Bulkhead Failure due to Flooded Holds

GBvdB, ARC, JG, GWH, JL, ABM, NVM,
GO, JRO, MAP, SS, LvW, JPW

1 Statement of the Problem

> *From 1989 the world shipping industry has experienced an increasing number of serious casualties and total losses of large vessels as a consequence of initial hull damage. This represents a new and disturbing trend (...).*[1]

The note from which this was taken continues by describing typical scenarios and stating a number of possible causes. One of them is the collapse of a watertight bulkhead between two holds of which one is flooded. The company WS Atkins brought this problem to the 1997 Study Group in order to investigate the processes of cracking and buckling that are expected to play an important role.

2 General remarks

One of the main obstacles encountered by the Study Group was the relative lack of data. With only general statements about the circumstances that in the past have led to the collapse of bulkheads as input we had to restrict ourselves to indicating a number of mechanisms that might be involved. We hope that more precise information will allow us in the future to be more specific.

We based our investigation on the following scenario: for some reason (collision, for instance) a hold floods with sea water; after some time, the bulkhead gives way, allowing the adjacent hold to flood, and eventually causing the ship to sink. WS Atkins gave us the following information:

[1] Det Norske Veritas report, *Bulk carrier losses*, November 1991

1. Bulkhead failure typically happens at sea, not in harbour; with old ships rather than new ones; with high-density bulk (iron ore) rather than low-density bulk (wheat).
2. Bulkheads are designed to the situation of static loading by water up to the 'tidal' level and without taking into account the force that the bulk may exert on it. The stresses involved might well be near or beyond the yield limit.
3. If the bulkhead collapses, this may well happen within several hours after the flooding of the hold.
4. There is evidence of one case in which the load had not shifted; there is no evidence of shifted loads.

We concentrated throughout on mechanisms that are consistent with these data.

3 Cracking, low-cycle fatigue, and corrosion

At the final stage of the failure, the bulkhead collapses because of large-scale crack propagation. This is consistent with the observation that bulkhead failure is more likely at sea than in harbour, since the cyclic loading due to the wave motion will strongly enhance crack propagation; also the old age of the ships involved makes it likely that large cracks are already present.

Since microcracks are introduced by every manufacturing process, the design of the bulkhead should take their existence into account and minimize the probability of such cracks growing into large-scale cracks. The literature on crack initiation and crack propagation is extensive, and we can not do it justice here. We therefore just pick out a few interesting points.

3.1 Orthotropic plate theory

On a large scale the corrugated bulkhead can be modelled as an anisotropic plate with different characteristics in different directions. The propagation of a crack in a connection between two such plates is governed by the following system of equations (see Figure 1):

$$
\begin{aligned}
L^{(1)}\left(\frac{\partial}{\partial x}\right) u^{(1)}(x) = 0 \qquad & \text{in plate 1} \\
L^{(2)}\left(\frac{\partial}{\partial x}\right) u^{(2)}(x) = 0 \qquad & \text{in plate 2,}
\end{aligned}
$$

with boundary conditions at the crack

$$
\begin{aligned}
\sigma_{i2}^{(1)}(u^{(1)}) &= 0 \\
\sigma_{i2}^{(2)}(u^{(2)}) &= 0,
\end{aligned}
$$

and at the connection

$$
\begin{aligned}
& u^{(1)} = u^{(2)} \\
& \sigma_{i2}^{(1)}(u^{(1)}) = \sigma_{i2}^{(2)}(u^{(2)}) \qquad i = 1, 2.
\end{aligned}
$$

Here the differential operators $L^{(\cdot)}$ are given by

$$
\begin{aligned}
\begin{pmatrix} \sigma_{11} \\ \sigma_{22} \\ \sqrt{2}\sigma_{12} \end{pmatrix} &= A \begin{pmatrix} \epsilon_{11} \\ \epsilon_{22} \\ \sqrt{2}\epsilon_{12} \end{pmatrix} \\
\epsilon_{ij} &= \frac{1}{2}\left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i}\right) \\
D(\xi_1, \xi_2) &= \begin{pmatrix} \xi_1 & 0 \\ 0 & \xi_2 \\ \xi_2/\sqrt{2} & \xi_1/\sqrt{2} \end{pmatrix} \\
L^{(i)}\left(\frac{\partial}{\partial x}\right) &= D_T\left(\frac{\partial}{\partial x_1}, \frac{\partial}{\partial x_2}\right) A^{(i)} D\left(\frac{\partial}{\partial x_1}, \frac{\partial}{\partial x_2}\right)
\end{aligned}
$$

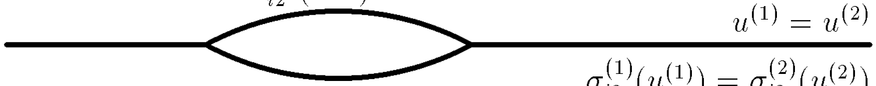
The matrix A characterizes the elastic properties of the two plates.

The speed of propagation of the crack depends on the magnitude of the stress concentration at the tip of the crack. For a given corrugation geometry one can determine the matrix A; it is then possible to find which direction of the corrugation will minimize crack propagation.

$$L^{(1)}\left(\frac{\partial}{\partial x}\right)u^{(1)} = 0$$

$$\sigma_{i2}^{(1)}(u^{(1)}) = 0$$

$$u^{(1)} = u^{(2)}$$



$$\sigma_{i2}^{(1)}(u^{(1)}) = \sigma_{i2}^{(2)}(u^{(2)})$$

$$\sigma_{i2}^{(2)}(u^{(2)}) = 0$$

$$L^{(2)}\left(\frac{\partial}{\partial x}\right)u^{(2)} = 0$$

Figure 1: Equations for an orthotropic plate

3.2 Low-cycle fatigue

The fact that many of the ships have sunk within a couple of hours after the flooding points towards low-cycle fatigue: supposing a cyclic excitation period of $10\,s$ (a typical pitch period) and three hours' delay we find collapse after 1080 cycles.

Low-cycle fatigue is generally associated with plastic deformation, and the number of cycles before failure depends very strongly on the level of plastic strain. This is shown in Figure 2, taken from [2], Figure 10.1. According to this figure, going from 1% to 10% plastic strain implies a decrease in the endurance of two orders of magnitude. Similar results can be found in [4], p. 23.

3.3 Corrosion

Corrosion has a number of adverse effects on the endurance of the structure:

1. The amount of material, and therefore the overall strength, decreases.

2. The chemical composition of the remaining material changes, resulting in a more brittle material ([5], Chapter 12).

3. Corrosion removes material in a non-uniform manner, introducing surface irregularities. For certain kinds of steel, notably those whose *notch sensitivity* is high, such irregularities cause large stress concentrations which enhance crack propagation.

3.4 Cyclic softening and hardening

> The stress-strain behaviour obtained from a monotonic tension or compression test can be quite different from that obtained under cyclic loading. (...) His experiments indicated that the yield strength in tension or compression was reduced after applying a load of the opposite sign that caused inelastic deformation. Thus one single reversal of inelastic strain can change the stress-strain behaviour of metals.[2]

This effect is well illustrated by Figure 3.

A calculation based on static loading of a bulkhead consisting of virgin material will lead to insufficient strength in the light of the three effects mentioned above: the low-cycle fatigue, corrosion, and cyclic softening. In addition, we shall see below that the pitching movement of the ship results in an effective increase of the lateral force on the bulkhead.

4 Forms of dynamic loading

Given the fact that the bulkhead eventually collapses by large-scale cracks, we would like to identify the main external influence responsible for the growth of the cracks. We have already remarked that we should be looking for an intrinsically dynamic phenomenon.

4.1 Varying gravity due to pitching

The vertical movement of the flooded hold due to pitching (the see-sawing movement of a ship) causes an increase in the load in the lower half of the cycle and a decrease in the upper half. Taking an amplitude of $A = 5\ m$ at

[2] [1], p.22

Figure 2: Typical (theoretical) relationship between plastic strain per cycle and endurance.

Figure 3: Stress-strain curves for monotonic and cyclic loading [3].

the bow and a period of $T = 11\ s$, the movement at the bow of the ship is described by

$$z(t) = A \sin \frac{2\pi t}{T}$$

and therefore

$$z''(t) = -\frac{4\pi^2}{T^2} A \sin \frac{2\pi t}{T}.$$

The amplitude of this acceleration is $1.6\ m/s^2$, or 17% of the acceleration of gravity. This variation results in a variation in the (hydrostatic) pressure in the water and therefore in the force exerted by the water on the bulkhead by 17% above and below the value of the static loading.

4.2 Resonance: multi-scale modelling

Although the variation of 17% mentioned above may be relatively small, this excitation can lead to large variations in the load on the bulkhead if the frequency coincides with an eigenfrequency of the system of ship, bulkhead and hold water. We mention here a number of mechanisms that might be responsible for this resonance phenomenon, and we give an estimate for the eigenfrequency involved. It should be kept in mind that the numbers we calculate here are only indications; real-life measurements are needed for accurate estimates, as well as a good understanding of the way the bulkhead is attached to the rest of the ship.

4.2.1 Changing eigenfrequencies

On a large scale, the ship may be considered as a girder and the flooded hold as a high-density perturbation. Such a perturbation may change the eigenfrequencies of the ship and therefore change an initially safe excitation frequency into a resonance frequency. This is consistent with another observation: in circumstances where the flooding was not caused by collision and therefore happened gradually, the flooded hold was detected by sailors who noticed a different ‘feel’ to the ship.

We ran a numerical experiment to determine the influence of replacing part of a homogeneous girder with a higher-density material (Figure 4). For each of the three configurations we calculated the first six eigenfreqencies.

reference	2.1	1.3	0.31	0.20	**0.18**	0.13
(1)	3.7	2.3	0.40	0.29	**0.28**	0.20
(2)	4.0	2.4	0.38	**0.29**	0.25	0.20

Table 1: Oscillation periods in seconds for the first six modes. Bold type indicates the first torsional mode.

The results are set out in Table 1. Note that the first torsional mode moves up in the sequence of modes.

For these calculations we took the values $E = 2 \cdot 10^{11}$, $\nu = 0.3$; the initial density was 500 kg/m^3 and the density of the perturbation 3000 kg/m^3.

4.2.2 The mass-spring system I

On a smaller scale we may consider the system of the bulkhead and the water in the hold. A simple form of localized resonance may appear when the bulkhead functions as an elastic spring and the water in the hold as a dead mass, as in Figure 5. To obtain a first approximation of the eigenfrequency of this system we suppose that the bulkhead is simply supported all around and that the displacement is sinusoidal in the vertical direction, i.e.

$$\vec{u}(x, y, z, t) = \vec{A} \sin \frac{2\pi t}{T} \sin \frac{\pi z}{H}$$

where H is the height of the bulkhead and the depth of the water. $\vec{A}$ is oriented perpendicular to the bulkhead.

A typical way of modelling corrugation is to suppose that there is no strength perpendicular to the corrugation and that therefore each corrugation acts like a beam by itself [6]. Here this beam is simply supported above and below. A beam with the geometry and dimensions shown in Figure 6 has a moment of inertia I given by

$$I = \frac{hd^2}{6}(3b + c) = 0.6 \cdot 10^{-2}\ m^4.$$

If a force of the form

$$q = q_0 \sin \frac{\pi z}{H}$$

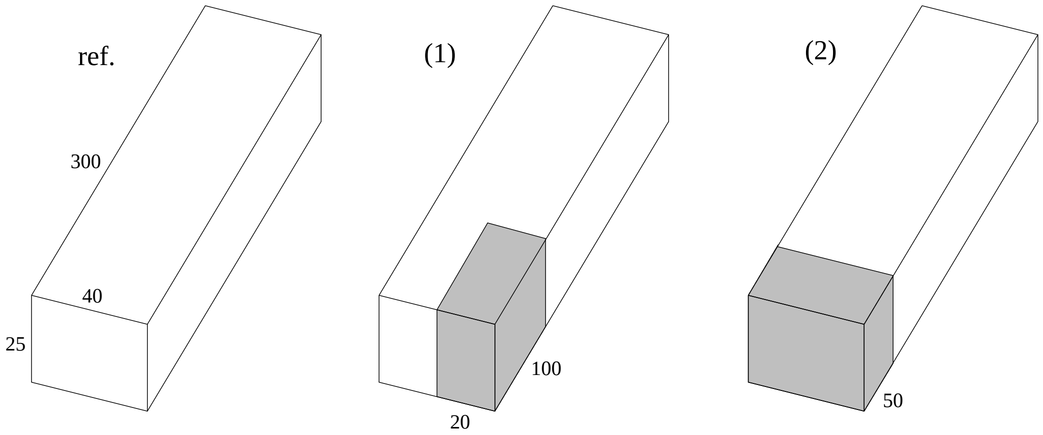


Figure 4: The reference configuration and two perturbations. Dimensions are in meters.

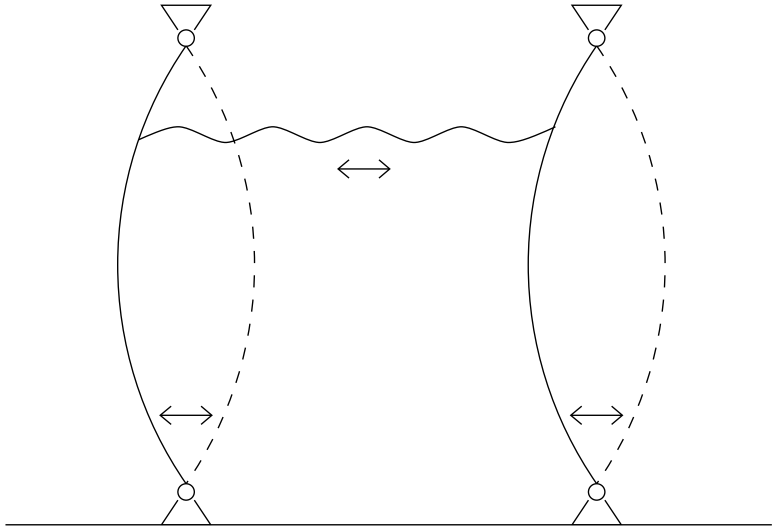


Figure 5: Mode of vibration of the system of bulkhead and water.

is applied to this beam, we find the deflection u by solving

$$EIu''''(z) = q_0 \sin \frac{\pi z}{H},$$

or

$$u(z) = \frac{q_0 H^4}{\pi^4 EI} \sin \frac{\pi z}{H}.$$

If we load the bulkhead with a (marginal) pressure $p = p_0 \sin(\pi z/H)$ then $q_0 = Sp_0$. Therefore the bulkhead responds to an increase in pressure p by a deflection

$$u = k^{-1} p = \frac{SH^4}{\pi^4 EI} p.$$

If W is the width of the bulkhead and L the length of the hold, the total energy of this system is

$$\frac{1}{2} W \int_0^H ku^2 + \frac{1}{2} \rho W L \int_0^H \left(\frac{\partial u}{\partial t} \right)^2 = \frac{1}{4} W H k A^2 \sin^2 \frac{2\pi t}{T} + \rho W L H \frac{\pi^2}{T^2} A^2 \cos^2 \frac{2\pi t}{T}.$$

Since this is supposed constant we have $k = 4\pi^2 \rho L / T^2$ or

$$T = 2\pi \sqrt{\frac{\rho L}{k}}.$$

Using the data $E = 2 \cdot 10^{11}\, N/m^2$, $L = 25\, m$, $\rho = 10^3\, kg/m^3$, $H = 25\, m$, and the geometry of Figure 6, we find

$$T = 2.8\, s.$$

Note that T depends in an important way on the boundary conditions that are applied at the upper and lower ends of the bulkhead: if the simple support conditions are replaced by a clamping at one end, the value of k is divided by 16, and T is multiplied by 4.

4.2.3 The mass-spring system II

Aa slightly different form of oscillation, shown in Figure 7, was proposed during the Study Group; however, this form is probably unstable, and would evolve into an oscillation of the kind described above. The eigenfrequency of this form of oscillation is similar in magnitude to the one above.

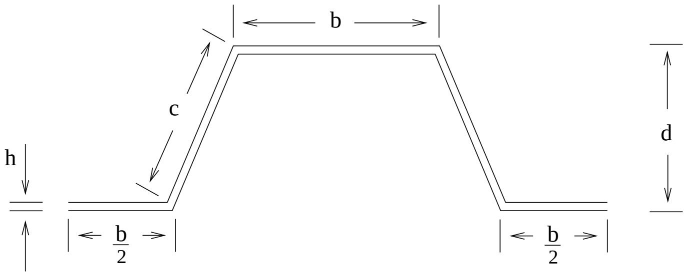


Figure 6: Geometry of the corrugation: $b = 1\ m$, $c = 1\ m$, $d = 0.94\ m$, $h = 10\ mm$, $S = 2.3\ m$.

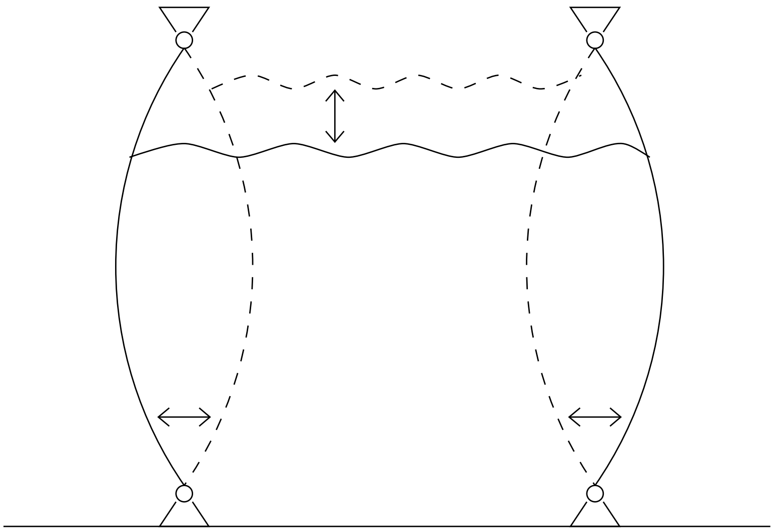


Figure 7: Alternative oscillation of water and bulkhead.

4.2.4 Sloshing

Due to the motion of the ship there will be waves on the surface of the water in the hold. If these waves are large enough this may create an important oscillating force on the bulkhead.

The linearized model problem for waves in a tank is

$$\begin{aligned} &\Delta\phi = 0 && \text{in the tank} \\ &\frac{\partial\phi}{\partial\vec{n}} = 0 && \text{on fixed surfaces} \\ &\frac{\partial^2\phi}{\partial t^2} + g\frac{\partial\phi}{\partial z} = 0 && \text{on the free surface.} \end{aligned}$$

Here ϕ is the velocity potential and g the acceleration of gravity.

Taking a rectangular tank with length L, width W and height H, we find the sequence of solutions

$$\phi_{mn} = e^{i\omega t}\cos\frac{m\pi x}{L}\cos\frac{n\pi y}{W}\cosh kz,$$

with

$$k = \pi\sqrt{\frac{m^2}{L^2} + \frac{n^2}{W^2}} \qquad \text{and} \qquad \omega^2 = gk\tanh(kH).$$

Taking again the values $L = 40\,m$, $W = 40\,m$, $H = 25\,m$, we find the lowest frequency for $(m, n) = (0, 1)$ or $(m, n) = (1, 0)$: then $\omega = 0.86\ s^{-1}$ and the period of this oscillation is $7.3\ s$.

References

[1] H. O. Fuchs & R. I. Stephens, *Metal Fatigue in Engineering*, Wiley (1980).

[2] T. R. Gurney, *Fatigue of Welded Structures*, Cambridge University Press (1968).

[3] R. W. Landgraf, J. Morrow, & T. Endo, Determination of the Cyclic Stress-Strain Curve, *J. Mater.*, Vol. 4, No. 1 (1969), p. 176.

[4] S. T. Rolfe & J. M. Barsom, *Fracture and Fatigue Control in Structures*, Prentice-Hall, Englewood Cliffs, New Jersey (1977).

[5] S. Suresh, *Fatigue of Materials*, Cambridge University Press (1991).

[6] *A Guide for the Analysis of Ship Structures*, Government Research Report PB 181168, prepared for the Ship Structure Committee (1960).