Dimension of Roll bars for offshore raceboat

In Paragraph 508 it is stated that a roll bars must be fitted into the canopy. It is here assumed that the most important task for the roll bar is to guarantee that the canopy does not collapse in the event of a high-speed stuffing or if the boat lands upside down. Here a boat with a tandem canopy is studied. (If a side by side configuration is used the load on the windshield will be much greater. Mark Lavin foundation guidelines demand the use of a water deflector in front of the windshield and also to use two roll bars.) To make the canopy as resistant as possible the roll bar is positioned at the top of the windshield and slanted slightly backwards. Depending of the design of the canopy a second roll bar positioned at the backrest or the driver's chair may be needed.

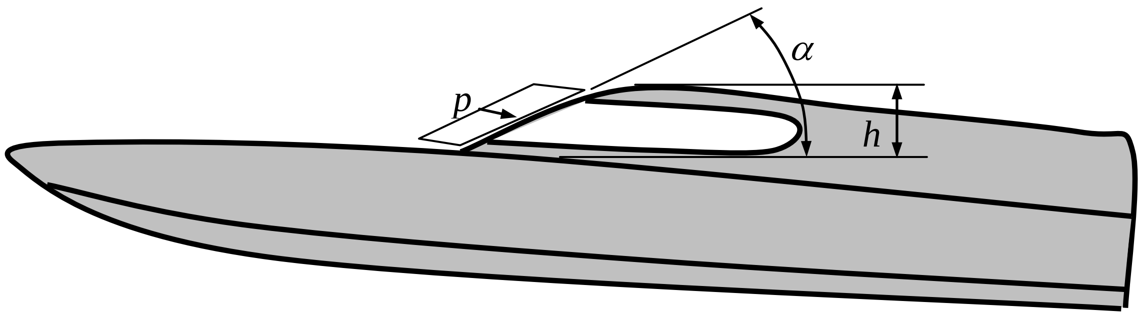

Figure 1. Side view of UIM class 3 race boat. Showing dynamic pressure p from impact with water acting on the canopy. Height of canopy h and angle of windshield α. According to Mark Lavin the angle of the windshield α shall be less than 35°.

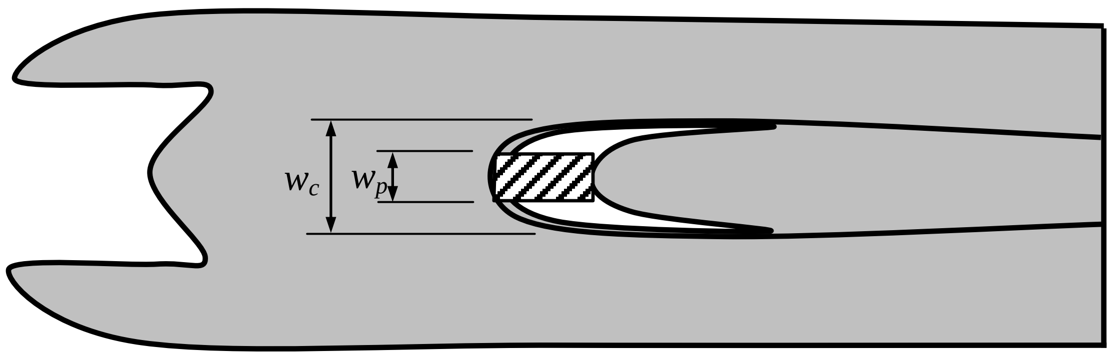

Figure 2. Top view of UIM class 3 race boat fitted with canopy. Width of canopy w_c and pressurised width of canopy w_p.

It's here assumed that the pressurised width of the canopy w_p is one third of the width of the canopy w_c and also that full dynamic pressure is reached, which is on the conservative side and therefore no safety factors will be used in the following analyse. The load from the water on the windshield F_w will be dynamic pressure p times front area A there the dynamic pressure p depends on the velocity v of the boat and the density ρ of the water.

$$p = \frac{\rho\ v^2}{2} \tag{1}$$

$$F_w \approx p\ A = p\ h\ w_p = \frac{p\ h\ w_c}{3} \tag{2}$$

The forward roll bar position at the top of the windshield must be able to fully support this load but also a load from the side since the might turn some before hitting the water surface. The side load is here assumed to be 50% of the load from the windshield, see figure 4.

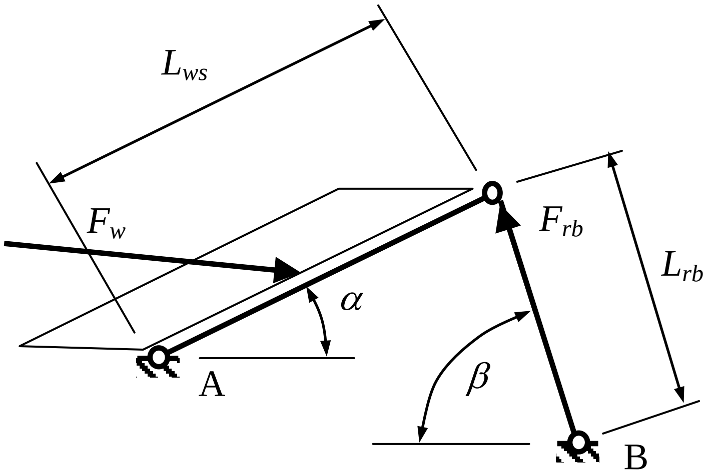

Figure 3. Side view sketch of impact load windshield and roll bar. L_{ws} is length of windshield and L_{rb} is length of roll bar. β is the angle of the roll bar.

A equilibrium around point A gives

$$\text{A}: \quad F_w\, h/2 - F_{rb} L_{ws} \sin(\alpha+\beta) = 0 \tag{3}$$

Where

$$L_{ws} = h/\sin\alpha \tag{4}$$

Equations (1)-(4) gives

$$F_{rb} = \frac{\rho\ v^2 h\ w_c \sin(\alpha)}{12\ \sin(\alpha+\beta)} \tag{5}$$

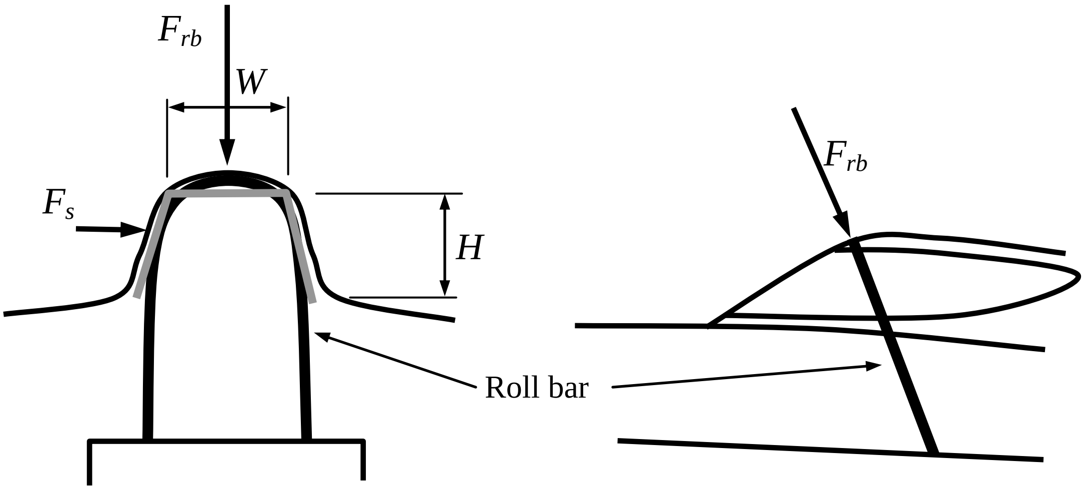

Figure 4. Front roll bar. The gray three beam are an approximation of the roll bar with the loads F_s and F_{rb} acting on it. The roll bar has the height H and the width W.

It is assumed that the roll bar has a good support sideways at deck level and therefor is only the top part of the roll bar studied here. The roll bar is here modelled as simply supported since the stiffness of the mounting to the side walls is unknown.

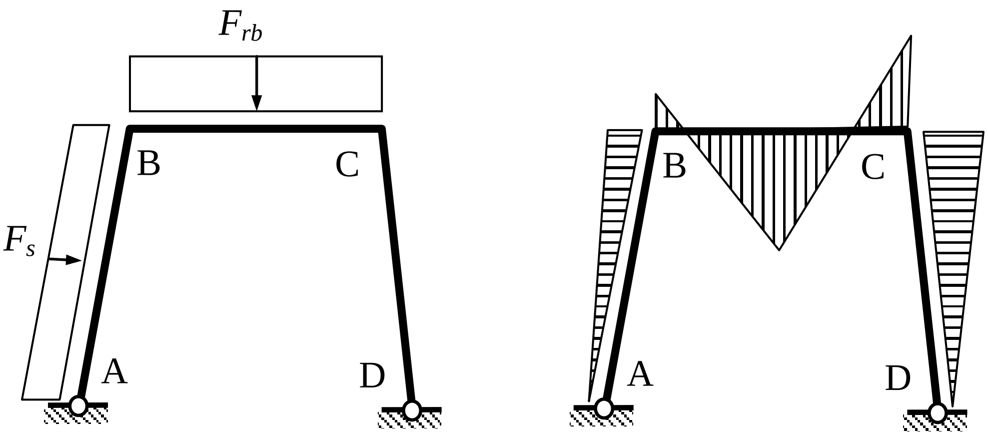

Figure 5. a) Schematic shetch of the roll bar and b) the bending moment acting on it.

Bending moment at point B

$$M_b = -\frac{F_{rb}L}{4\left(\frac{2H}{3}+3\right)} + \frac{3F_sH}{8} * \frac{\frac{H}{L}+2}{\frac{2H}{L}+3} = \tag{6}$$

$$\frac{3F_sH^2 + 6F_sH\ L\ - 2F_{rb}L^2}{8(2H+3L)}$$

Bending moment at the middle of the to beam

$$M_m = \frac{F_{rb}L}{2} - \frac{F_{rb}L}{4\left(\frac{2H}{L}+3\right)} = \tag{7}$$

$$\frac{F_{rb}L(4H+5L)}{4(2H+3L)}$$

Bending moment at point C

$$M_b = -\frac{F_{rb}L}{4\left(\frac{2H}{L}+3\right)} - \frac{F_sH}{8} * \frac{\frac{5H}{L}+6}{\frac{2H}{L}+3} = \tag{8}$$

$$-\frac{2F_{rb}L^2 + 5F_sH^2 + 6F_sHL}{8(2H+3L)}$$

The stress may then be calculated with equation (9)

$$\sigma = \frac{M}{I}y \tag{9}$$

An roll bar with rectangular cross section made of glass fibre reinforced plastic is studied. This solution has many advantages ie a large surface against the canopy to spread out forces evenly, can take high impact loads since it is flexible and can easily be made to a good fit against the canopy. Tri-axial or uni-axial fabrics are to be used with a high strength iso-polyester or epoxy.

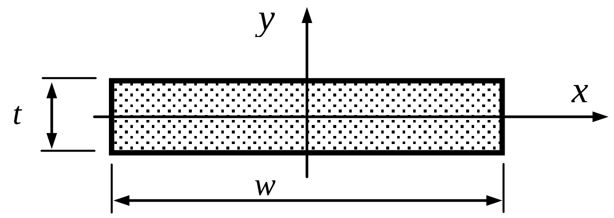

Figure 6. Cross-section of roll bar with the width *w* and the thickness *t*

The maximal stress of the roll bar with rectangular cross section of figure 6 is with equation (9) and the maximal value on mending moments from equation (6)-(8)

$$\sigma = \left|\frac{6M}{w * t^2}\right| \tag{10}$$

The allowed flexural stress in glass fibre reinforced plastics in MPa using bi- and tri axial fabrics according to the coming ISO 12 215 - 5 standard for scantlings of pleasure crafts is

$$\sigma = 400 * wf - 10 \tag{11}$$

And for uni directional fabrics

$$\sigma = 1800 * wf^2 - 1400 * wf + 510 \qquad (12)$$

There *wf* is fibre content by weight. Using (11) or (12) together with (10) gives the dimension of the cross section.

Example

An example of typical UIM 3C catamaran fitted with canopy with a roll bar built mainly of uni directional glass fabrics

	Value	Unit
Density of water ρ	1025	kg/m^3
Velocity of boat v	40	m/s (~78 knots) note higher for many boats
Height of canopy h	0.750	m
Width of canopy w_c	0.800	m
Angle of windshield α	35	°
Angle of roll bar β	75	°
Load on roll bar F_{rb}	50	kN (~5 tons)
Height of roll bar H	0.7	m
Width of roll bar W	0.7	m
Fibre content wf	55	%

Table 1. Data used as an example of roll bar for a canopy equipped UIM class offshore 3C race boat.

The data above gives the cross section combinations of width and thickness in figure 7 to chose from.

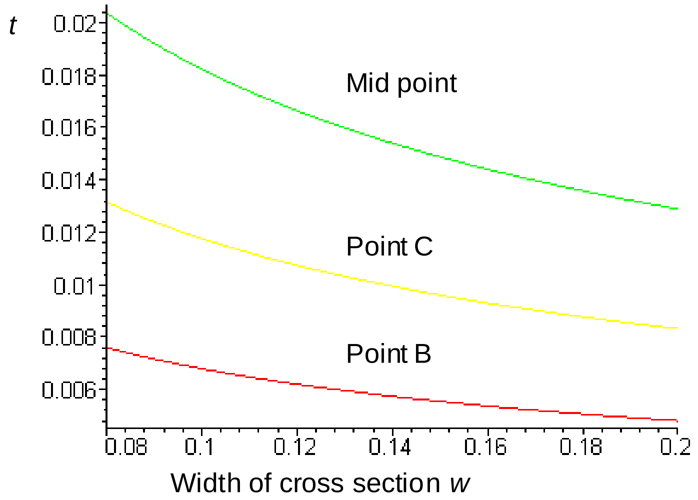

Figure 7. Width *w* versus thickness *t* fore the cross section of the roll bar for the different points of the roll bar according to figure 5b.

If the cross section width of the roll bar is chosen to 150 mm the thickness has to be 16mm.

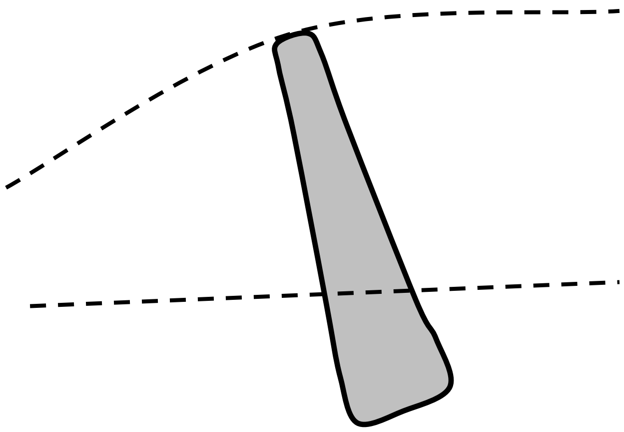

Figure 8. A side view of a prefered design of roll bar

Since the bending moments on the lower part of the roll bar a wider and thinner cross section has benefits, larger mounting surface to the boat structure and requires less space, see figure 8.