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# Minimum cost design of a truck floor welded from aluminium-alloy profiles

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**Abstract:** The aluminium floor of a truck produced by the US-Hungarian company Alcoa-Köfém in Hungary consists of extruded al-alloy longitudinal and cross members as well as a tread deck plate. It is shown that, using an optimum design process, significant mass and cost savings may be achieved by decreasing the deck plate thickness and changing the profile, dimensions and number of cross members. Design constraints relate to fatigue stress range of welded joints, to local buckling of extruded profiles and to fabrication size limitations. A special loading case is also considered when a wheel is staying on a curb and the floor is distorted.

**Key words:** truck floor, welded aluminium joints, fatigue of welded joints, aluminium-alloy extruded profiles, minimum cost design

## Introduction

The US-Hungarian company Alcoa-Köfém Ltd produces trucks for beverage transport. The truck structure has a steel chassis consisting of two longitudinal beams. The al-alloy subframe is constructed from two longitudinal beams bolted on steel beams. The al-alloy floor structure has three layers as follows (Fig.1): cross members welded to subframe, longitudinal members welded to cross members, tread deck plate distributing the pallet loads. The material of cross members is an al-alloy AlMgSi0.7 according to German standard DIN 1725 [1] of $R_{p.0.2}$ = 215 MPa according to DIN 1748 [2] (international alloy type 6005A).The tread deck plate material is an al-alloy AlMg2.5 (international alloy type 5052). These main structural parts are framed by side rails, which carry the loads from roof, sidewalls and doors.

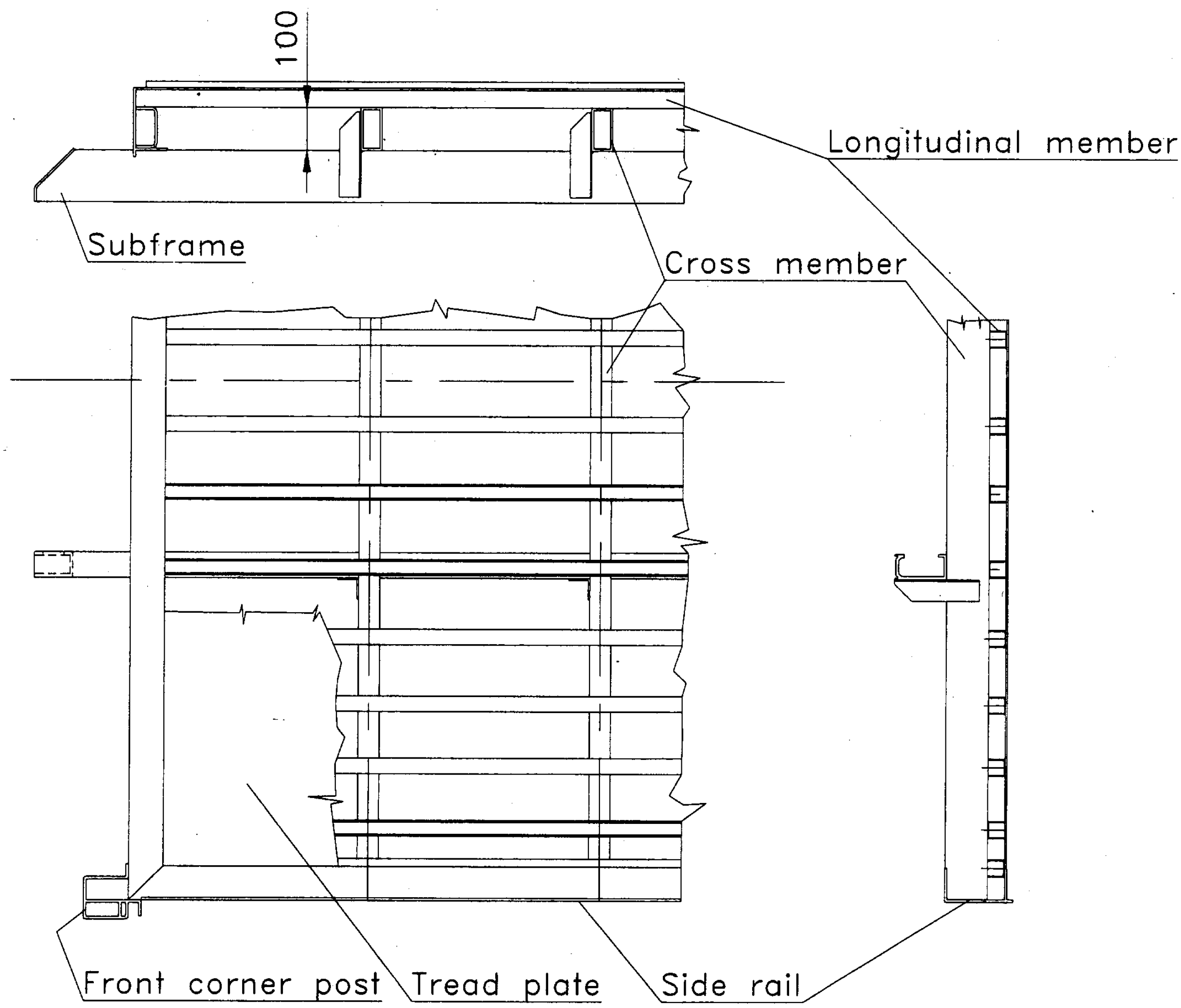


Fig.1. Truck floor structure

Our aim is to decrease the material cost of floor structure by changing the profile, dimensions and number of cross members as well as the thickness of deck plate.

**Load cases**

Two load cases should be considered in the design of cross members as follows: (a) loads due to pallets, roof, door and side walls in the horizontal floor position; (b) the same loading as in (a) but a wheel is staying on a curb, thus, the floor is distorted.

Loads acting on an outside cross member are as follows:

| | | |
|---|---|---|
| a corner column | | 205 N |
| roof | 2060/4 | 515 N |
| upper door | 1420/2 | 710 N |
| front wall | 1033/2 | 516 N |
| | $F_1$ = | 1946 N |

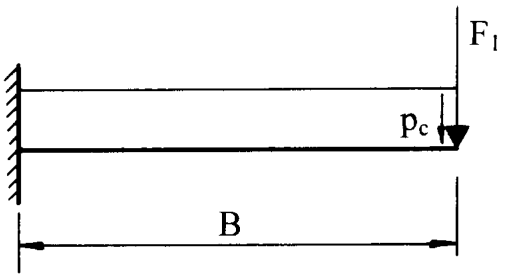


Fig.2. Loads on the cantilever part of cross members

Load from pallets: mass of a pallet is $F_p$ = 8500 N, intensity of the uniformly distributed load is $p = F_p n_p/(BL)$, where the number of pallets placed on the half floor $n_p$ = 5, $B$ and $L$ are the dimensions of a half cantilever floor surface. The uniformly distributed normal load acting on a cross member is $p_c = pL/(n_c-1)$, $n_c$ is the number of cross members.

The maximum bending moment in a cross member is (Fig.2)

$$M_{\max} = \frac{p_c B^2}{2} + F_1 B = \frac{F_p n_p B}{2(n_c - 1)} + F_1 B \qquad (1)$$

Calculating with $F_p$ = 8500 N, $n_p$ = 5, $B$ = 720 mm, $F_1$ = 1946 N one obtains bending moments for different numbers of cross members. This number is limited by the dimension of pallets (800 mm) to $n_{c.min}$ = 10. Since the original number of cross members is 14, we calculate with $n_c$ = 14, 12 and 10. For these values of $n_c$ one obtains

$M_{14}$ = 2.578, $M_{12}$ = 2.792 and $M_{10}$ = 3.1011 kNm.

The corresponding shear forces are as follows:

$Q = F_p n_p/(n_c\text{-}1) + F_1$; $Q_{14}$ = 5215, $Q_{12}$ = 5810 and $Q_{10}$ = 6668 N.

*Loads on the distorted floor*

Measurements have been carried out on a truck loaded by pallets and with a wheel staying on a curb in a height of 91 mm. The measured deflections have shown that the cross members near the wheel being lifted up are loaded by bending as it is seen on Fig.3. This cross member can be modelled as a cantilever beam of its whole length $L_c$ loaded by a force $F$ corresponding to a deflection $w$. This deflection can be approximately calculated as $w = 138 - L_c\varphi$, where $L_c$ = 2427 mm, $\varphi(rad) = 2.91^0 \pi / 180^0 = 0.0508$, thus, $w$ = 15 mm. Furthermore

$$F = \frac{3EI_x w}{L_c^3}; M_{c.\max} = FL_c \tag{2}$$

where $E = 7\text{x}10^4$ MPa is the elastic modulus, $I_x$ is the second moment of area.

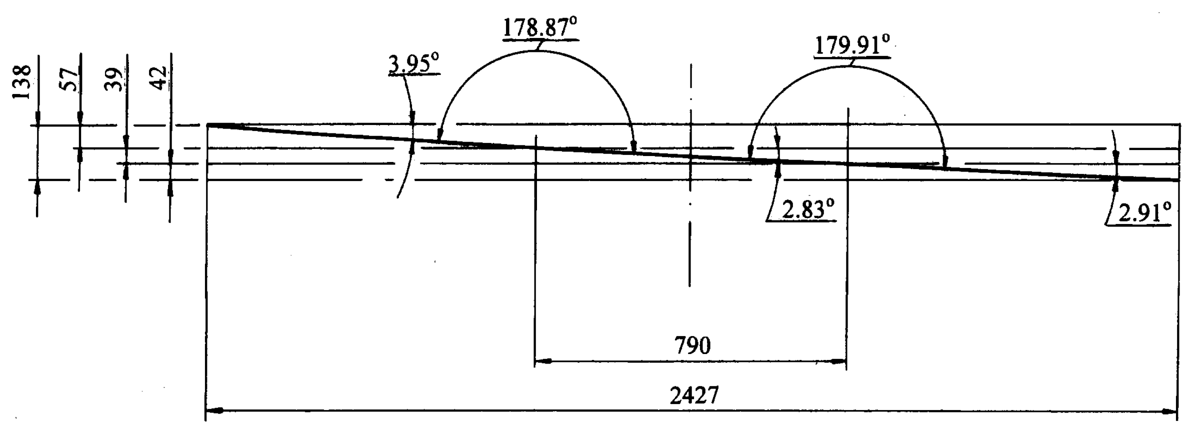


Fig.3. Measured deflections of a distorted cross member, when a left truck wheel is staying on a curb

## Geometric characteristics of cross members

The cross-section loaded by bending and shear consists of a cross member and a part of the deck plate (Fig.4). We calculate an effective width of the deck plate 50$t$, $t$ is the thickness. In the case of a rectangular hollow section (RHS) the geometric characteristics of this cross section are as follows:

$$A = A_1 + A_2; \quad A_1 = 2ht_w + 2bt_f; \quad A_2 = 50t^2 \tag{3}$$

$$y_G = \frac{A_1}{A}\left(\frac{h+t}{2} + c\right); \quad y_c = h + c + \frac{t}{2} - y_G \tag{4}$$

$$I_x = \frac{h^3 t_w}{6} + \frac{bt_f h^2}{2} + A_1\left(y_c - \frac{h}{2}\right)^2 + A_2 y_G^2 \tag{5}$$

In the case of I- and C-profiles (Fig.4) the characteristics are as follows:

$$A_1 = ht_w + 2bt_f \tag{6}$$

$$I_x = \frac{h^3 t_w}{12} + \frac{bt_f h^2}{2} + A_1\left(y_c - \frac{h}{2}\right)^2 + A_2 y_G^2 \tag{7}$$

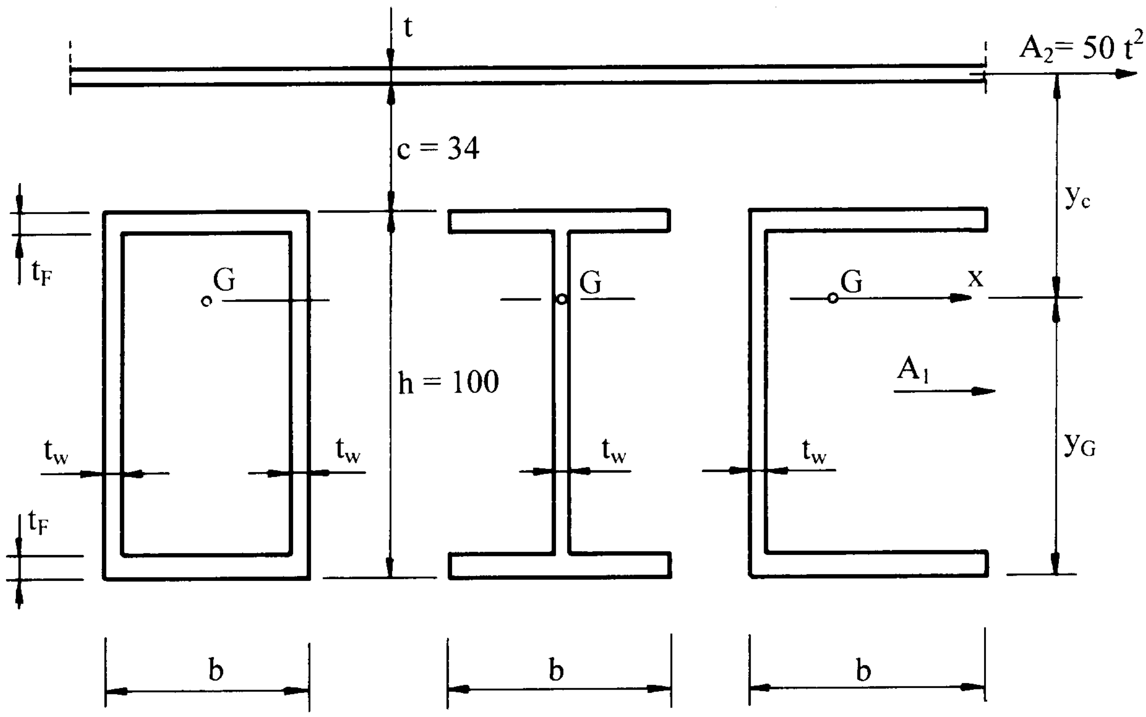


Fig.4. Cross-sections of cross members

**Design constraints**

*Constraints on fatigue stress range for horizontal floor position*

$$\sigma_1 = \frac{M_{\max}}{I_x} y_{\max} \le \frac{\Delta\sigma_N}{\gamma_{Mf}}; \qquad y_{\max} = \max(y_G, y_c) \tag{8}$$

$$\tau_1 = \frac{Q}{A_w} \le \frac{\Delta\tau_N}{\gamma_{Mf}}; \tag{9}$$

where $A_w = 2ht_w$ for rectangular hollow section and $A_w = ht_w$ for other profiles.

Since the cross members are welded to longitudinal subframe beams, they should be designed considering the fatigue of welded joints. According to [3] the fatigue stress range for number of cycles 2x10$^6$ in the case of transverse stiffener welded on girder web (detail 512 for structural aluminium alloys) is $\Delta\sigma_C = 28$ MPa. Calculating with a realistic number of cycles $N$ = 2x10$^5$,

$$\log\Delta\sigma_N = \frac{1}{3}\log\frac{2x10^6}{2x10^5} + \log\Delta\sigma_C = 1.78049; \quad \Delta\sigma_N = 60.3 \text{ MPa} \tag{10}$$

With a safety factor of 1.25

$$\frac{\Delta\sigma_N}{\gamma_{Mf}} = \frac{60.3}{1.25} = 48.2 \text{ MPa} \tag{11}$$

For shear it is $$\Delta\tau_C = \Delta\tau_N = 71/3^{1/2} = 40.99; \quad \frac{40.99}{1.25} = 32.8 \text{ MPa} \tag{12}$$

It should be mentioned that we calculate with the bending moment also from static load $F_1$ in the fatigue constraint as an approximation in the safe side.

*Constraint on fatigue stress range for distorted floor position*

$$\sigma_2 = \frac{M_{c.\max}}{I_x} y_{\max} = \frac{3Ew}{L^2} y_{\max} \le \frac{\Delta\sigma_{N1}}{\gamma_{Mf}} \tag{13}$$

In the case of distorted floor position the maximum bending moment arises at the end of cross member, where it is welded to subframe by fillet welds. For this joint, according to [3] (detail No.413) $\Delta\sigma_{C1} = 22$ MPa and a realistic number of cycles $N = 10^5$ it is

$$\frac{\Delta\sigma_{N1}}{\gamma_{Mf}} = \frac{59.7}{1.25} = 47.7 \text{ MPa} \tag{14}$$

*Constraints on local buckling of profiles*

Flange of rectangular hollow section (according to [4])

$$b/t_f \le 22\varepsilon; \quad \varepsilon = \left(\frac{250}{1.5\sigma_{\max}}\right)^{1/2}; \quad \sigma_{\max} = \max(\sigma_1, \sigma_2) \tag{15}$$

where the multiplier 1.5 is the safety factor for static buckling.

Webs of hollow section

$$h/t_w \le 22\varepsilon/g; \tag{16}$$

$$g = 0.65 + 0.35\frac{y_0}{y_c} \qquad \text{when} \qquad 1 \geq \frac{y_0}{y_c} \geq 0$$

$$g = 0.65 + 0.30\frac{y_0}{y_c} \qquad \text{when} \qquad 0 \geq \frac{y_0}{y_c} \geq -1$$

$$y_0 = y_G - \frac{t}{2} - c \tag{17}$$

$$y_c = h + c + \frac{t}{2} - y_G \tag{18}$$

Flange of I-section (unreinforced)

$$b/(2t_f) \leq 7\varepsilon \tag{19}$$

Flange of C-section (unreinforced)

$$b/t_f \leq \varepsilon \tag{20}$$

*Fabrication constraints: size limitations*

Some constant dimensions are prescribed by the original structure as follows:

$$h = 100, \quad c = 34 \text{ mm} \tag{21}$$

The web thickness is limited to

$$t_{w.min} = 3.4 \text{ mm} \tag{22}$$

to guarantee the quality of welding.

The tread plate thickness is limited to

$$t_{min} = 2 \text{ mm} \tag{23}$$

Since the cross members should be welded to side rails, the extruded shapes should not have any reinforcing ribs or bulbs, since they are in the way of welding.

It should be mentioned that the extruded I- or C-profiles with or without reinforcing ribs or bulbs optimized for pure bending have the same minimum cross-section area, thus, the use of ribs or bulbs does not result in mass savings.

## Optimization characteristics and results

*The objective function* is the cross-sectional area of cross members and deck plate part (Eq. 3).

*The unknown variables* are the dimensions of profile flanges $b$ and $t_f$.

*The constraints* are as follows: Eqs 8, 9, 13, 15, 19, 20, 21, 22, 23.

The optimization is performed for *three profiles* (RHS, I and C) and for *three numbers of cross members* $n_c$ = 14, 12 and 10.

*Mathematical method*: the Rosenbrock's hillclimb algorithm is used [5].
Results are summarized in Table 1.

Table 1. Optimum flange dimensions ($b$, $t_f$) in mm, minimum cross-section areas of profiles $A_1$ in mm$^2$, the mass of all the cross members ( $\rho A_1 L_{cm}$ ) ($L_{cm}$ = 2440$n_c$) in kg for three profiles and three numbers of cross members as well as the cost of tool $C_T$

| Profile | | $n_c$ = 14 | $n_c$ = 12 | $n_c$ = 10 |
|---|---|---|---|---|
| RHS | $b$ | 55 | 115 | 12 |
| | $t_f$ | 5.4 | 3.0 | 3.4 |
| | $A_1$ | 1274 | 1370 | 1496 |
| | $\rho A_1 L_{cm}$ | 117.50 | 108.31 | 98.56 |
| | $C_T$ $ | 1320 | 3537 | 3537 |
| I | $b$ | 55 | 60 | 65 |
| | $t_f$ | 7.2 | 7.2 | 7.8 |
| | $A_1$ | 1132 | 1264 | 1354 |
| | $\rho A_1 L_{cm}$ | 104.41 | 99.93 | 89.20 |
| | $C_T$ $ | 927 | 927 | 927 |
| C | $b$ | 65 | 65 | 65 |
| | $t_f$ | 6.1 | 6.9 | 7.9 |
| | $A_1$ | 1133 | 1237 | 1367 |
| | $\rho A_1 L_{cm}$ | 104.50 | 97.79 | 90.06 |
| | $C_T$ $ | 927 | 927 | 927 |

**Mass savings**

It can be seen from Table 1. that, for the minimum mass the I- or C-shapes can be selected using 10 cross members. In this case the mass of cross members and the tread plate of thickness $t$ = 2 mm is 89.20 + 2.7x2x2.280x6.570 = 89.20 + 80.89 = 170.09 kg.

For the sake of comparison we calculate the mass of the original solution having 14 cross members of rectangular hollow section with dimensions of $h$ = 100, $t_w$ = 5, $b$ = 50 and $t_f$ = 5 ($A_1$ = 1400 mm$^2$) and $t$ = 4.5 mm: 2.7x4.5x2.28x6.57 + 2.7x1.4x14x2.44 = 182.00 + 129.12 =

311.12 kg. Thus, the optimization results in 311.12 - 170.09 = 141.03 kg mass savings for one truck.

**Cost savings**

*Cost of tread deck plate*

| | |
|---|---|
| London Metal Exchange (LME) price of aluminium | 1.5885 $/kg |
| surcharge | 0.9750 |
| total | 2.5635 $/kg |
| Cost of the original plate ($t$ = 4.5) 182x2.5635 | 466.6 $ |
| Cost of the optimized plate ($t$ = 2 mm) 80.89x2.5635 | 207.4 $ |

*Cost of cross members*

| | |
|---|---|
| LME | 1.5885 $/kg |
| extrusion work upcharge | 1.3250 $/kg |
| total | 2.9135 $/kg |
| cost of original cross members 2.9135x129.12 | 376.19 $ |
| cost of tool | 1320 $ |
| total extruded length for 50 trucks/year 14x2.44x50 | 1708 m |
| tool surcharge 1320/1708 | 0.77 $ |
| total cost of original cross members | 376.96 $ |
| total cost of the original tread plate and cross members | 466.6 + 376.96 = 843.56 $ |
| cost of optimized cross members (I, $n_c$=10) 2.9135x89.2= | 259.88 $ |
| cost of tool | 1854 $ |

It should be mentioned that, as it is seen in Table 1, the tool cost is very high for RHS profiles for $n_c$ = 12 and 10, thus, we calculate with I- or C-profiles

| | |
|---|---|
| tool surcharge 1854/(10x2.44x50) | 1.52 $ |
| total cost of optimized cross members | 261.4 $ |
| total cost of the optimized tread plate and cross members | 207.4 + 261.4 = 468.8 $ |
| *Cost savings for one truck* | 843.56 - 468.8 = 374.76 $ |

## Conclusions

In the case of a truck floor welded from al-alloy extruded profiles and a deck plate the systematic optimum design process can result in significant savings in mass and cost.
A cross-section is optimized consisting of an extruded cross member and an effective part of the deck plate. The objective function is the cross-sectional area, the design constraints relate to fatigue stress range of welded joints and to local buckling of extruded profiles. Fabrication aspects regarding the size limitations are also considered.
In addition to the loading by pallets in horizontal floor position the case of distorted floor position is also taken into account, when a truck wheel is staying on a curb. The bending moments arising in this position have been calculated on the basis of experimental measurements of deflections.
Optimization shows that the thickness of deck plate can be decreased from 4.5 to 2.0 mm, the original number of cross members can be decreased from 14 to 10, and the original cross member shape (RHS) can be replaced by I- or C-profile having optimum dimensions. These changes can result in 141 kg mass and 375 $ cost savings for a truck structure.

## Acknowledgement

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