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DOUBLE MAIN GIRDER DESIGN OF AN OVERHEAD TRAVELLING CRANE FOR MINIMUM COST

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Abstract: *A crane structure of two doubly symmetric welded box beams is designed for an overhead travelling crane for minimum cost. The following design constraints are considered: local buckling of web and flange plates, fatigue of the butt K weld under rail and fatigue of fil-let welds joining the transverse diaphragms to the box beams. The rails are placed over the inner webs of box beams. To increase the fatigue strength of the last mentioned welds, an efficient post welding treatment (PWT) is considered. For the formulation of constraints the relatively new standard for cranes EN 13001-3-1 [1] is used. The cost function consists of cost of material, assembly, welding and PWT. PWT is economic, since it is used only for diaphragms near the span centre of box beams, where the bending stresses are high. The optimization is performed by systematic search using a MathCAD program.*
Keywords: *crane girder, fatigue, post welding treatment, optimum design*

1. INTRODUCTION

The main girder of overhead travelling cranes can be designed as a single or double box beam. The rail can be placed in the middle of the upper flange or over the inner web of the box beams. In our case we designed a double box beam with rails over the inner webs (Fig. 1). The research of post-welding treatments (PWT) does not give any data for these welds. PWT can cause a significant increase of fatigue strength for welds joining the transverse diaphragms to the upper flange, so we use these data. Our research shows that PWT can result in significant cost savings using them in welds joining the transverse diaphragms to the box or I-beams (Jármai et al. [2]).

2. DATA OF THE TREATED CRANE

The British Standard for cranes BS 2573-1 [3] is valid at present also. This BS gives characteristic parameters for crane groups. We select a workshop crane with a dynamic factor of $\psi_d = 1.3$, the governing number of cycles is $N = 4\text{x}10^6$, the coefficient of spectrum is according to EN 13001-3-1 [1] $s_3 = 2$. The safety factor for fatigue is $\gamma_f = 1.25$.
Yield stress $f_y = 355$ MPa, according to EN 13001-3-1 the maximum design stress for plate thicknesses $t < 16$ mm is 323 MPa, for $16<t<40$ mm 314 MPa. We do not treat hybrid beams constructed with steels of two different yield stresses.
Span length is $L = 16.5$ m, hook load $P = 200$ kN, mass of the trolley $G_k = 42.25$ kN, distance of wheels $k = 1.9$ m, height of rail $h_s = 70$ mm, specific mass of the service-walkway and rail $p = 1900$ N/m, steel density $\rho = 7.85\text{x}10^{-6}$ kg/mm^3 or $\rho_0 = 7.85\text{x}10^{-5}$ N/mm^3, distance of transverse diaphragms $a = L/10 = 1650$ mm. The box beams are doubly symmetric.

2.1. BUCKLING CONSTRAINTS OF THE WEB UNDER THE RAIL

2.1.1. *Bending*

Stress from the vertical bending

$$\sigma_x = \frac{M_h}{W_x} \tag{1}$$

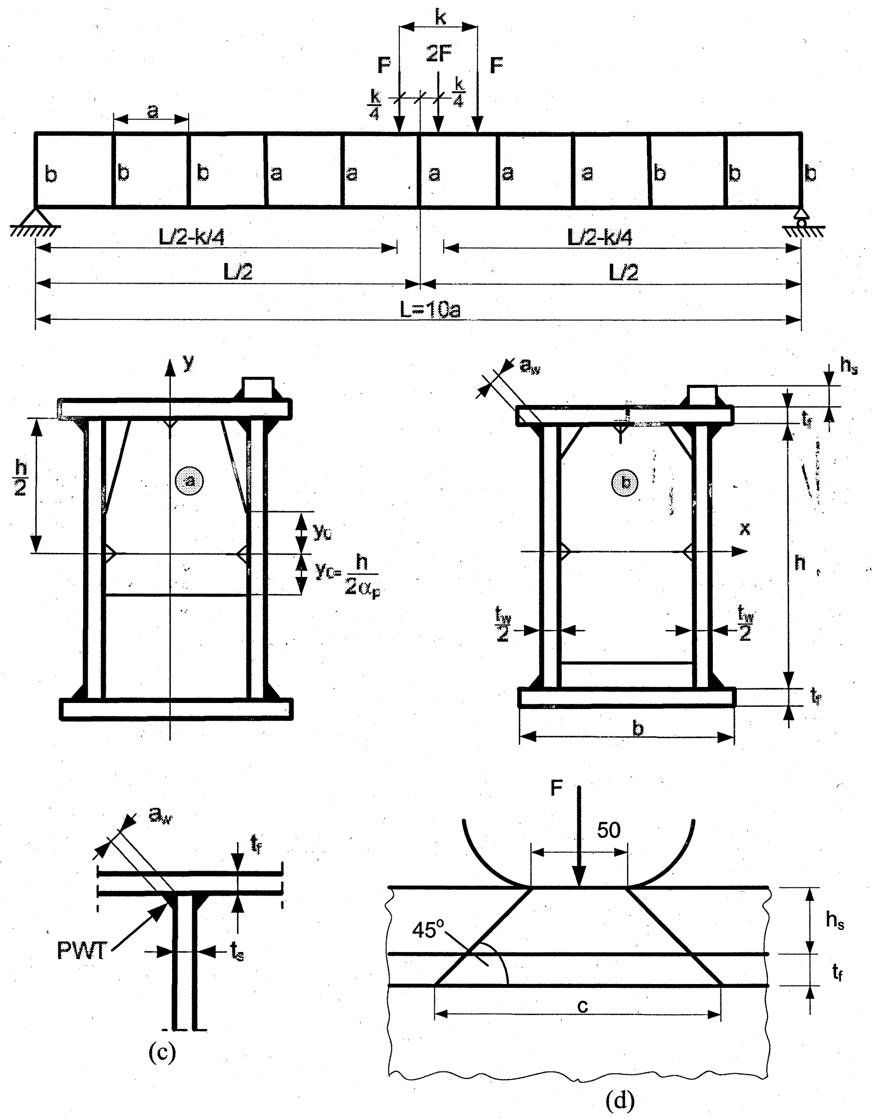

Figure 1: Data and cross-sections of the crane beams. Diaphragms (a) are used in the middle of beams for high bending stresses, PWT is used for the welds joining the diaphragms, diaphragms (b) are used near the beam ends, (c) shows the welds with PWT, (d) shows the load distribution in the beam web from the crane wheel.

Maximum bending moment in the case of the load position of two concentric forces

$$M_x = (1.05\rho_0 A + p)\frac{L^2}{8} + \frac{F}{2L}\left(L - \frac{k}{2}\right)^2, \quad F = \frac{\psi_d P + G_k}{4} \tag{2}$$

$$A = ht_{w0} + 2bt_{f0} \tag{3}$$

$$W_x = \frac{h^2 t_{w0}}{6} + bht_{f0} \tag{4}$$

t_{w0} and t_{f0} are the rounded plate thicknesses.

Bending moment from the horizontal bending

$$M_y = 0.3x0.5\left[(1.05\rho_0 A + p) + \frac{G_k}{8L}\left(L - \frac{k}{2}\right)^2\right] \quad (5)$$

The multiplier 0.5 expresses that two wheels are driven from four, 0.3 is the coefficient of mass force.

$$\sigma_y = \frac{M_y}{W_y}, \; W_y = \frac{b^2 t_{f0}}{3} + \frac{h t_{w0} b}{2} \quad (6)$$

It is not necessary to calculate with effective width, when

$$\sigma_x \le k_x f_y, k_x = 1 \quad (7)$$

$$\lambda_x = \sqrt{\frac{f_y}{k_{\sigma x}\sigma_e}} \le 0.673, \; k_{\sigma x} = 7.81 - 6.29\psi_x + 9.78\psi_x^2, \; \psi_x = -\frac{\sigma_x - \sigma_y}{\sigma_x + \sigma_y} \quad (8)$$

$$\sigma_e = \frac{\pi^2 E}{12(1-\nu^2)}\left(\frac{2t_{w0}}{h}\right)^2, E = 2.1x10^5 \text{ MPa}, \nu = 0.3 \quad (9)$$

The required plate thickness

$$t_{w.req} = \frac{2h}{0.673x28.42\varepsilon\sqrt{k_{\sigma x}}}, \; \varepsilon = \sqrt{\frac{235}{f_y}} \quad (10)$$

2.1.2. *Shear and torsion*

From shear (approximately)

$$\tau_{ny} = \frac{V}{h t_{w0}}, \; V = (1.05\rho_0 A + p) + \frac{F}{2L}\left(L - \frac{k}{2}\right) \quad (11)$$

From torsion

$$\tau_t = \frac{2M_t}{2bht_{w0}}, \; M_t = \frac{F}{2L}\left(L - \frac{k}{2}\right)\frac{b}{2} + \frac{pLb}{4} \quad (12)$$

The constraint on shear buckling

$$\text{if} \quad \tau = \tau_V + \tau_t \le k_{\tau 0} f_y / \sqrt{3}, k_{\tau 0} = 1 \quad (13)$$

$$\lambda_\tau = \sqrt{\frac{f_y}{k_\tau \sigma_e \sqrt{3}}} \le 0.84, \; k_\tau = 5.34 + \frac{4}{\alpha^2}, \quad \alpha = \frac{a}{h} = \frac{L}{10h} \quad (14)$$

$$\text{i.e. } t_{w.req} = \frac{2h}{31\varepsilon\sqrt{k_\tau}} \quad (15)$$

2.1.3. *Compression from a wheel*

According to Figure 1d

$$\sigma_{y1} = \frac{2F}{c t_{w0}}, \; c = 50 + 2(h_s + t_{f0}) = 50 + 2x100 = 250 \text{ mm} \quad (16)$$

$$\text{If} \quad \sigma_{y1} \le k_y f_y, k_y = 1 \quad (17)$$

$$\lambda_y = \sqrt{\frac{f_y}{k_{\sigma y}\sigma_e \frac{a}{c}}} \le 0.831 \quad (18)$$

From the diagram of EN13001-3-1 [1] *c/a* = 250/1650 = 0.15 and α = *a/h* = 1650/620 = 2.7 $k_{\sigma y} = 1$

$$t_{w.req} = \frac{2h}{60.97\varepsilon} \quad (19)$$

The complex check

$$\left(\frac{|\sigma_x|}{f_{bx}}\right)^{e_1} + \left(\frac{|\sigma_y|}{f_{by}}\right)^{e_2} - V_0\left(\frac{|\sigma_x \sigma_y|}{f_{bx} f_{by}}\right) + \left(\frac{\tau}{f_{b\tau}}\right)^{e_3} \le 1, \; e_1 = 1 + k_x^4, e_2 = 1 + k_y^4, e_3 = 1 + k_x k_y k_{\tau 0}^2 \quad (20)$$

$$V_0 = (k_x k_y)^6 \text{ if } \sigma_x \sigma_y \ge 0, \; V_0 = -1 \text{ if } \sigma_x \sigma_y \le 0 \quad (21)$$

In our case $k_x = k_y = k_{\tau 0} = 1$ (22)

$$\sigma_{red} = \sqrt{(\sigma_x + \sigma_y)^2 + \sigma_{y1}^2 - (\sigma_x + \sigma_y)\sigma_{y1} + 3\tau^2} \le f_y \tag{23}$$

3. BUCKLING CONSTRAINTS OF THE UPPER FLANGE

3.1. Vertical and horizontal bending

Similarly to the constraint on web buckling

$$t_{f.req} = \frac{b}{0.673x28.42\varepsilon\sqrt{k_{\sigma y}}},\ k_{\sigma y} = \frac{8.2}{1.05 + \psi_y},\ \psi_y = \frac{\sigma_x - \sigma_y}{\sigma_x + \sigma_y} \tag{24}$$

3.2. Torsion

Similarly to the web

$$t_{f.req} = \frac{b}{31\varepsilon\sqrt{k_{\tau b}}},\ k_{\tau b} = 5.34 + \frac{4}{\alpha_b^2},\ \alpha_b = \frac{a}{b} \tag{25}$$

4. FATIGUE CONSTRAINT FOR THE WELD UNDER THE RAIL

According to the EN 13001-3-1 [1] the fatigue strength of a K butt weld for the number of cycles $N = 4x10^6$ is $\Delta\sigma_C = 112$ MPa, the allowed stress for the spectrum factor $s_3 = 2$

$$\Delta\sigma_{Rd} = \frac{\Delta\sigma_C}{\gamma_f \sqrt[3]{s_3}} = 71.1 \text{ MPa} \tag{26}$$

and for shear

$$\Delta\tau_{Rd} = \frac{\Delta\tau_C}{\gamma_f \sqrt[3]{s_3}} = 50.8 \text{ MPa} \tag{27}$$

The complex constraint on fatigue is expressed as

$$\eta = \left(\frac{\sigma_x + \sigma_y}{\Delta\sigma_{Rd}}\right)^3 + \left(\frac{\sigma_{y1}}{\Delta\sigma_{Rd}}\right)^3 + \left(\frac{\tau_V + \tau_t}{\Delta\tau_{Rd}}\right)^5 \le 1 \tag{28}$$

5. FATIGUE CONSTRAINT FOR FILLET WELDS JOINING THE TRANSVERSE DIAPHRAGMS

The fatigue strength [4]

$$\Delta\sigma_C = \alpha_P 63 \text{ MPa} \tag{29}$$

α_P is the coefficient of the effect of PWT, for ultrasonic treatment 1.3, for HiFIT high frequency impact treatment 1.6.
The allowed stress

$$\Delta\sigma_{f.adm2} = \frac{\Delta\sigma_C}{\gamma_f \sqrt[3]{2}} \tag{30}$$

The constraint is given by

$$\sigma_x \le \Delta\sigma_{f.adm2} \tag{31}$$

6. THE COST FUNCTION

The cost function is formulated according to the fabrication sequence (Farkas & Jármai books [5,6,7,8]).

(1) Welding of the upper flange, webs and transverse diaphragms, PWT of the welds joining the diaphragms. Two forms of diaphragms are used: the 5 diaphragms near the span centre are cut according to the Figure 1a, the other 6 diaphragms are constructed according to Figure 1b.
The structural volume for this fabrication phase is

$$V_1 = L(ht_{w0} + bt_{f0}) + 6bht_s + 2.5bht_s\left(1 + \frac{1}{\alpha_P}\right),\ t_s = 6\text{ mm},\ \alpha_P = 1.6 \quad (32)$$

The number of the assembled structural elements is $\kappa_1 = 14$, the factor of the complexity of assembly is $\Theta_1 = 3$. The welding cost consists of four parts: GMAW-C welding of Butt K welds under the rail (K_{w11}), GMAW-C welding of the fillet welds joining the other web, welding of the diaphragms (K_{w12}) and PWT of the welds of 5 diaphragms (K_t).

$$K_{w1} = k_w(\Theta_1\sqrt{\kappa_1\rho V_1} + 1.3x0.3394x10^{-3}a_w^2L + K_{w11}),\ k_w = 1.0\text{ \$/min} \quad (33)$$

$$K_{w11} = k_w 1.3x0.1520x10^{-3}a_{w1}^{1.94}L;\quad a_{w1} = t_{w0}/2 \quad (34)$$

$$K_{w12} = k_w 1.3x0.7889x10^{-3}a_w^2L_w,\ a_w = t_{w0}/4,\ L_w = 2\left[6(b+2h) + 5\left(b + \frac{h}{\alpha_P}\right)\right] \quad (35)$$

$$K_t = k_w L_t T_0,\ L_t = 10b,\ T_0 = 0.0033\text{ min/mm} \quad (36)$$

(2) Welding of the lower flange with two GMAW-C fillet welds

$$K_{w2} = k_w\left(\Theta_2\sqrt{\kappa_2\rho V_2} + 1.3x0.3394x10^{-3}a_w^2 2L\right),$$
$$\Theta_2 = 2,\ V_2 = V_1 + bt_{f0}L,\ \kappa_2 = 2 \quad (37)$$

Welding of the two webs from 11x1500 mm parts with GMAW-C butt K-welds

$$K_{w3} = k_w\left(\Theta_2\sqrt{11\rho V_3} + 1.3x0.152x10^{-3}x10h\left(\frac{t_{w0}}{2}\right)^{1.94}\right),\ V_3 = Lht_{w0}/2 \quad (38)$$

Welding of the two flanges from 11x1500 mm parts with GMAW-C butt K-welds

$$K_{w4} = k_w\left(\Theta_2\sqrt{11\rho V_4} + 1.3x0.152x10^{-3}x10bt_{f0}^{1.94}\right),\ V_4 = Lbt_{f0} \quad (39)$$

Material cost

$$K_m = k_m\rho V_2,\ k_m = 1.0\text{ \$/kg} \quad (40)$$

Total cost

$$K = K_m + K_{w1} + K_{w11} + K_{w12} + K_t + K_{w2} + 2K_{w3} + 2K_{w4} \quad (41)$$

7. RESULTS OF OPTIMIZATION

The results are given in Table 1.

Table 1: Dimensions and deflection in mm, stresses in MPa, volume in mm^3, costs in \$. Minima are marked by bold letters.

h	710	660	620	600
b	340	380	420	440
t_{w0}	30	28	26	26
t_{f0}	40	40	40	40
σ_x	61.95	62.6	62.7	62.8
Equation (19)	26.9	25.0	23.5	22.7
Equation (10)	20.0	18.4	17.2	16.6
w_{max}	9.3	10.0	10.5	10.7
Equation (28)	0.978	0.995	0.992	0.983
V_2x10^{-8}	**8.153**	8.222	8.367	8.547
K_t	11.2	12.5	13.9	14.5
K	14230	13890	**13690**	13930

8. CONCLUSIONS

The optimization has been performed by using a MathCAD program. Since the welding cost depends on the web thickness, the cost can be decreased by decrease of web thickness or web height. This decrease is stopped by the increase of cost caused by the increase of flange width. The web thickness is determined by the constraint on the maximal stress from the wheel load. In the systematic search we select a b and for this value h is searched, which fulfils the constraints. The web thickness is determined by the quality of the weld under the rail. Therefore, it is necessary to use high quality butt K weld. The governing constraints are the constraint on the compressive stress under rail and those on the fatigue. η should be smaller than 1 and σ_x should be smaller than $\Delta\sigma_{f.adm2} = 64.0$. The constraint of Equation 15 is passive.

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