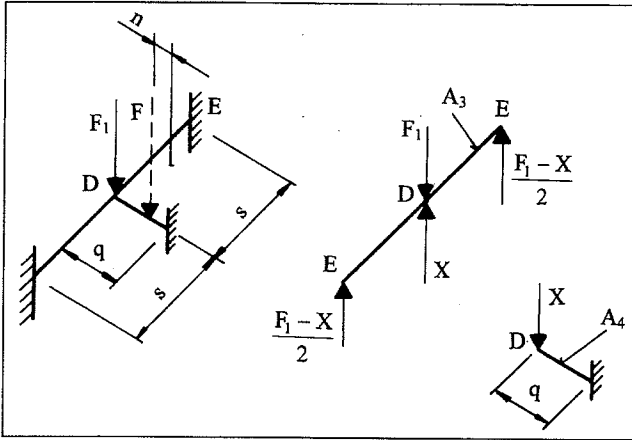


Fig. 5. Grid model of outer stiffeners.

In our case it is $F = 1,600$ kN, $L_1 = 492$, $b = 320$ mm, $A_1 = 444 \times 20$, $A_2 = 434 \times 20$, thus $X_1 = X_2 = 4.225 \times 10^5$ N. The maximum deflection is

$$w_A = 0.041 < 0.12 \text{ mm, OK.} \quad (22)$$

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

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

3.5 Outer stiffeners of the upper flange

The pressing load F is acting on a transverse stiffener at a distance of $n = q - 200 = 32$ mm. From this action a force of $F_1 = 32F/232 = 220.7 \times 10^3$ 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

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

The solution of (24) is

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

With $q = 232$, $s = 320$ mm, $A_3 = 6,318$, $A_4 = 4,280$ mm² we obtain $X = 70.3 \times 10^3$ N. According to (24) the maximum deflection is

$$w_D = 0.047 < 0.12 \text{ mm, OK.} \quad (26)$$

The shear stress in the transverse stiffener is

$$\tau = \frac{70.3 \times 10^3}{214 \times 20} = 16 < 43.0 \text{ MPa, OK.} \quad (27)$$

and that in the longitudinal stiffener is

$$\tau = \frac{(220.7 - 70.3)10^3}{27 \times 234} = 24.0 < 43.0 \text{ MPa, OK.} \quad (28)$$

3.6 Cost calculation

The cost is calculated disregarding the fabrication sequence. It should be noted that some costs, such as flattening and planing of plate parts and that of painting, are not considered here. It is assumed that the welding method for all the butt welds is GMAW-C (Gas Metal Arc Welding with CO₂). 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

$$C_2 a_w^n = 0.152 a_w^{1.9358} \quad (29a)$$

and for 1/2 V butt welds

$$C_2 a_w^2 = 0.2245 a_w^2 \quad (29b)$$

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	1,050 x 2,086 x 46	100.75 x 10 ⁶
lower flange plate	810 x 2,086 x 25	42.24
two webs	2 x 20 x 1,084 x 2,086	90.45
inner longitudinal stiffener	444 x 20 x 2,086	18.42
outer longitudinal stiffeners	2 x 250 x 27 x 2,086	28.16
outer horizontal plates	2 x 232 x 20 x 2,086	19.36
inner transverse stiffeners	2 x 472 x 20 x 1,295	24.45
outer transverse stiffeners	14 x 232 x 20 x 214	13.90

The total volume is $V = 337.73 \times 10^6$

The total mass is $(\rho = 7.85 \times 10^{-6} \text{ kg/mm}^3) \quad \rho V = 2,651 \text{ kg}$

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

The cost is as follows:

$$K = k_m \rho V + k_f \left(C_1 \Theta_d (\kappa \rho V)^{0.5} + 1.3 \sum_i C_{2i} a_{wi}^{1.9358} L_{wi} \right) = 2651 + 0.417 [3(37 \times 2651)^{0.5} + 1.3(0.152 \times 20^{1.9358} \times 28.838 + 0.152 \times 27^{1.9358} \times 4172 + 0.2245 \times 20^2 \times 8344)] = 4431\$ \quad (30)$$

4 OPTIMUM DESIGN OF A NEW STRUCTURAL VERSION

The verification of the original version shows that some dimensions can be reduced. Since the main table dimensions should remain, the number of upper transverse stiffeners and the thicknesses of plates can be reduced. 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 reduced from 7 to 5 and their distance is increased from 320 to 410 mm. Thus, a flange field has the side dimensions of 410 x 242.5 mm. The required plate thickness can be calculated according to [11]

$$t_{f1} = \left(\frac{0.026 \times 10 \times 242.5^4}{2.1 \times 10^5 \times 0.12} \right)^{1/3} = 32.9 \text{ mm} \quad (31)$$

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 reduced to 2, and we calculate the I_x neglecting the holes.

The distance of the center of gravity is

$$y_G = 419 \text{ mm} \quad (32)$$

The moment of inertia is

$$I_x = 1.8253 \times 10^{10} \text{ mm}^4 \quad (33)$$

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

$$w_b = 0.0789 \text{ mm} \quad (34)$$

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

$$w_s = 0.3845 \text{ mm} \quad (35)$$

The total deflection is

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

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

$$\sigma_{\max} = 31.2 \text{ MPa} \quad (37)$$

The shear stress, similarly to (12), is

$$\tau = 24.6 \text{ MPa} \quad (38)$$

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

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

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:

$$\frac{K}{k_m} = \rho(4bA_1 + 3L_1A_2) + \frac{k_f}{k_m} \left[3(7\rho V)^{0.5} + 1.3 \times 0.152(2.528t_1^{1.9358} + 6.663t_2^{1.9358}) \right] \quad (40)$$

where $A_1 = 444t_1$ and $A_2 = 434t_2$, $F = 1,600 \text{ kN}$, $L_1 = 500$, $b = 410 \text{ mm}$, $k_f/k_m = 0.417 \text{ kg/min}$.

Deflection constraint for the point A is expressed as

$$w_A = \frac{X_1 L_1}{4GA_2} \leq 0.12 \text{ mm} \quad (41)$$

The shear stress constraint is given by

$$\tau = \frac{X_1}{2A_2} \leq 43.0 \text{ MPa} \quad (42)$$

X_1 is calculated with (21).

In the minimum cost design, 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 $t_1 = t_2 = 15 \text{ mm}$. $X_1 = 441.5 \times 10^3 \text{ N}$

Check of the constraints:

$$w_A = 0.1050 < 0.12 \text{ mm, OK.} \quad (43)$$

$$\tau = 33.9 < 43.0 \text{ MPa, OK.} \quad (44)$$

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). $F_1 = 45F/245$, $q = 245$, $s = 410 \text{ mm}$, $A_3 = 250t_{w2}$, $A_4 = 214t_3$.

The cost function is expressed as

$$\frac{K}{k_m} = \rho V + \frac{k_f}{k_m} \left[3(12\rho V)^{0.5} + 1.3 \times 0.152(2 \times 2.086t_{w2}^{1.9358} + 10 \times 0.676t_3^{1.9358}) \right] \quad (45)$$

where $V = 2 \times 250 \times 2,086t_{w2} + 10 \times 245 \times 214t_3$.

The deflection constraint is defined by

$$w_D = \frac{Xq}{GA_4} \leq 0.12 \text{ mm} \quad (46)$$

The shear stress constraint is

$$\tau = \frac{X}{A_4} \leq 43.0 \text{ MPa} \quad (47)$$

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

Check of the constraints:

$$w_D = 0.1159 < 0.12 \text{ mm, OK.} \quad (48)$$

$$\tau = 38.2 < 43.0 \text{ MPa, OK.} \quad (49)$$

4.5 Cost calculation

The volume is as follows:

upper flange plate	1,050 x 2,086 x 36	78.8508 x 10 ⁶
lower flange plate	810 x 2,086 x 20	33.7932
two web	2.15 x 1,084 x 2,086	67.8367
inner longitudinal stiffener	444 x 15 x 2,086	13.8928
outer longitudinal stiffeners	2 x 250 x 15 x 2,086	15.6450
outer horizontal plates	2 x 245 x 15 x 2,086	15.3321
inner transverse stiffeners	2 x 242.5 x 15(2 x 294 + 3 x 434)	13.7498
outer transverse stiffeners	10 x 245 x 15 x 214	5.9578

The total volume is 245.0582 x 10⁶ mm³.

The total mass of the modified structure is $\rho V = 1,924 \text{ kg}$. This means that the mass saving achieved by optimization is $(2,651 - 1,924)/1,924 = 38\%$.

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

The cost is as follows:

$$K = 1,924 + 0.417 [3(29 \times 1,924)^{0.5} + 1.3(0.152 \times 15^{1.9358} \times 31.317 + 0.2245 \times 15^2 \times 8.344)] = 2,936 \text{ \$}.$$

The cost saving is $(4,431 - 2,936)/4,431 = 51\%$.

It should be mentioned that the cost saving is high, since the fabrication cost is $1,012 \times 100/2,936 = 34\%$ of the total cost. Note that the cost saving would be affected, if other cost elements (flattening, planing, painting) were considered.

5 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 reduced. 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

This work has been supported by grant OTKA 22846 from the Hungarian Fund for Scientific Research and the Fund for Higher Education grant 8/2000.

REFERENCES

1. Farkas, J., Structural synthesis of press frames having columns and cross beams of welded box cross-section. *Acta Techn. Hung.*, Vol. 83 (1974), pp.191-201.
2. Farkas, J., Optimum design of metal structures. Budapest, Akadémiai Kiadó, Chichester, Ellis Horwood, 1984.
3. Farkas, J., Jármai, K., Analysis and optimum design of metal structures. Balkema, Rotterdam-Brookfield, 1997.
4. Jármai, K., Farkas, J., Cost calculation and optimisation of welded steel structures. *J. Constructional Steel Research*, Vol. 50 (1999), pp. 115-135.
5. Farkas, J., Jármai, K., Minimum cost design and comparison of uniaxially compressed plates with welded flat, L- and trapezoidal stiffeners. *IIW Doc. XV-F-71-99 / XV-1024-99*.
6. Farkas, J., Jármai, K., Fabrication cost calculation and optimum design of welded steel silos. *Welding in the World*, Vol. 37 (1996) No. 5, pp. 225-232.
7. Farkas, J., Jármai, K., Minimum cost design of a bunker constructed from welded stiffened plates. *IIW-1516-00 (ex-doc. XV-1038-00)*, *Welding in the World*, Vol. 45 (2001), No. 1/2.
8. Farkas, J., Jármai, K., Optimum design of a stiffened conical roof considering the residual welding distortions. *IIW-Doc. XV-986-98 / XV-F-66-98 / X/XV-RSDP-28-98*.
9. Farkas, J., Jármai, K., Minimum cost design of SHS Vierendeel trusses. *Tubular Structures VII*. Balkema Rotterdam 1996. *Proc. 7th Int. Symposium on Tubular Structures*, Miskolc 1996. Eds. Farkas, J. and Jármai, K., pp. 463-468.
10. Jármai, K., Farkas, J., Horikawa, K., Economic design of steel bridge decks. *Welding in the World*, Vol. 41 (1998), No. 1, pp. 49-59.
11. Timoshenko, S., Woinowsky-Krieger, S., *Theory of plates and shells*. 2nd ed. New York, McGraw Hill, 1959.
12. Eurocode 3 Design of steel structures. Part 1.1: General rules and rules for buildings. European Committee for Standardization, Brussels, 1992.