Appendix D
Design of Padeye

In addition to the basic requirements in 4.4.1, the designer of an offshore container should take the following into account:

D.1 Padeye Position

The pad eyes should be positioned on the container to preclude, as far as practicable, the risk of slings fouling against the container or its cargo during normal use. Pad eyes should be so designed as to permit free movement of the shackle and sling termination without fouling the pad eye. They should as far as possible be designed to avoid damage from other containers.

Pad eye that are placed vertically and aligned towards a central point can normally accommodate variations in sling angles, i.e. a lifting set with the legs 45° from vertical can be replaced with a longer lifting set, giving a smaller angle to the vertical without any adverse effect on the pad eyes or the container. If the pad eyes instead are positioned at an angle to the vertical, they can not accommodate any variation in the sling angle and the legs must always be of exactly the right length.

Containers that are designed to have the centre of gravity offset from the geometric centre may be fitted with lifting sets of asymmetric length, so that the container will hang horizontally when lifted. If the lifting set is asymmetric, the pad eyes must be aligned towards the lifting centre.

D.2 Design

Pad eye that are partly slotted into primary structure members are generally considered to be preferable, but other designs will also be considered for approval.

The pad eye design must take into account the size and shape of the shackles that are going to be used. Once the pad eye has been designed, only one size of shackle will fit.

Since shackles come in standard sizes, the designer should determine the size of the shackles to be used before designing the pad eyes. He must take into account the thickness of the shackle pin, the inside width and length of the shackle and the free space needed to fit the shackle.

Dee or bow (omega) shackles are usually designed according to one of the common standards used internationally (EN 13889, US Federal Specification RR-C-271 or ISO 2415). However, some shackle manufacturers use their own standards, with different design and sizes of shackles. Standard shackle sizes according to EN 13889 are given in the following table.

Table D-1 Standard shackles

Nominal WLL (tonnes)	*Pin diameter* (mm)*	*Inside width at pin (mm)*	*Inside length of Dee shackles (mm)*	*Inside length of Bow shackles (mm)*
3.25	19	27	47	57
4.75	22	31	52	65
6.5	25	36	65	76
8.5	28	43	74	88
9.5	32	46.5	83	101
12	35	51.5	87	108
13.5	38	57	104	126
17	42	60	115	139
25	50	74	139	168

* According to 8.3.2.3 the shackle pins shall have a tolerance of -0/+3 % on the diameter.

Note:

The rated WLL of shackles of the same size may vary due to the use of materials of different strength. EN 13899 is for shackles of Grade 6, but is it also possible to design shackles according to e.g. EN 1677-4 (for forged lifting components of grade 8) in combination with the standard sizes listed above.

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Since screw pin shackles are not allowed (see 8.3.2.3), the pad eyes must be located such that there is sufficient space to fit shackles with pin, nut and split cotter pins.

D.3 Pad eye strength

According to 4.4.1, the max. concentrated stresses at the bolt hole (geometry effects and contact stress) should be below 2 x minimum specified yield stress, i.e. 2 x Re, at the design load. The method outlined below can be used to determine the minimum size of a pad eye which satisfies this requirement:

edges shall not exceed 2 x Re at design load.

More refined methods can of course be used if an optimised pad eye design is required. In such cases the calculations should be submitted with the approval documentation.

The two following criteria should be fulfilled:

1. Tear-out stress

$$Re \geq \frac{3 \times RSL}{2 \times H \times t - D_H \times t}$$

2. Contact stress

$$Re \geq 23.7 \times \sqrt{\frac{RSL}{D_H \times t}}$$

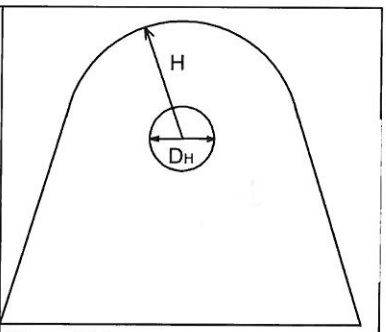

Figure D-1
Padeye

where:

R_e = minimum specified yield strength of the pad eye material in N/mm^2
RSL = resulting sling load in N
H = the shortest distance from centre of bolt hole to edge of pad eye in mm
D_H = bolt hole diameter in mm
t = pad eye thickness in mm

Equation 1 is verifying that the stress level at the edge of the bolt hole is acceptable, assuming a stress concentration factor of 3.

Equation 2 is the formula for peak compressive stresses at the contact line between two concentric cylinders of steel, with a difference in diameter of 6 %.

If fillet welded cheek plates are used to obtain the pad eye thickness required in 3.4.1 (clearance between pad eye and inside of shackle) these should not be taken into account in equation 1. The contact stress criterion may be calculated using the total thickness of pad eye and cheek plates.

Appendix A
Padeye Calculations

A.1 General

Normally the design checks listed below are sufficient to verify a padeye design. However, for special padeye designs additional checks may be necessary, and the need for such checks should hence be evaluated in each case.

Cheek plates may be considered both for tear out and bearing if they are properly welded, see A.5, and their pin hole has the same diameter and is aligned with the main plate hole.

A.2 Definitions

In the equations in this subsection the below listed definitions are applied. Nominal dimensions could be considered.

RSF	Padeye in line design load, see 3.5.4. Note that RSF in *N* shall be used as input in the equations in this appendix.
σ_e	Allowable stress of padeye material in MPa, see 3.4.3.
E	Elastic modulus, i.e. 210 000 MPa for steel
D_{pin}	Diameter of shackle pin (mm)
D_H	Diameter of pinhole (mm)
t	Total thickness of padeye at hole including cheek plates (mm)
a	Weld throat thickness (mm)
R_{pad}	Radius of padeye, taken as: $R_{pad} = \dfrac{R_{pl} \times t_{pl} + 2 \times R_{ch} \times t_{ch}}{t}$

Where:
- R_{pl} is minimum distance form centre hole to edge of plate
- R_{ch} is radius of cheek plates (two equal plates assumed)
- t_{pl} is the thickness of the padeye plate
- t_{ch} is the thickness of the cheek plates

A.3 Bearing pressure

If $D_{pin} \geq 0.94 \times D_H$ the following criterion applies:

$$\sigma_e \geq 0.045 \times \sqrt{\frac{RSF \times E}{D_H \times t}}$$

For smaller pin diameter (i.e. $D_{pin} < 0.94 \times D_H$) the following criterion shall be fulfilled:

$$\sigma_e \geq 0.18 \times \sqrt{\frac{RSF \times \left(\dfrac{1}{D_{pin}} - \dfrac{1}{D_H}\right) \times E}{t}}$$

A.4 Tear out

A tear out check is normally considered sufficient to check the padeye material above (i.e. in the load direction) the hole. The following criterion shall be fulfilled:

$$\sigma_e \geq \frac{2 \times RSF}{\left(2 \times R_{pad} - D_H\right) \times t}$$

A.5 Cheek plate welds

The cheek plate welds should fulfil the following criterion:

$$\sigma_e \geq \frac{RSF \times t_{ch}}{t \times D_{ch} \times a}$$

The above equation is based on the following assumptions: