Fundamentals and preliminary sizing of sections and joints

Objectives:
- Overview of section and joint structural behaviour
- To outline the design and sizing approach

Characteristics of thin walled sections:
- Open section – thin sheet metal is formed with a discontinuity
- Closed section – the section forms a complete loop
Open sections
- the angle section (a) is not a suitable structural member because bending will occur both axes. In addition stress distribution is very asymmetric and results in large parts of the section being under stresses.
- the z section (b) has similar undesirable characteristics
- the channel section (c) is a suitable structural section and used in commercial vehicle chassis and body structures. However, it is less stiff about Y-Y axis.
- the wide flange results in a low stress at which buckling occurs.
Adding a lip to the channel (d) can improve in buckling stress
- the hat section (e) has good bending provided $2b_2=b_1$
Open section

The major limitation of all open section is their lack of torsional stiffness due to very low polar second moment of area:

Section a: \( J_x = (a + b)t^3/3 \)
Section b: \( J_x = (b_1 + b_2 + d)t^3/3 \)
Section c: \( J_x = (2b + d)t^3/3 \)
Section d: \( J_x = (2d_1 + d + 2b)t^3/3 \)
Section e: \( J_x = (b_1 + 2b_2 + 2d)t^3/3 \)

\[ \Theta = \frac{TL}{GJ_x} \]
Closed sections

• The main advantages of all closed sections is that they have greatly improved torsional stiffness.
• the closing of the section by the spot weld
• polar second moment of area: $J_x = 4A^2t/s$
  where $A(a) = (b_1-2b_2)d+4(b_2-b_3)t$
  $s = 2(b_1-2b_3)+2d+4(b_2-b_3)$
Passenger car sections

- Actual passenger car sections are more complex in detail profiles.
- When calculating stress, evaluation of section moments of area and torsion constant can be found using modern CAD methods.
- Structural properties, practical installation and functional requirements are the criteria for choosing types of section.
Example of initial section sizing
Front floor cross-beam

Bending moment
\[ M = K_2 l_1 - F_{fp} l_2 \]

Stress due to bending
\[ f = \frac{M y}{I} \]

Shear stress
\[ \tau = \frac{K_2 A y}{z I} \]
The A-pillar

- The A-pillar has large bending loads when structure is loaded in torsion.
- Bending moment about X-X axis:
  \[ M_x = \frac{Q_1 h}{4} \]

And bending stress:

\[ F_{bx} = \frac{M_x b}{I_{xx} 2} \]

The normal force gives rise to a bending moment:

\[ M_y = (nQ_2 \sin \alpha) h / 2 \cos \alpha \]

will cause a bending stress:

\[ f_{by} = \frac{M_y d}{I_{yy} 2} \]

Direct force into the pillar giving s stress:

\[ f_c = \frac{nQ_2 \cos \alpha}{(2b + 2d) t} \]

\[ f_{total} = f_c + f_{by} + f_{bx} \]
Engine longitudinal rail

This component is subjected to shear forces and bending moments.

Due to high shear and bending moments, deeper section is needed and leads to larger depth, $d$.

This section should also be designed to absorb energy in frontal impact.
Sheet metal joints

- If a moment is applied it will cause tension and compression. The net result is there cannot be any significant bending stiffness.
- Vertical shear force is carried by the side flange. If the side flange is removed, the top/bottom flanges will be ineffective to resist the normal force and hence they will distort.
- Horizontal shear force will result in opposite condition as shown above.
- If the cross-beam is subject to a torsion the shear flow around the section is:
  \[ q = \frac{T}{2A} = \frac{T}{2d'v'} \]

Two important rules for designing joints:

a) Avoid out-of-plane bending on thin sections
b) Load thin sections with in-plane bending and shear.
Spot welds
Spot weld and connector patterns

\[
\bar{y} = \frac{3y_1 + 3y_2}{8}
\]
\[
\bar{x} = \frac{2x_1 + 3x_2}{8}
\]

\[
\frac{F_{E1}}{r_1} = \frac{F_{E2}}{r_2} \ldots \frac{F_{En}}{r_n} = K
\]

\[
\sum_{n=1}^{n} F_{En} r_n = M
\]

\[
\sum_{n=1}^{n} Kr_n^2 = M
\]

\[
K = \frac{M}{\sum_{n=1}^{n} r_n^2}
\]

- Spot weld 7 has very little load while spot weld 3 experience heavy load.
- Some repositioning could be made to redistribute loads.
Spot welds along a closed section

\[ \tau = q / t = T / 2At = T / 2d(b_1 - 2b_2)t \]

\[ N = qL / F_s \quad \text{No of spots required} \]

\[ P_e = K\pi^2 EI / L^2 \quad \text{Buckling load, K}=3.5 \]

\[ f_{cr} = P_e / A = 3.5\pi^2 EAk^2 / Ap^2 \]

\[ f_b = Md / 2I \quad \text{Actual stress due BM} \]

Pitch requirements to prevent buckling

\[ p = \sqrt{\left(3.5E\pi^2 t^2 / 12f_b\right)} \]
Shear Panels: Roof

Roof is the largest panel in a passenger car and under the torsion. This may lead to buckle issue.

The shear stress at which the panel buckles is given by:

\[
\tau = KE(t/b)^2 \quad \text{K=buckling coefficient}
\]

Investigation made into a roof panel 980mm x 1250mm x 1mm x 2425mm (curvature), the stress cause buckling is 13.88N/mm² – 4x than applied shear stress

Another approach to designing roof and other panels is to consider the vibration characteristics. ESDU 75030 can be used to predict panel vibrations. Range of frequency from 4 to 33 Hz.

Low frequency vibrations will cause ride and comfort problems.