OPTIMUM DESIGN OF WELDED ALUMINIUM STRUCTURAL PARTS WITH FATIGUE CONSTRAINTS

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A fáradási feltételeknek jelentős hatása van az alumínium szerkezeti elemek optimális méreteire. Az elemek helyi horpadása a határfeszültség helyett a tényleges statikus feszültség figyelembevételével számítható, ha a fáradási feltétel aktív. A statikus és a dinamikus lengő terhelés arányától függően a fáradási, a kihajlási, vagy a lehajlási feltétel lehet aktív. Négy illusztratív számpélda került kidolgozásra. Ezek a következők: csomólemezhez kapcsolódó nyomott rúd, nyomott rúd középen hegesztett homloklemezes illesztéssel, I-szelvényű konzol sarokvarrattal oszlophoz rögzítve, szekrényszelvény belső diafragmákkal.

A BS 8118 és az IIW új fáradási ajánlásait alkalmaztuk.

ABSTRACT

The fatigue constraints significantly affect the optimum dimensions of aluminium structural parts. The local buckling of elements may be calculated considering the actual static stress level instead of limiting stress when the fatigue constraint is active. Depending on the ratio of permanent static and fluctuating variable load the fatigue constraint or other constraints on overall buckling as well as deflection can be active. Four illustrative numerical examples are worked out as follows: a compressed strut connected to gusset plates, a compressed strut with a welded splice, a cantilever I-beam with fillet welded connection to a column and a box-beam with welded inside diaphragms. The design rules of BS 8118 and the new IIW fatigue recommendations are considered.

KEYWORDS

Welded aluminium structures, optimum design, fatigue of welded joints, structural hollow sections, overall and local stability, thin-walled structures

1. INTRODUCTION

The aim of the present study is to show the optimum design procedure of welded aluminium structural components subject to fluctuating forces in addition to permanent static loads. In the optimum design structural versions are sought which minimize the objective function and fulfil the design constraints. As *objective function* the weight is selected here, since fabrication cost data for aluminium weldments are in this moment not available. *Design constraints on fatigue, stability and deflection* are formulated according to British Standard for aluminium structures BS 8118 (BS) [1]. The new IIW recommendations on fatigue of welded components (Recommendations) (1995) [2] are also considered.

The problem is that it cannot be known in advance, which constraints are active, therefore all constraints should be formulated and investigated for an actual case. Special problem is the interaction of fatigue and local stability constraints. Local stability constraints are formulated by means of the limiting plate slenderness which depends on the maximum static stress. In static design, when the deflection constraint is passive, this maximum stress is calculated from limiting stress for an aluminium alloy. When the fatigue constraint is active, the design stress level may be much lower than this limiting stress, thus we can calculate with this actual stress level.

In structural components containing welded joints the fatigue constraints can be governing as it will be shown in numerical examples. In the case of *aluminium* welded structures some differences emerge as compared to the design of welded steel structures as follows.

1) There are several *aluminium alloys*, thus, the material selection is an important task;
2) There are *different types of profiles* depending on the fabrication procedure: extrusions, drawn tubes, profiles welded from plates, open profiles with bulbs, etc.
3) The *welded profiles have different strength* as compared to unwelded ones;
4) In static design the *softening effect of the heat affected zone* (HAZ) should be considered;
5) Some *characteristic data are different* as compared to steels: the elastic modulus and the material density is the third of the data for steels, the thermal expansion coefficient is double of steels. The material costs are also different.

In the present paper 4 numerical examples are selected to illustrate these specialities, mainly the

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fatigue design of welded aluminium joints. It should be noted that similar studies have been worked out by the authors for welded steel components [3, 4].

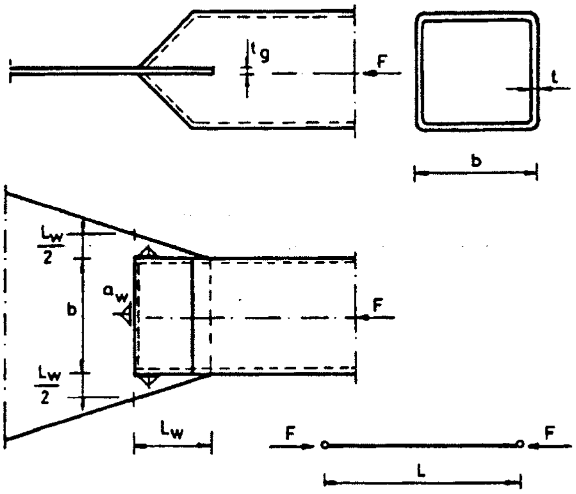

Fig. 1. A SHS strut connected to a gusset plate,

2. ILLUSTRATIVE NUMERICAL EXAMPLES

2.1 Optimum design of a square hollow section (SHS) compressed strut and its connection to a gusset plate (Fig. 1.)

The strut is connected to a gusset plate by means of a groove and fillet welds on both sides. The strut of length L = 5 m is cyclically loaded in tension-compression by a pulsating force F = 50 kN, the number of cycles is N = 5*105.

In general case a permanent load FG and a variable load FQ act, for which the safety factors, according to BS, are γ_G=1.20, γ_Q=1,33.

The objective function is the cross-sectional area, considering the effect of corner roundings approximately by a factor of 0.9

$$A = 0.9^* 4bt \qquad (1)$$

The fatigue constraint is expressed by

$$2F / A \le \Delta\sigma_N / \gamma_{mf} \qquad (2)$$

where $\Delta\sigma_N$ is the fatigue stress range corresponding to the number of cycles N, γ_{mf} is the fatigue material factor. Since this factor is not given in BS, we take it according to the Eurocode 3 (EC3) [5], which prescribes for important (non fail-safe) and accessible joints a factor of 1.24. The fatigue stress range $\Delta\sigma_C$ corresponding to N=2*10^6 is given in BS or in Recommendations in tables for various welded connections. When N≤5*10^6, then the following formula can be used

$$\log\Delta\sigma_N = \frac{1}{m}\log\frac{2^*10^6}{N} + \log\Delta\sigma_C \qquad (3)$$

In our example, according to BS, for the edge point of the gusset plate it is $\Delta\delta_C$=17 MPa and m = 3, thus, with Eq. (3) one obtains $\Delta\delta_N$=27 MPa. It should be noted that, in Recommendations the fatigue stress range for such aluminium joint is not given.

The overall buckling constraint is, according to BS (the permanent load is taken as F_G = 0)

$$\gamma_Q F / A \le \chi p_{01}; \quad p_{01} = p_0 / \gamma_m \qquad (4)$$

Note that the overall buckling formulae given in BS are the same as those in EC3, thus, we use the EC3 formulae as follows:

$$1/\chi = \phi + \sqrt{\phi^2 - \bar{\lambda}^2};\ \phi = 0.5\left[1 + \alpha\left(\bar{\lambda} - 0.2\right) + \bar{\lambda}^2\right] \qquad (5)$$

where for unwelded profiles α=0,2, for welded ones α=0,4

$$\bar{\lambda} = \lambda / \lambda_E;\ \lambda = KL / r;\ \lambda_E = \pi\sqrt{E / p_{01}} \qquad (6)$$

K is the effective length factor, for pinned ends K = 1, $r = \sqrt{I_x / A}$ is the radius of gyration, E=7*10^4 MPa is elastic modulus for aluminium alloys, p_0 is the limiting stress, γ_m=1,2 is the material factor. We select the aluminium alloy 6061-T6 and extruded SHS profiles for which p_0 = 240 Mpa and λ_E=58,77.

The local buckling constraint for unwelded profiles is defined by (p_{01} in MPa)

$$\delta_S = b / t \le \delta_{SL} = 18\sqrt{250 / p_{01}} \qquad (7)$$

Since the actual maximum static stress can be lower than p_{01} we can calculate with $\delta_{max} = \gamma_Q F/A$ instead of p_{01}.

Depending on the ratio of F_G / F_Q the fatigue or the overall buckling constraint may be active. Since the active constraint is not known in advance, we assume that in our example the fatigue constraint is active, thus, we perform the design for fatigue and then check the result for overall buckling.

The required cross-sectional area from Eq. (2) is $A_{req}=2F\gamma_{mf}/\Delta\sigma_N$=4630 mm². The actual static stress is $\sigma_{max}=\gamma_Q F/A_{req}$=14,4 MPa and the limiting *b/t* is δ_{SL}=75. We select the SHS profile 260*5 mm with A = 4680 mm² and *b/t* = 52, thus, this profile fulfils the fatigue and local buckling constraint. Check for overall buckling:

$$r = b / \sqrt{6} = 106;\ \bar{\lambda} = 5000 / (58.77 * 106) = 0.8026;$$

χ=0,8007 and Eq. (4) is in our case 14.2<160 Mpa, OK.

Calculation of the overlapping length for the fillet welded connection according to Fig. 1., neglecting the transverse fillet welds. For shear it is $\Delta\tau_C$ = 14 MPa and for the given N using Eq. (3) $\Delta\tau_N$ = 22,2 MPa. The fatigue constraint for the welds is

$$2F/(4L_w a_w) \le \Delta\tau_N/\gamma_{mf} = 22.2/1.25 = 17.76 \quad \text{MPa} \tag{8}$$

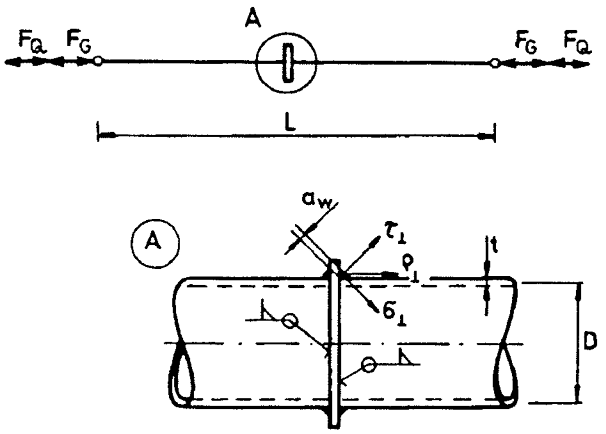

Fig. 2. CHS compressed strut with a welded splice

Taking the size of the fillet welds a_W=5 mm, from Eq. (8) one obtains L_W = 281, rounded 290 mm.

Calculation of the required gusset plate thickness. For the gusset plate welded on both sides it is $\Delta\sigma_C$=20, $\Delta\sigma_N$=31,7 MPa. Assuming a gusset plate width of L_W+b (Fig. 1.), from

$$\frac{2F}{(L_W+b)t_g} \le \frac{Ds_N}{g_{mf}} = \frac{31.7}{1.25} = 25.36MPa \tag{9}$$

we get t_g = 7,3 rounded 8 mm.

2.2 Compressed rod of circular hollow section (CHS) with a welded splice (Fig. 2.)

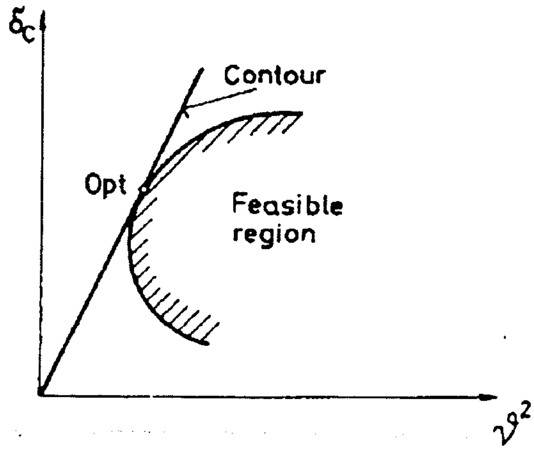

Fig. 3. Grapho-analytical optimization method

A CHS strut is centrally compressed by a permanent force -F_G (minus denotes compression) and a variable load F_Q pulsating between +F_Q and -F_Q. F_Q contains also a dynamic factor. The strut is constructed with a splice fillet welded end-to-end with an intermediate transverse plate. In the optimization of the strut section the unknown dimensions D and t are sought to minimize the strut cross-sectional area.

In the case of two unknowns *the grapho-analytical optimization method* can be advantageously applied. It is based on the theorem that, in the coordinate-system of the two unknowns, one of contours of the objective function touches the feasible region in the optimum point (Fig. 3.) This theorem can be derived by means of the method of Lagrange-multipliers [6].

In the optimum design of a compressed strut we use the unknowns ϑ=100D/L and δ_C=D/t instead of D and t since the objective function can be expressed by

$$A = \pi D t = \frac{\pi D^2}{\delta_C} = \frac{\pi L^2 \vartheta^2}{10^4 \delta_C} \tag{10}$$

and the contours of the objective function are defined by

$$\delta_C = \text{const} * \vartheta^2 \tag{11}$$

which are straight lines in the coordinate-system δ_C-ϑ^2, so it is easy to find the touching point.

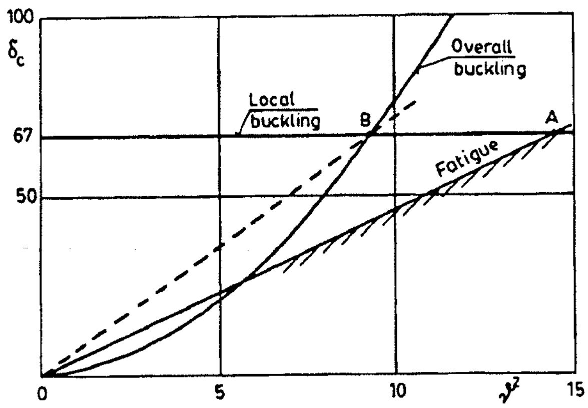

Fig. 4. Grapho-analytical optimization of the strut shown in Fig. 2.

The fatigue constraint is expressed by

$$2F_Q/A \le \Delta\sigma_N/\gamma_{mf} \quad \text{or} \quad \delta_C \le \frac{L^2\pi\Delta\sigma_N}{2*10^4 F_Q \gamma_{mf}} \tag{12}$$

F_Q = 12 kN, L = 5 m. According to BS the fatigue stress range at N = 2*10⁶ is $\Delta\sigma_C$ = 20 MPa for splice fillet weld toe crack (Recommendations give 22 MPa). γ_{mf} = 1,25. For a given N = $3*10^6$, using Eq. (3). one obtains $\Delta\sigma_C$ = 17,5 MPa, thus Eq. (12) takes the form

$$\delta_C \le 4.58159\vartheta^2 \tag{13}$$

This gives a straight line in our coordinate-system (Fig. 4.).

The local buckling constraint, according to BS for compact sections, is

$$D/t \le \delta_{CL} = \left(\frac{22}{3}\right)^2 \varepsilon^2 = 53.78 * 250/200 = 67 \qquad (14)$$

Note that we could calculate with a maximum static stress instead of p_{01}, but a value larger than 67 is not realistic.

The static overall buckling constraint is expressed by

$$\left(\gamma_G F_G + \gamma_Q F_Q\right)/A \le \lambda p_{01} \qquad (15)$$

χ is given by Eqs (5) and (6), F_G = 50 kN. Eq. (15) takes the form

$$\delta_C \le \frac{20.68}{\phi + \sqrt{\phi^2 - \bar{\lambda}}} \vartheta^2 \qquad (16)$$

where with $r = D/\sqrt{8}$, $\lambda_E = 58.77$, with K = 1 for pinned ends

$$c_0 = 100K/\lambda_E = 1.7014 \quad \text{and} \quad \bar{\lambda} = c_0\sqrt{8}/\vartheta = 4.8124/\vartheta \qquad (17)$$

Fig. 4. shows the limiting lines of constraints in the coordinate-system of the two unknowns. These lines define the feasible region and the optimum point at the intersection of the constraint on fatigue and local buckling (point A), since the contour touches the feasible region in this point. This means that the overall buckling constraint is in this case passive. The result is

$$\vartheta^2_{opt} = 14.62,\ \vartheta_{opt} = 3.824,\ D_{opt} = 191.2,\ t = 2.85 \text{ mm}.$$

We take a CHS profile 200*3 mm (A = 1885 mm²)

The point B gives optimum when the fatigue constraint is not considered. The solution of Eq. (16) with δ_{CL} = 67 is ϑ^2 = 9,3, ϑ = 3,05, D = 152, t = 2,3 mm. If the value of F_Q is kept constant, it is easy to find F_Q corresponding to point A, this means, when all three constraints are active (ϑ = 3,824). From Eq. (16) F_Q = 129 kN.

Finally, *the required fillet weld size* can be calculated from the fatigue constraint for root cracking. According to BS $\Delta\rho_{\perp C}$ = 14 MPa (Recommendations give 16 MPa).

With Eq.(3), for the given N one obtains $\Delta\rho_{\perp N=}$ =12,2 MPa. The fatigue constraint is

$$\Delta\rho_\perp = 2F_Q/(\pi D a_w) \le \Delta\rho_\perp/\gamma_{mf} = 12{,}2/1{,}25 = 9{,}8 \quad \text{MPa} \qquad (18)$$

From Eq.(18) a_w = 3,89 rounded 4 mm.

The constraint on the static fillet weld resistance according to BS is

$$\sqrt{\sigma_\perp^2 + 3\left(\tau_\perp^2 + \tau_{//}^2\right)} \le 0{,}85 p_w/\gamma_m \qquad (19)$$

where p_W is the limiting stress for weld metal, for alloy 6061-T6 it is 190 MPa.

Since $\sigma_\perp = \tau_\perp = \rho_\perp/\sqrt{2}, \quad \tau_{//} = 0,$

Eq.(19) can be written as

$$\sqrt{2}\rho_\perp = \sqrt{2}\frac{\gamma_G F_G + \gamma_Q F_Q}{\pi D a_w} = \frac{\sqrt{2} * 75{,}96 * 10^3}{200\pi a_w} \le \frac{0{,}85 * 190}{1{,}2} = 134{,}6 \quad \text{MPa} \qquad (20)$$

from which a_w = 1,3 mm this constraint is not active.

2.3. Optimum design of a welded I-section cantilever connected to a column by fillet welds (Fig.5)

For the calculation of the optimal dimensions of the welded I-section cantilever the constraint on fatique is used. The objective function is the cross-sectional area

$$A = ht_w + 2bt_f \qquad (21)$$

The fatigue constraint for the parent material at the toes of fillet welds connecting the flanges, in the case of a force F fluctuating between $+F$ and $-F$, is defined by

$$\Delta\sigma = 2FL/W_x \le \Delta\sigma_N/\gamma_{mf} \qquad (22)$$

The moment of inertia is expressed by

$$I_x \cong h^3 t_w/12 + 2bt_f(h/2)^2 \qquad (23)$$

Eq.(22) can be written in the from

$$W_x \cong I_x/(h/2) = h^2 t_w/6 + bt_f h \ge W_0 = 2FL(\Delta\sigma_N/\gamma_f) \qquad (24)$$

where $\Delta\sigma_N$ is the fatigue stress range, γ_{mf} =1,25 is the safety factor. W_0 is the required section modulus. From Eq.(21) one obtains

$$bt_f = A/2 - ht_w/2 \qquad (25)$$

Substituting Eq.(25) into (24) we get

$$A \ge 2W_0/h + 2ht_w/3 \qquad (26)$$

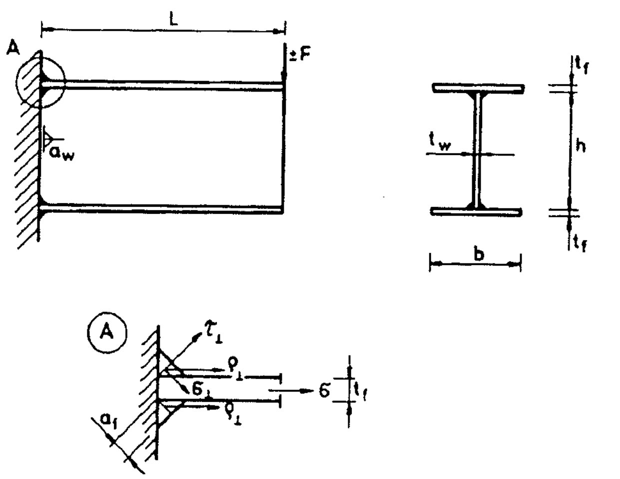

Fig.5. Welded I-section cantilever and the stress components in the connecting fillet welds

The local buckling constraint can be expressed by means of the limiting plate slenderness which can be derived from the basic formula for the critical local plate buckling stress. This stress should be larger than the σ_{max} static design stress

$$\sigma_{cr} = \frac{k\pi^2 E}{12(1-\nu^2)}\left(\frac{t}{b}\right)^2 \geq \sigma_{max} \quad (27)$$

where k is the plate buckling factor, ν is the Poisson's ratio, b and t are the width and thickness of the plate, respectively. The limiting plate slenderness is

$$\left(\frac{b}{t}\right)_L = \sqrt{\frac{k\pi^2 E}{12(1-\nu^2)\sigma_{max}}} \quad (28)$$

For a bent web plate of an I-beam the theoretically calculated value of k=23,9, with the elastic modulus of E=7*10⁴ MPa, ν = 0,3,σ_{max} = p_{01} = 250 MPa one obtains

$$h / t_w \leq 77{,}8 \quad (29)$$

This value should be decreased according to BS to 51,4 considering the effect of initial imperfections and residual welding stresses. For another σ_{max} values and different design stresses Eq.(29) can be generalized as

$$\frac{h}{t_w} \leq \frac{1}{\beta} = 51{,}4\varepsilon; \quad \varepsilon = \sqrt{250/\sigma_{max}} \quad (30)$$

Since in the present example the maximum compressive stress is caused by $-F$, so $\sigma_{max} = FL/W_x$, or, assuming that the fatigue constraint Eq.(22) is active

$$\sigma_{max} = \Delta\sigma_N / (2\gamma_{mf}) \quad (31)$$

and *the local buckling constraint for the web* can be written as

$$t_w \geq \beta h; \quad 1/\beta = 51{,}4\sqrt{\frac{250}{\Delta\sigma_N/(2\gamma_{mf})}} \quad (32)$$

The calculations show that this constraint is always active thus it can be treated as equality. Then the objective function (26) is

$$A = 2W_0/h + 2\beta h^2 \quad (33)$$

and the condition $dA/dh = 0$ gives the optimal web height

$$h_{opt} = \sqrt[3]{2W_0/(2\beta)} \quad (34)$$

The local buckling constraint for the compression flange as an outstand with k=0,4 is given by

$$b/(2t_f) \leq 10{,}1\varepsilon \quad (35)$$

this value is decreased to 6ε (for semicompact cross-section), thus

$$t_f \geq \delta b; \quad 1/\delta_L = 12\varepsilon \quad (36)$$

and from Eq.(25) one obtains

$$b_{opt} = h_{opt}\sqrt{\frac{\beta}{2\delta}} \quad (37)$$

Finally, using Eq.(33) we get

$$A_{min} = \sqrt[3]{18\beta W_0^2} \quad (38)$$

which shows that the cross-sectional area (weight) can be decreased using higher values of 1/β.

Numerical data: F=50 kN fluctuating between $+F$ and $-F$, L=2 m, the number of cycles N=3*10⁵. The fatigue detail category for toe cracking of fillet welds connecting the flanges to a rigid column, according to BS is $\Delta\sigma_C$=20 MPa (Recommendations give 22 MPa). For the given number of cycles we get $\Delta\sigma_N$=37.6 MPa, thus, with Eq.(24) W_0=6.6489*106 mm³. Using Eq.(32) one obtains 1/β=210, this is unrealistic, so we use the value of 100 and Eq.(34) gives h_{opt}=1000 mm. Furthermore t_w=10 mm, 1/δ=49, we use 20, b_{opt}=320, t_f=16 mm.

For the root cracking of fillet welds connecting the flanges to the column $\Delta\rho_{\perp C}$=14, $\Delta\rho_{\perp N}$=26,3 MPa. The stress range in a flange is approximately (with 1/β=100)

$$\sigma_{max} = \frac{2FL}{2\beta h^3/3} = 30 \quad \text{MPa}$$

This stress is reduced to fillet welds to obtain the fatigue constraint

$$\rho_\perp = \sigma_{max}\frac{t_f}{2a_f} \leq \frac{\Delta\rho_{\perp N}}{\gamma_{mf}} \quad (39)$$

From Eq.(39) we get a_f=11,4 rounded 12 mm.

For fillet welds connecting the web to the column the fatigue constraint is defined by a vector resultant of the stress components

$$\Delta\rho_\perp = \sigma_{max}\frac{t_w}{2a_w} \leq \frac{150}{\gamma_w} \quad \text{and} \quad \Delta\tau_{//} = \frac{2F}{2ha_w} = \frac{50}{a_w}$$

From the fatigue constraint

$$\sqrt{(\Delta\rho_\perp)^2 + (\Delta\tau_{//})^2} \leq \Delta\rho_{\perp N}/\gamma_{mf} = 21{,}0 \quad \text{MPa} \quad (40)$$

we get a_w=7,5 rounded 8 mm.

2.4 Optimum design of welded box beams (Fig.6)

A simply supported beam is loaded by a constant, uniformly distributed static load p_G and by a variable force Q pulsating between $+Q$ and $-Q$ (Fig.6). In order to stiffen the box beam against torsional deformation of the cross-sectional shape, transversal diaphragms should be welded inside by fillet welds. The dimensions of the box beam h, $t_w/2$, b and t_f should be optimized to minimize the cross-sectional area

$$A = ht_w + 2bt_f \tag{41}$$

and fulfil the design constraints as follows.

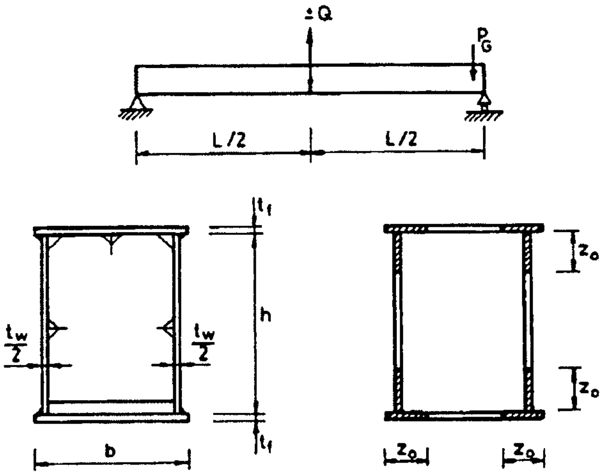

Fig.6. Main dimensions of a box beam and the reduced cross-section of an aluminium beam

The *fatigue constraint* is defined by

$$\Delta\sigma = \Psi_d \frac{2QL}{4W_x} \le \frac{\Delta\sigma_N}{\gamma_{Mf}} \tag{42}$$

$$W_x = 2I_x/(h+t_f); \quad I_x = h^3t_w/12 + 2bt_f(h+t_f)^2/4 \tag{43}$$

where Ψ_d is a dynamic factor, W_x is the elastic section modulus, I_x is the moment of inertia, $\Delta\sigma_N$ is the stress range corresponding to the given number of cycles N. According to Recommendations, for plates with transverse stiffeners $\Delta\sigma_C$=28 MPa. Eq.(3) can also be used. Note that, according to BS, in the fatigue constraint the section properties should not be reduced for HAZ.

The *static stress constraint* is expressed by

$$\sigma_{max} = \left(\gamma_G p_G L^2/8 + \gamma_Q \Psi_d QL/4\right)/W_{x.red} \le p_{01} \tag{44}$$

The section modulus should be reduced considering the HAZ softening effect.

According to BS

$$W_{x.red} = \frac{2I_{x.red}}{h+t_f}, \quad I_{x.red} = I_x - 4z_0\frac{t_f}{2}\left(\frac{h+t_f}{2}\right)^2 - 4z_0\frac{t_w}{4}\left(\frac{h}{2}-\frac{z_0}{2}\right)^2 \tag{45}$$

where the width of the HAZ is

$$z_0 = 3t_B^2/t_A \tag{46}$$

if $t_w/2 \le t_f$ $\quad t_B = t_w/2$

if $t_w/2 > t_f$ $\quad t_B = t_f$

if $0{,}5(t_w/2 + t_f) \le 1{,}5t_B$ $\quad t_A = 0{,}5(t_w/2 + t_f)$

if $0{,}5(t_w/2 + t_f) > 1{,}5t_B$ $\quad t_A = 1{,}5t_B$

The local buckling constraints, according to BS, are as follows:
for flange

$$(b-40)/t_f \le 18\sqrt{250/\sigma_{max}} \tag{47}$$

and webs

$$2h/t_w \le 18/0{,}35\sqrt{250/\sigma_{max}} \tag{48}$$

The deflection constraints, according to BS for beams in buildings, are as follows:

$$\frac{5p_G L^4}{384EI_x} \le \frac{L}{200} \tag{49}$$

$$\frac{5p_G L^4}{384EI_x} + \frac{QL^3}{48EI_x} \le \frac{L}{200} \tag{50}$$

Numerical data: Q=6 kN, L=12 m, Ψ_d=2; the values of p_G are varied to show that for low p_G/Q ratios the fatigue constraint, for large ratios the deflection constraints are active. The Rosenbrock's Hillclimb mathematical programming method has been used for computerized optimization. Rounded values are computed by a complementary special program. The results are summarized in *Table 1*. It can be seen that, depending on the p_G/Q ratio, the fatigue or the deflection constraint is active.

p_G (N/mm)	3	12	21	25	30
h*t_w/2 b*t_f	705*10 245*18	625*14 290*17	665*19 280*14	665*19 260*19	695*20 290*19
A (mm²)	15870	18610	20275	22515	24920
active constraint	fatigue	fatigue	fatigue and deflection	deflection	deflection

Table 1. Optimal dimensions (mm) and minimal cross-sectional areas of welded aluminium box beams for Q=6 kN and various values of p_G

3. CONCLUSIONS

In the case of a SHS strut cyclically loaded by tension-compression and welded to gusset plates the optimum width and thickness can be calculated considering the fatigue and local buckling constraints. The fatigue stress range of welded aluminium joints is determined according to BS 8118.

A CHS strut with a welded splice is designed for fatigue stress range of toe cracking of the connecting fillet welds and the required weld size is determined on the basis of root cracking.

Similar design aspects are applied to the optimum design of a welded I-section cantilever beam.

In the case of a simply supported box beam with welded diaphragms the optimum dimensions of webs and flanges are determined either by the fatigue or by the deflection constraints.

In the local buckling constraints the actual static design stress level is taken into account which can be lower than the limiting static stress.

In the static stress constraint the section modulus should be decreased considering the effect of the heat affected zone.

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