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# IAMU 2017 Research Project

(No. 20170401)

# Marine Structural Failures Database (MarStruFail)

By

University of Rijeka, Faculty of Maritime Studies (UR-FMS)

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# Marine Structural Failures Database (MarStruFail)

## Theme 4: Improving safety by learning from failures of marine structures

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***Abstract*** Marine structures are designed with a requirement to have reasonably long and safe operational life with a risk of catastrophic failures reduced to the minimum. Still, in a constant wish for reduced weight structures that can withstand increased loads, failures occur due to one or several following causes: excessive force and/or temperature induced elastic deformation, yielding, fatigue, corrosion, creep, etc. Therefore, it is important to identify threats affecting the integrity of marine structures. In order to understand the causes of failures, structure's load response, failure process, possible consequences and methods to cope with and prevent failures, probably the most suitable way would be reviewing case studies of common failures. Roughly, marine structural failures can be divided into structural failures of ships, propulsion system failures, offshore structures failures and marine equipment failures. This report provides an overview of such failures taking into account failure mechanisms, tools used for failure analysis and critical review of possible improvements in failure analysis techniques.

***Keyword***: *marine structures, failure analysis, fracture, fatigue, failure.*

## 1. Introduction

Marine structures must comply with such design requirements that the probability of failures or stability loss of parts and/or complete structures is reduced to minimum. Studies and analysis of marine structures failures had shown that a significant percentage of failures were a consequence of inadequate design due to lack of operational considerations, incomplete structural elements evaluations and incorrect use of calculation methods.

Structural safety level is determined during design process by defining specific structural elements, material properties and functional requirements based on the expected lifetime of the structure, ramifications of eventual failures and costs of failures. Time dependency of strength and loads has to be taken into account because the strength of a structure will decrease with time while the load is varying through the lifetime of the structure.

Successful material selection process implies reconciling requirements like suitable strength of material, sufficient level of rigidity, appropriate heat resistance, etc. Structures that are susceptible to crack growth need to be made of materials selected on the basis of fracture mechanics parameters. Fracture mechanics parameters that define material resistance to crack propagation are usually determined through experimental research, but nowadays some of the experiments can be successfully substituted with numerical analysis. Material fracture behavior is usually estimated using some of the well-established fracture parameters, like stress intensity factor (*K*), *J*-integral or crack tip opening displacement (CTOD). Besides that, fatigue limit has to be taken into account, also. It has become customary to perform an optimal fatigue design analysis as an integral part of design calculations. Such analyses are also largely based on data and procedures developed from experimental and empirical research.

Marine structural failures can be divided into three main groups: failures of ships, offshore structures and marine equipment. This report will provide an overview of most common case studies of such failures. Further, failure mechanisms will be emphasized and tools used for failure analysis outlined. Possible improvements in failure analysis techniques are discussed in the end of the report. Database of case studies is available at URL address: https://www.pfri.uniri.hr/web/en/projekti/aktivni/2017_-_MarStruFail_-1_eng.pdf.

## 2. Ship Structural Failures

Maybe the most notable case of ship failures are failures of Liberty ships in the early 1940's. These failures gave a serious boost in the development of fracture mechanics. Ships, mass produced in assembly-line style out of prefabricated sections as an all-welded construction, exhibited nearly 1500 cases of brittle fractures with 12 ships breaking in half. The results of failure investigation had shown that inadequate grade of steel allowed for brittle fracture at low temperatures. Further, rectangular hull openings such as hatch square corners that coincided with a welded seam acted as stress concentrations points and crack origins [1].

There has been a considerable amount of failures in recent times, also. For instance, structural failure of container ship MOL Comfort [2, 3] in 2013. A yearlong failure investigation concentrated on finding the possibility of fracture occurrence and structural safety level. Results had shown that the hull fracture originated from the bottom butt joint in the midship part. A possibility that the load's upper limit exceeded strength's lower limit was also estimated using probabilistic approach. Furthermore, safety inspections of the MOL Comfort sister ships have shown buckling deformations (concave and convex) of the bottom shell plating of up to 20 mm (4 mm allowable) in height observed near the center line. Finally, a numerical analysis of the ship hull taking the load history into account was done. After the investigation it was concluded that the load of the vertical bending moment probably exceeded the hull girder ultimate strength when the deviations of the uncertainty factors are taken into account, which caused the bottom shell plates to buckle due to excessive load. The reduction of breadth of bottom shell plate between girders increased the stress in the girder which yielded in the lower part resulting in the collapse occurs in the middle part of the ship, at the bottom, near the center line.

Bilge keels structures are used to enhance the transverse stability of ships. Cracks have been noticed in various ships in the internal structure of the bilge keels and on the connecting points to the ship's hull. Failure analysis of the damage can identify the causes of failure and the analysis results serve as basis for design improvements. It has been shown, both theoretically and applying FEM analysis, that the failure locations in bilge keels structures occur in the stress concentration regions that are present due to the structure geometry commonly used, therefore new structural elements are proposed that significantly reduce the possibility of failure occurrence [4].

Corrosively aggressive cargo (acids, alkalis etc.) can represent a danger to the integrity of ship structures. In the case of the "Stolt Rotterdam" freighter which sank during the cargo loading in the port the investigation (visual, macro-fractographic and chemical) following the sinking has shown that the residue valve has cracked due to a design-specific stress (stiffer main valve was missing), thus causing a leak of the acid that accelerated the corrosion process of the floor panels in the area of the

leak. Also, the valve gaskets were made of a material not resistant to acid which also contributed to the speed of the leak [5].
Marine engines and propellers produce dynamic loads on their supportive structures which can lead to fatigue failures. One of the most stressed components of the engine structure is the bearing bushing foundation. A state-of-the art design procedure for the bearing girders is comprised of essential procedures such as bearing loads determination, stresses calculation and the bearing girder fatigue strength assessment [6]. The fatigue and structural durability analysis is conducted for multi-axial stresses and opens the possibility to construct lightweight engines.

### *2.1 Case Study of MV Kurdistan*

Table 1. Data regarding failure of MV Kurdistan

| Technical data/general information | |
|---|---|
| **Structure type:** | All welded tanker built to construction category Ice Class I (Lloyds +100Al Oil Tanker Ice class 1+LMC) |
| **Material:** | Steel |
| **Fate:** | Loss of ship by intentional sinking |
| **Date of accident:** | March 15$^{th}$ 1979 |
| **Failure description** | |
| **Failure mode:** | |
| Brittle fracture, the ship broke in two, the bow rose, hinging about the deck at the No.3 cargo tanks before finally separating from the stern. | |
| **Failure cause:** | |
| Presence of defect in bilge keel welds combined with high thermal stresses. | |
| **Load type/conditions:** | |
| Moderately high seas, air temperature near 0°C, cargo temperature approximately 60°C. | |
| **Analysis data** | |
| **Failure analysis tools and methods used:** | |
| Elastic-plastic fracture mechanics<br>Visual crack inspection<br>Pellini (drop weight test) NDT<br>crack tip opening displacement (CTOD) tests<br>fracture mechanics calculations performed using PD6493 (1980) procedures | |
| **Crack initiation:** | |
| The initial fracture through the bottom and side shell plates was brittle<br>The origin of the crack was a defective butt weld in the port bilge keel | |
| **Crack propagation:** | |
| The inquiry into the failure of the Kurdistan did not establish precisely the sequence of failure of the ship's longitudinal structure, which showed both brittle and ductile fracture. | |

| Analysis results and conclusions |
| --- |
| The fracture occurred forward of the wash bulkheads in No. 3 tank. The failure of the bottom shell plate occurred as a clean break with little or no deformation. Significant deformation was present on the ship's plate on both sides in the region 20-30 ft (6-9 m) below the deck plate. The failure appeared to be macroscopically brittle, showing signs of little or no ductility.<br>The site with the most significant damage was the port bilge keel. There was no evidence of the crack having arrested at any point along the bottom shell. Visual inspection had shown that the crack initiation occurred from the fatigue-cracked areas situated in the ground bar weld metal, eventually progressing into the bulb and the shell due to inadequate dynamic toughness of the fillet welds, due to the low sea water temperature (-1°C), connecting the mentioned sections of the ship.<br>The subsequent breaking of the ship in two was inevitable due to the extensive structural damage caused by the fatigue crack propagation.<br>The fracture mechanics calculations performed had shown that the combination of the position of the bilge keel defect under the still water bending moment loading, the influence of the thermal stresses caused by carrying a hot cargo in cold waters, the effect of high tensile residual stresses and the wave loading on exiting the ice field caused the bilge keel defect to grow into high level displacements.<br>Thermal stresses caused by the temperature difference of the cargo and the see resulted in high tensile stresses in the shell and the bilge keel. The additional wave load stresses combined with the thermal stresses triggered the fracture of the Kurdistan's bilge keel. The mechanical properties of the shell material were not sufficient to counter the propagation of the crack, thus resulting in complete failure<br>The initiation of the fracture was due to the classic combination of poor weld metal toughness and high stresses in the presence of a defect. |
| **References** |
| 1. Garwood, S.J., Investigation of the MV Kurdistan casualty, Engineering Failure Analysis, Vol. 4, No. I, pp. 3 24, 1997<br>2. TWI Report 632/1998, Catastrophic Failures of Steel Structures in Industry: Case Histories, B Hayes and R Phaal, February 1998 |
| **Summary** |
| The MV Kurdistan suffered a catastrophic brittle fracture initiating in the port bilge keel weld, which propagated into the ship's structure, causing the vessel to break in two. Despite all the materials tested met the required standards, the inadequately done weld in the ground bar of the port bilge keel induced a large weld defect, thus reducing the local toughness. This weld defect was subject of fatigue damage, increasing the local notch acuity, finally resulting in a brittle fracture as the vessel encountered "head on" seas on emerging from an ice field.<br>The combination of still-water bending moment, thermal stresses, wave loading, residual stresses from welding, defect size, and low toughness made brittle fracture initiation inevitable.<br>The combination of events leading to the Kurdistan encountering the ice field, and the characteristics of its bunker oil cargo, reduced the temperature of the ship's plate to the external water temperature (-1°C) despite carrying a hot cargo. This resulted in the catastrophic propagation of the brittle fracture from the bilge keel initiation site as the vessel emerged from the ice field, resulting in the eventual complete fracture of the vessel. |
| **Legacy/Lessons learned** |

This casualty illustrates the importance that secondary stresses and thermal stresses can have on the conditions that lead to failure.
The investigation introduced the use of elastic-plastic fracture mechanics in formal investigation conclusions presentations in a UK court.
This failure showed important failings of the requirements for ships of the size of the Kurdistan built as First Year Ice Class vessels:

- the ship could be built entirely of Class A steel with no notch impact requirements
- no calculation of thermal stresses was required for cargoes at temperatures below 65°C.

Additionally, this failure showed how critical the quality of workmanship could be even for a detail of apparently little significance such as the bilge keel.

**Figures**

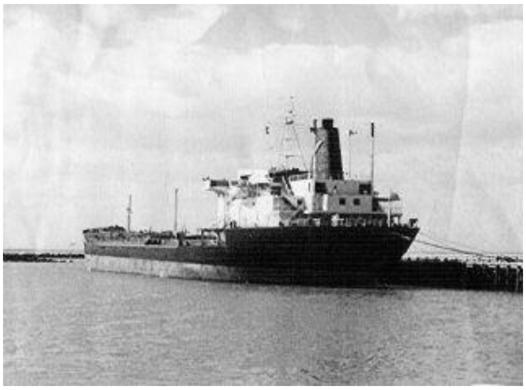


Fig. 1. MV Kurdistan [2.1]


Fig. 2. Sinking of MV Kurdistan [2.1]


Fig.3 Extent of damage [2.1]

| Further reading |
|---|
| 1. https://www.twi-global.com/news-events/case-studies/m-v-kurdistan-tanker-141/<br>2. https://www.wrecksite.eu/wreck.aspx?34872 |

## *2.2 Case Study of MOL Comfort*

Table 2. Data regarding failure of MOL Comfort

| Technical data/general information | |
|---|---|
| **Structure type:** | 8000 TEU class large container ship, 316 m length |
| **Material:** | Steel |
| **Fate:** | Broke in two. Stern section sank on 27th June and bow section on 11 July. |
| **Date of accident:** | June 17th 2013 |
| **Failure description** | |
| **Failure mode:** | |
| Crack amidships in bad weather | |
| **Failure cause:** | |
| Bottom shell plates experienced plastic deformation in the transverse direction just before the ship reached the maximum load of the longitudinal hull girders | |
| **Load type/conditions:** | |
| Significant wave height of 5.5 m with a mean wave period of 10.3 s, encountered wave direction of 114° | |
| **Analysis data** | |
| **Failure analysis tools and methods used:** | |
| Numerical simulation<br>3-hold model elasto-plastic analyses<br>Probabilistic load estimation<br>On-board full scale measurements on sister ships | |
| **Crack initiation:** | |
| Mid-ship bottom shell plates buckling | |
| **Crack propagation:** | |
| Subsequent hull girder fracture | |
| **Analysis results and conclusions** | |
| The analysis results have shown that the container loads are relatively smaller than the bottom sea pressure in general as the lateral loads. The main loads always acting on the double bottom structure of container ships are as follows:<br>• compressive loads in longitudinal direction due to vertical bending moment in hogging condition, | |

- lateral loads in upward direction due to bottom sea pressure,
- compressive loads in transverse direction due to side sea pressure.

The compressive loads due to vertical bending moment causes longitudinal compressive stress and the compressive loads due to side sea pressure causes transverse compressive stress respectively on the bottom shell plates.
The above-mentioned stresses superimpose one to the other resulting in an always-compressive condition both in the longitudinal and transverse directions in the middle part of the double bottom structure. In other words, the stiffened bottom panels are subjected to a multiaxial compressive stress composed of compressive stress in the longitudinal direction due to vertical bending moment, compressive stress in the transverse direction due to side sea pressure and double bottom local stresses due to the lateral loads both in the longitudinal and transverse directions.
In conclusion, the mechanism of the buckling collapse of the bottom shell plates to the hull girder fracture can be described as follows: "the upward loads of bottom sea pressure are dominant among the lateral loads acting on the double bottom structure of container ships. The lateral loads are mainly supported by I beams with flanges of bottom shell plates and inner bottom plates and with webs of girders and floors. Once bottom shell plates are locally buckled and collapsed with plastic deformations, the effective breadth of the flange of bottom shell plates attached to the girder is reduced. The reduction of the effective breadth of bottom shell plate flange increases the compressive bending stress of the girder caused by the lateral loads. As the result of the superimposing with vertical bending stress of compression, the lower half of the girder partly yields.
Bending strength of double bottom structure against the lateral loads is reduced due to the local buckling collapse of bottom shell plates and due to the partial yielding of adjacent girders, which causes the subsequent propagation of the buckling collapse of bottom shell plates and the yielding of the girders leading to the hull girder fracture finally.
The buckling collapse of the bottom shell plates which might trigger the above phenomenon generally occurs in the middle part of the hold around one floor space before or after the partial bulkhead in the longitudinal direction of the ship and near the centre line of the ship, mainly in the stiffened bottom panel adjacent to the keel plate in the transverse direction of the ship. In both cases, compressive local stress of the bottom shell plates is relatively high.

**References**


1. https://en.wikipedia.org/wiki/MOL_Comfort
2. http://gcaptain.com/mol-comfort-incident-photos/
3. ClassNK Investigation Report on Structural Safety of Large Container Ships, September 2014
4. https://www.rina.org.uk/mol_comfort_accident.html


**Summary**

Results of the investigation had shown that the hull fracture originated from the bottom butt joint in the mid-ship part. A possibility that the load's upper limit exceeded strength's lower limit was also estimated using probabilistic approach. Furthermore, safety inspections of the MOL Comfort sister ships have shown buckling deformations (concave and convex) of the bottom shell plating of up to 20 mm (4 mm allowable) in height observed near the centre line.
Finally, a numerical analysis of the ship hull considering the load history was done. The investigation concluded that the load of the vertical bending moment probably exceeded the hull girder ultimate strength when the deviations of the uncertainty factors are taken into account, which caused the bottom shell plates to buckle due to excessive load. The reduction of breadth of bottom shell plate between girders increased the stress in the girder which yielded in the lower

part resulting in the collapse occurs in the middle part of the ship, at the bottom, near the centre line.

**Legacy/Lessons learned**

- The local strength of the double bottom structure, i.e. the transverse strength, against lateral loads such as bottom sea pressure and container loads is closely related to the hull girder ultimate strength through the buckling collapse of bottom shell plates.
- Double bottom structure of a container ship is always subjected to upward loads of the bottom sea pressure. Under this condition, there is a possibility that local buckling collapse of bottom shell plates causes reduction in the strength of double bottom structure and it leads to the hull girder fracture due to superimposition of the vertical bending moment.
- Hull structural strength can be adequately assessed relating to the hull girder fracture accident when the hull girder ultimate strength is evaluated in consideration of the effects of lateral loads.

**Figures**

Fig. 1. APL Poland, identical sister ship of MOL Comfort [3.1]

Fig. 2. Damage extent


Fig.3 Damage extent (detail)

**Further reading**

1. http://www.mlit.go.jp/common/001029660.pdf
2. http://gcaptain.com/mol-comfort-investigation-report-released/
3. https://www.rina.org.uk/mol_comfort_accident.html

### *2.3 Case Study of Algowood*

Table 3. Data regarding failure of Algowood

| **Technical data/general information** | |
|---|---|
| **Structure type:** | Great Lakes bulk carrier, self-unloader, five cargo holds |
| **Material:** | Lloyd's Grade A Steel |
| **Fate:** | Flooded, sat on the bottom, later salvaged and repaired, still active |
| **Date of accident:** | June 1st 2000 |
| **Failure description** | |
| **Failure mode:** | |
| Structure buckling | |
| **Failure cause:** | |
| Inadequate loading and de-ballasting procedures and miscommunication caused excessive bending stresses | |
| **Load type/conditions:** | |
| Aggregates and manufactured sand | |
| **Analysis data** | |
| **Failure analysis tools and methods used:** | |
| Ultrasonic material thickness measurements<br>Chemical and mechanical characteristics analysis | |

| |
|---|
| **Crack initiation:** |
| Hogging/bending moment about 2.3 times the maximum permissible (sea going) |
| **Crack propagation:** |
| Loss of structural strength |

**Analysis results and conclusions**

The Algowood experienced a sudden, major structural hull failure, in the form of extensive structural buckling and distortion on the deck above the water line in the cargo holds 3 and 4 part of the ship. The inspection in dry dock revealed deformations, distortions, and localised fractures in the forward and after sections of the bottom shell plating. These were caused by the vessel contacting and settling on the bottom during the event.
The ultrasonic material thickness measurements, conducted in dry dock, showed wastage of 1 to 7% in the shell, bilge, keel, and bottom structural members, none of which exceeded accepted limits at which replacement of the material would be required.
Furthermore, chemical and mechanical characteristics examination of the material showed no abnormalities that would negatively affect weldability. The material conformed to Lloyds Grade A steel.
Still water bending moment calculations have been performed after the accident, showing that, immediately before hull failure, the vessel was subjected to a hogging/bending moment about 2.3 times the maximum permissible (sea going) moment. This kind of bending moment puts the main deck plating in tension and the bottom structure in compression. The hogging condition was due to the excess of weight over buoyant support at the ends of the vessel.
The investigation had concluded that:

- The intended loading and de-ballasting sequence was violated and the vessel was subjected to excessive bending stress, which resulted in structural failure of the hull. The disposition of the cargo and ballast at the time of the failure caused a still water bending moment about 2.3 times the maximum permissible.
- A lack of feedback communication between the port personnel as well as the inadequate frequency of draught marks reading during loading were noticed

The magnitude of stresses that occurred due to inadequate loading sequence remained unnoticed and unappreciated by shipboard personnel, as the ship's approved loading manual on board the vessel contained representative loading conditions but does not outline loading and de-ballasting sequences.

**References**

1. http://www.boatnerd.com/pictures/fleet/algowood.htm
2. Transportation Safety Board of Canada (TSB), Marine investigation report M00C0026, structural failure bulk carrier ALGOWOOD, Bruce Mines, Ontario, 2000

**Summary**

The bulk carrier Algowood experienced a sudden major structural failure due to inadequate loading and de-ballasting procedures. The investigation of the occurrence did not show any material and structural inadequacies nor any kind of uncharted obstructions, boulders, or other features that could have contributed to the initiation of the hull failure. The accident occurred due to inadequate loading sequence causing the appearance of stresses that exceeded nominal values, hence causing a fracture in the hull allowing water to flood the ship.

**Legacy/Lessons learned**

- Cargo handling policy modified in order to include procedures that require all split loading and unloading revision by the company to determine if the proposed load/unload falls within the allowable limits set for various vessels with respect to stress and shear forces.
- Personnel additional education regarding stresses and strain during cargo handling operations
- Stricter control of loading procedures needed
- The importance of loading distribution on local high stress occurrence
- The importance of adherence to loading manuals and loading plans

**Figures**

Fig. 1. Self-Discharging Bulk Carrier ALGOWOOD

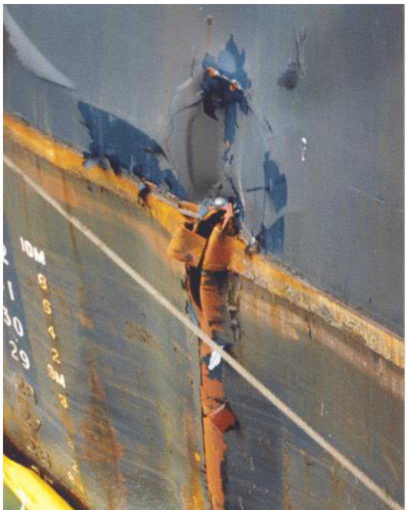


Fig. 2. Damage detail

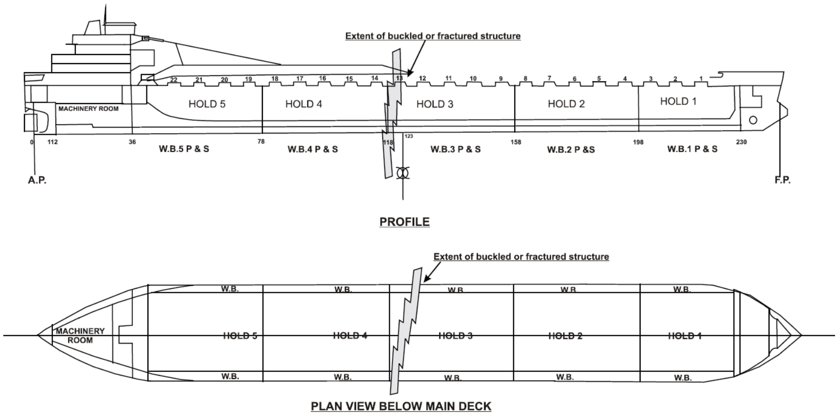



Fig.3 Damage location

**Further reading**

1. https://www.maritime-executive.com/blog/five-common-causes-of-crane-failure#gs.WnQ1WO8
2. http://www.tsb.gc.ca/eng/rapports-reports/marine/2000/m00c0026/m00c0026.asp#Photo_2

## 3. Propulsion System Failures

The propulsion system has a pivotal role on ships. A typical marine propulsion system is comprised of main engine, driving device, marine shaft and propeller. Most of the failures occur on the propulsion shaft and bearings that is subjected to various types of loading during operation (torque moment, bending moment, axial thrust force and transversal loads). The operating environment of the propulsion system is characterized by significant changes in temperatures and humidity, aggressive atmosphere, long lasting interrupted operating time and variations in load amplitudes. The risk of failures of the propulsion system additionally increases with the severity of sea and weather conditions as they have a direct effect on the dynamics of the load variation. All of the above has direct influence on fatigue behavior and life time of the power transmission system.

Shaft keys are recognized as a potential origin of growing cracks. The geometry of the ends of keyways represents a stress concentration factor in the cases of torque transmission through shaft keys for dynamic vibrational loads. Faulty machining of shaft key elements (key groove, keyway and key) geometry, inadequate run out radii or material imperfection can be root causes of torsional fatigue failure in shaft keys. The characteristic torsional failure indicator is the crack pattern that initiates at the end of the keyway and propagates in a 45° rotational direction in a helical shape. Also, interaction between engine body and hull must be taken into account, especially thermal loads that can affect the integrity of shafts and can be successfully solved numerically [7].

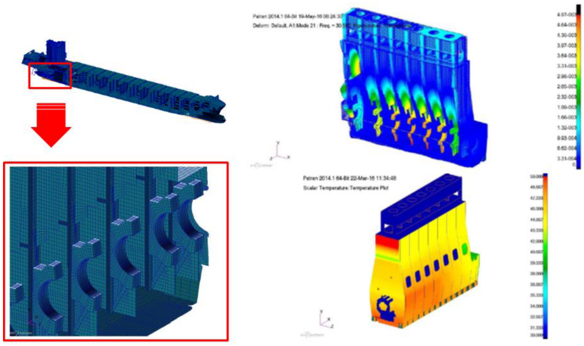


Fig. 3.1. Numerical model of the engine body-ship hull interaction and thermal loads presenting a threat to structural integrity [7]

A case study [8] have shown that inadequate torsional vibration calculation parameters (shaft elements stiffness and damping, natural frequencies, safety factors) and a subsequent poor design of the shaft's keyway cause failures. In this case a root cause analysis was done by the analytical stress calculation process MIL G 17859D and VDI 3822 standards. A FEM model was used in order to verify the existing fracture characteristics and causes.
The alternative to shaft key joints are spline joints which are press fitted to other shaft elements. Analysis of spline joint failure [9] shows that the press fitting of the joining elements can cause surface deformation which in turn causes surface cracks formation. Cracks usually start on the spline teeth at the shaft junction zone. Torsional fatigue caused by fluctuating stress promotes crack growth and propagation. Inhomogeneity of the shaft material can additionally assist crack propagation. In this case, visual and macroscopic inspection was performed, followed by material chemical analysis, hardness measurement, optical and scanning electron microscope (SEM) microstructure analysis with X-ray dispersive analysis of particles under the SEM.
Bolted connections are used in collar coupling of shaft elements and in propeller blades connections. The changes of rotation direction of the shaft results in torque moment overloading and direction change as well as thrust force direction change. The resulting effect is a dynamic load on collar coupling bolts in a longer operating time [10] which can result in fatigue failure. The fretting that occurs on adjacent connecting surfaces in these cases creates micro notches that develop into fatigue cracks with the direction of failure growth in planes angled from 35° to 60° which is not characteristic of pure torsional fatigue failures. The analysis showed that the coupling bolts are subjected to an increasing bending moment which contributes to fatigue crack growth. The experimental research and numerical calculation done in this case study proved the hypothesis of variable bending stress in the coupling as the failure cause. Bolted connections of propeller blades and the shaft are often in a cathodic protection environment. Hydrogen inclusions in the material and variable stress conditions can cause crack nucleation and propagation, finally causing a failure [11]. Fractographic analysis, chemical analysis, micro hardness tests, slow strain rate test, microstructure analysis and finite element analysis was performed in this case.
Abnormal performance of the propeller by way of one non-performing malformed blade can generate a uniaxial force which fluctuates once per rotation in a consistent transverse direction across the shaft. The fluctuating force generates a couple which can cause fatigue failure of the propeller hub [12]. Uniaxial type of failure is characterized by a fatigue fracture with a single origination point that progresses across the shaft from the side where the force is being applied and results in the final overload failure occurring on the opposite side from the fluctuating force. Visual inspection, detail axis alignment measurements, microscopic metallurgical examination, hardness measurements and ultrasonic scanning were used during the analysis.

### *3.1 Case Study of Ship Engine Crankshaft Failure*

Table 4. Data regarding Ship Engine Crankshaft Failure

| **Technical data/general information** | |
|---|---|
| **Structure type:** | Engine crankshaft |
| **Material:** | Steel |
| **Fate:** | Fatigue failure |
| **Date of accident:** | - |
| **Failure description** | |
| **Failure mode:** | |
| Bending-torsional fatigue crack | |
| **Failure cause:** | |
| Material imperfections<br>Fatigue stresses | |
| **Load type/conditions:** | |
| combination of cyclic bending and steady torsion | |
| **Analysis data** | |
| **Failure analysis tools and methods used:** | |
| Microscopy (eye seen) observation<br>Linear elastic fracture mechanics<br>Micro-fractography | |
| **Crack initiation:** | |
| On the fillet of the crankpin, starting as three short parallel cracks nucleated by rotary bending. | |
| **Crack propagation:** | |
| From the web crankpin to the main journal, with a typical helical surface due to the effect of torsion. | |
| **Analysis results and conclusions** | |
| Crankshaft are loaded with a combination of cyclic bending and steady torsion due to dynamic variations of load conditions of the engine. After a certain amount of working hours, fatigue effects become important. A particular case of a crankshaft that failed after over 32834 h in service, and has broken on one of the web crankpins, in the transition to the main journal is used as a typical example.<br>During visual inspection, a crack in the middle of the crankshaft was found. The fatigue crack surface morphology lead to the conclusion that the fatigue crack initiation was caused by rotating bending stresses and the crack propagated by rotating bending combined with torsional stresses. Lines in the crack surface, known as benchmarks, were noticed. These lines correspond to the engine stopping or changes of loading in service and are helpful to calculate the number of | |

cycles.
Micro-fractography revealed no inclusions, pre-cracks, or other abnormal stress raisers.
Fracture mechanics approach was used in order to determine the viability of a fatigue fracture.
The two distinct surfaces on the fracture (one smooth and the other in a horizontal plane of the crankshaft), the records in the main engine book on board and the examination of the local microstructure close to the crack initiation zone showed that there were no inclusion, flaw or a latent defect in the material that could have caused the failure. Fatigue then remains as a culprit of the failure.

**References**

1. M. Fonte, M. de Freitas, Marine main engine crankshaft failure analysis: A case study, Engineering Failure Analysis 16 (2009) pp1940–1947, Engineering Failure Analysis (accepted manuscript)

**Summary**

During the investigation of a crankshaft failure, a microscopy (eye seen) observation has been carried out showing that the crack initiated on the fillet of the crankpin by rotary bending and the propagation was a combination of cyclic bending and steady torsion.
The fatigue fracture appears in two distinct surfaces: a smooth almost to perpendicular to the crankshaft and a second one in a horizontal plane with the crankshaft, with transition zones between two surfaces. Further analysis has excluded any material defect as possible causes of the failure, so the catastrophic fracture of this marine crankshaft was by fatigue, as a combination of rotating bending with steady torsion

**Legacy/Lessons learned**

- Fast crack propagation indicates relatively high bending stress levels
- After the crack initiation by rotating bending, the effect of steady torsion becomes significant

**Figures**

Fig. 1. Typical ship engine crankshaft

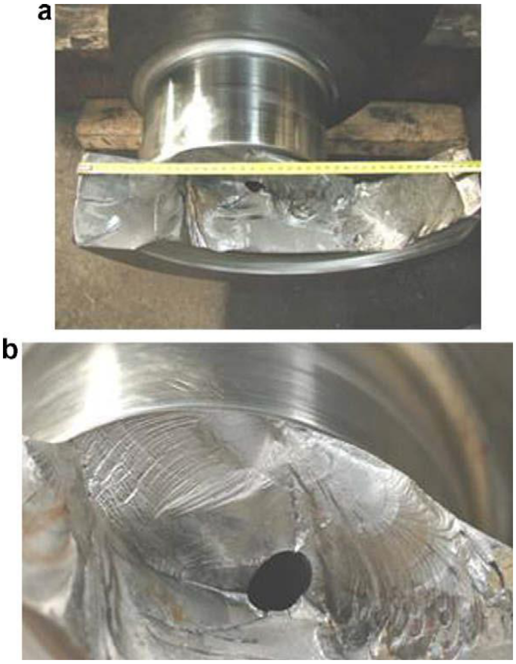


Fig. 2. Fracture details

**Further reading**

1. M. Fonte, P. Duarte, V. Anes, M. Freitas, L. Reis, On the assessment of fatigue life of marine diesel engine crankshafts

### *3.2 Case Study of Propulsion Shaft Failure*

Table 5. Data regarding Propulsion Shaft Failure

| **Technical data/general information** | |
|---|---|
| **Structure type:** | Ship propulsion shaft |
| **Material:** | Steel |
| **Fate:** | Fatigue failure |
| **Date of accident:** | - |
| **Failure description** | |
| **Failure mode:** | |
| Fatigue failure due to torsional-bending loads | |
| **Failure cause:** | |
| wear, corrosion effects, material imperfections, poor material quality, overloads, stress concentration and impact loads, shaft misalignment | |
| **Load type/conditions:** | |
| torque moment, bending moment, axial thrust force and transversal loads (gravitational and centrifugal forces) | |
| **Analysis data** | |
| **Failure analysis tools and methods used:** | |
| S-N based methodology for fatigue life assessment<br>Root Cause Analysis (RCA)<br>Fault Tree Analysis (FTA) method<br>Chemical composition analysis<br>Micro-structural characterization<br>Fractography<br>Hardness measurements<br>Finite element simulation | |
| **Crack initiation:** | |
| Ends of keyways (stress concentration factor)<br>Filets, tapers and chamfers in the shaft geometry<br>Shaft spline joints<br>Bolted connections<br>Propeller hub | |
| **Crack propagation:** | |
| 45° rotational direction in a helical shape | |
| **Analysis results and conclusions** | |
| The design procedure and calculation must be compliant to classification society's rules. The main idea is to make a real marine propulsion system that can enable an efficient, reliable, safe, | |

durable and low cost performance throughout its entire life cycle.

The geometry of the ends of keyways represents a stress concentration factor in the cases of torque transmission through shaft keys for dynamic vibrational loads. Faulty machining of shaft key elements (key groove, keyway and key) geometry, inadequate run out radii or material imperfection can be root causes of torsional fatigue failure in shaft keys. The characteristic torsional failure indicator is the crack pattern that initiates at the end of the keyway and propagates in a 45° rotational direction in a helical shape.

Filets, tapers and chamfers in the shaft geometry also represent geometrical stress concentrations. Inadequate design of these elements can lead to fatigue failure due to cyclic torsional-bending load, with a crack that originates in multiple points on fillet shoulders on the shaft, gradually reducing the load bearing area of the shaft as it grows, and finally resulting in a sudden failure during overload.

Analysis of spline joint failure shows that the press fitting of the joining elements can cause surface deformation, which in turn causes surface cracks formation. Cracks usually start on the spline teeth at the shaft junction zone.

The changes of rotation direction of the shaft results in torque moment overloading and direction change as well as thrust force direction change. The resulting effect is a dynamic load on collar coupling bolts in a longer operating time, which can result in fatigue failure.

Abnormal performance of the propeller by way of one non-performing malformed blade can generate a uniaxial force, which fluctuates once per rotation in a consistent transverse direction across the shaft. The fluctuating force generates a couple which can cause fatigue failure of the propeller hub.

**References**

1. Hyung Suk Han, Kyung Hyun Lee, Sung Ho Park, Estimate of the fatigue life of the propulsion shaft from torsional vibration measurement and the linear damage summation law ins hips, Ocean Engineering107(2015)212–221

2. Hyung Suk Han, Kyoung Hyun Lee, Sung Ho Park, Parametric Study to Identify the Cause of High Torsional Vibration of the Propulsion Shaft in the Ship, Engineering Failure Analysis, Volume 59, January 2016, Pages 334-346

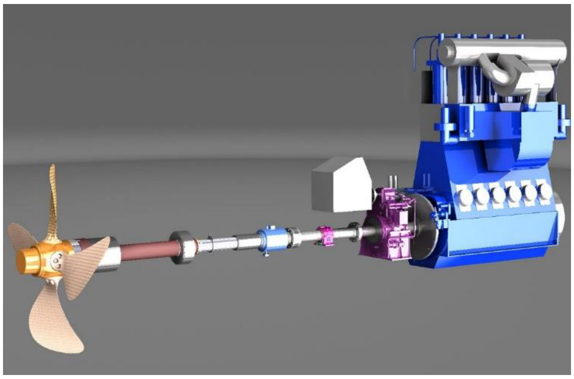
3. Goran Vizentin, Goran Vukelić, Mateo Srok, Common failures of ship propulsion shafts, Pomorstvo, 2017, 3

**Summary**

Constant load variation changes resulting in fluctuating torsional vibrations coupled with geometrical high stress concertation areas have been identified as main causes of fatigue failure of propulsion shafts. As poorly designed geometric shapes of specific shafting elements connections are shown to be the staring points of fatigue crack formation, special attention must be given to their dimensioning during design. Constant monitoring, measurement and data collection of fatigue indicators and indicative events that have influence on fatigue development is very important in order to form a knowledge base that can serve as basis for current design and maintenance procedures improvement.

**Legacy/Lessons learned**

- The importance of geometry details in the design phase

| Figures |
| --- |
|  Fig. 1. Representative propulsion system  Fig. 2. Shaft crack detail |
| **Further reading** |
| |

### *3.3 Case Study of Stern Tube Bearing Failure*

Table 6. Data regarding stern tube bearing failure

| Technical data/general information | |
| --- | --- |
| **Structure type:** | Stern tube bearing of the power transmission system |
| **Material:** | Steel |
| **Fate:** | Seizing during bad weather |
| **Date of accident:** | January 2013 |

| Failure description |
| --- |
| **Failure mode:** |
| Stuck of the propeller shaft |
| **Failure cause:** |
| Flawed shaft line alignment leading to fatigue of aft bearing |
| **Load type/conditions:** |
| High seas, overload of the bearing by propeller with lack of proper lubrication |
| **Analysis data** |
| **Failure analysis tools and methods used:** |
| Visual inspection and examination<br>Laser alignment measurements<br>Crankshaft springing test<br>Linear elastic fracture mechanics |
| **Crack initiation:** |
| Stress concentration caused by stern tube bearing overload |
| **Crack propagation:** |
| From aft to fore edge of stern tube bearing |
| **Analysis results and conclusions** |
| Slow-speed main engine connected directly by shaft line (intermediate shafts and propeller shaft) with propeller is typical for merchant ships. In that propulsion system there is no gears or flexible couplings. Power transmission system (crankshaft plus shaft line) is loaded by strongly unsymmetrical perpendicular forces. Especially stern tube bearing is loaded from one side by very heavy propeller. What is more, shaft line's rotational speed is very low. Therefore, stern tube bearing has to be relatively long. It is one of the main reasons for the necessity of shafting alignment. Shaft line alignment is performed and checked (by measurements) usually only during shipbuilding process. It is not monitored during ship exploitation. Shaft lines' improper operational parameters can be checked only indirectly, e.g. by bearings oil film temperature. Shaft line alignment can be dangerously changed under the influence of excessive operational loads, random events (ship grounding), and repairing process of propulsion system or ship hull in the engine room area.<br>The vessel has been docked in March 2004; several coupling bolts (between intermediate and propeller shaft) have been found stuck. In July 2007 the damage of the propeller shaft arrangement occurred and emergency repaired has been implemented. In January 2013, during bad weather, high temperature alarm occurred in the stern tube bearing. ME has been stopped; in the oil found water. Bad weather forced to use ME with minimum rpm for three days to protect ship.<br>Imperfect shaft line alignment (stuck coupling bolts) with bad weather (overloading caused by resurfacing propeller) was leading to shaft's seizing and damage of the lubricating system. Further, forced work of the propulsion system was leading to fatigue failure of the stern tube bearing. |
| **References** |
| Stern tube bearing damage. Inner report of Info Marine No. RCH/I-M/13-0727, 27.03.2013<br><br>Murawski L.: Shaft line alignment analysis taking ship construction flexibility and deformations |

| |
|---|
| into consideration. Marine Structures No 1, Vol. 18, pp. 62-84, January 2005<br><br>Murawski L.: Identification of shaft line alignment with insufficient data availability. Polish Maritime Research No 1(59), Vol. 16, pp. 35-42, January 2009 |
| **Summary** |
| Causes of damage: overload of stern tube bearing caused by additional hydrodynamic forces; lack of proper lubrication due contamination with water and not enough lubrication oil pressure; fatigue of the stern tube bearing. Nevertheless, origin cause is neglect of the bad shaft line alignment (9 years). |
| **Legacy/Lessons learned** |
| • Shaft line alignment and crankshaft springing should be checked periodically or the structural health monitoring system should be installed. |
| **Figures** |

Fig. 1. Condition of aft and fore of stern tube bearing

Fig. 2. Coupling between intermediate and tail shafts

Fig.3 Coupling between intermediate and tail shafts (detail)

| |
|---|
| **Further reading** |
| |

## 4. Offshore Structures Failure

Offshore structures can be divided into 3 groups: fixed platforms (steel template and concrete gravity structures), compliant tower (compliant, guyed and articulated tower, tension leg platform) and floating structures (floating production, storage and offloading systems).
The loads on offshore structures are gravity (self-weight, various equipment, fixed platform elements, fluid loads), environmental (winds, waves, currents, ice), exploitation loads and seismic loads. Environmental loads play a major role in offshore structures design process.
In complex structures, such as offshore platforms, a fatigue failure of a single structural element may not result in a catastrophic failure of the entire structure but it definitely changes the expected lifetime of the structure. The need for structural system failure probability estimation of typical marine structures in combination of fatigue and fracture arises. A proposed numerical and analytical method had been tested on real structures, like a Neka jack-up platform (Iran Khazar) [13], by applying various fatigue sequences that could lead to the collapse of the platform structure. This comparison has shown that the calculated system failure probability is higher for the case of combined fatigue and fracture scenarios than for only fatigue or fracture induced structure collapse which emphasizes the need for regular inspections of marine structures.
Offshore pipelines are usually damaged in the form of dents and gouges which reduces its static and dynamic load bearing capacity as well as the fatigue life reduction in comparison to undamaged pipelines. The extent of the fatigue lifetime change depends on the type of the dent, and it can be analyzed and assessed analytically or numerically (FEM) [14]. Fatigue life analysis helps in the decision on the necessity of repairs and/or replacement of the damaged pipelines, i.e. planning of inspection and maintenance activities. Offshore pipelines segments are usually connected by welds which usually contain surface of embedded defects which exhibit large plastic strain characteristics if fracture occurs. In such cases nonlinear elastic plastic fracture response should be modeled [15].
Subsea structures are subjected to significant external pressure loads which makes structural buckling a dominant failure mechanism. Ultra-deep water subsea separators are key equipment of subsea production in offshore petroleum industry. An experimental and numerical investigation on buckling and post-buckling of a 3.000 m subsea separator has been done by Ge et al. [16]. The analysis has shown that the buckling behavior of deep sea structures can be assessed accurately applying numerical nonlinear global buckling analysis, proven by the comparison with experimental analysis results.

### *4.1 Alexander L. Kielland*

Table 7. Data regarding Alexander L. Kielland failure

| **Technical data/general information** | |
|---|---|
| **Structure type:** | Pentagonal type semi-submersible drilling rig |
| **Material:** | Brace D6 and the hydrophone support - C-Mn structural steel (equivalent to a Lloyds' ship steel Grade EH); minimum specified yield strength of 355 $N/mm^2$ |
| **Fate:** | capsized / sunk at 56.464839°N 3.104464°E |
| **Date of accident:** | March 27th 1980 |
| **Failure description** | |
| **Failure mode:** | |

| |
|---|
| Fatigue failure followed by brittle fracture in one brace and ductile overload in the remaining adjacent braces |
| **Failure cause:** |
| Fatigue crack growth from a weld defect |
| **Load type/conditions:** |
| Bad weather, approximately 60-75km/h wind speeds, approximately 6-8 mm wave height |
| **Analysis data** |
| **Failure analysis tools and methods used:** |
| Visual examination<br>Material properties testing (Charpy) |
| **Crack initiation:** |
| Fatigue failure of one brace-initiated by a gross fabrication defect |
| **Crack propagation:** |
| Ultimate progressive failure of braces |
| **Analysis results and conclusions** |
| Examination of brace-supports fillet welds revealed poor penetration into the hydrophone tube material and an unsatisfactory weld bead shape. Significant cracking was also found which was dated to the time of fabrication by the presence of paint on the fracture surfaces.<br>The investigation of the disaster concluded that the structural failure had occurred in three stages:<br>• Fatigue crack growth in brace D6 initiating from pre-existing cracks in the fillet welds between a hydrophone support and the brace<br>• Final, mainly ductile, fracture of brace D6<br>• Subsequent failure of five remaining braces joining the column to the structure by plastic collapse |
| **References** |
| 1. https://en.wikipedia.org/wiki/Alexander_L._Kielland_(platform)<br>2. https://officerofthewatch.com/2013/04/29/alexander-l-kielland-platform-capsize-accident/<br>3. http://www.twi-global.com/news-events/case-studies/alexander-l-kielland-accommodation-platform-145/ |
| **Summary** |
| The weather conditions on the evening of the accident were bad. The platform had five columns (overall height 35.6 m mounted on 22 m diameter pontoons) braced together and to the deck of hull, acting as principal buoyancy elements. One of the columns (designated "Column D") broke off which was followed by an immediate heeling to an 30°to 35° angle and then a slowly progressing heeling and finally capsizing and sinking of the platform.<br>It was determined that the fatigue fracture initiated in one brace (designated "D6") from pre-existing cracks in the welds between a hydrophone support and the brace, then a final ductile fracture of the brace occurred which caused plastic collapse of the remaining five column braces.<br>Material analysis has shown poor ductility characteristics through the thickness of the material. |

| Legacy/Lessons learned |
| --- |
| The investigation has shown that material properties, welding quality as well as the design process played a significant part in the failure of the structure. Stability and buoyancy aspects of the structures were inadequate; the design did not include additional strengthening of highly stressed braced (D6) as important. The influence of the hydrophone attachment on the fatigue life of the structure was overlooked, all of which leaded to a fatal accident with 123 lives lost. |

**Figures**

Fig. 1. Alexander L. Kielland platform

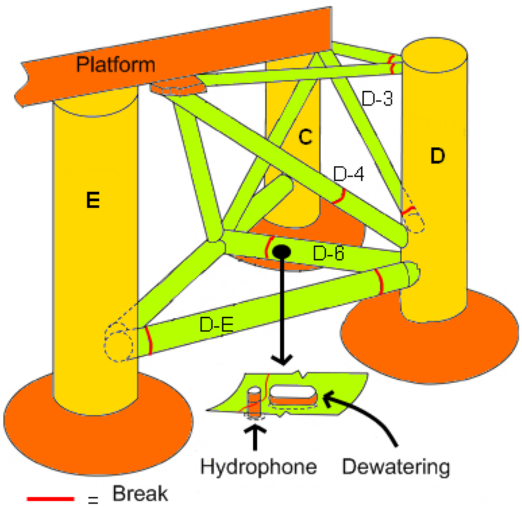



Fig. 2. Fractures on the rig

**Further reading**

B Hayes and R Phaal, TWI Industrial Member Report Summary 632/1998, Catastrophic failures of steel structures in industry: Case histories, 1998

### *4.2 Case Study of Sleipner A-1 (SLA-1) Gravity Base Structure*

Table 8. Data regarding Sleipner A-1 Gravity Base Structure failure

| **Technical data/general information** | |
|---|---|
| **Structure type:** | Condeep platform |
| **Material:** | Reinforced concrete |
| **Fate:** | Sank during a controlled ballast test |
| **Date of accident:** | 23rd August 1991 |
| **Failure description** | |
| **Failure mode:** | |
| Shear failure that split open several walls in one of the platform shafts, which led to rapid intake of water (crushing of the concrete, presumably at the intersection between the tri-cell wall and the cell joint due to lack of transverse structural reinforcement) | |
| **Failure cause:** | |
| The failure mechanism manifested because of several inconsistencies in the initial conditions defined in the design software (inappropriate use of finite element (FE) -code NASTRAN with regards to the global analysis of the finalized design, the finite element mesh used to analyse the tri-cells was too coarse to predict the shear stress accurately) | |
| **Load type/conditions:** | |
| Ballast test during deck mating | |
| **Analysis data** | |
| **Failure analysis tools and methods used:** | |
| Eyewitness accounts analysis<br>Analytical calculations<br>Testing of small and full scale models | |
| **Crack initiation:** | |
| Crack in concrete in the area of the tri-cell joint | |
| **Crack propagation:** | |
| Crushing of the concrete leading to significant water intake | |
| **Analysis results and conclusions** | |
| The SLA-1 platform had 24 buoyancy cells, four of which extended into the shafts that supported the deck. Two of the shafts served as "drill shafts" while the remaining two served as riser and utility shafts. The Gravity Base Structure was 110 meters tall, and designed to operate in 82 meters of water. The deck that would be mated to the SLA-1 Gravity Base Structure weighed approximately 57,000 tons. | |

| References |
| --- |
| 1. https://failures.wikispaces.com/Sleipner+A+-+North+Sea+Oil+Platform+Collapse<br>2. Collins, M., Vecchio, F., Selby, R., Gupta P (2000) "Failure of an Offshore Platform," Canadian Consulting Engineer - Structures, pp 43-48, March/April<br>3. Jakobsen, B. (1992) "Loss of the Sleipner A platform," The proceedings of the International Offshore and Polar Engineering Conference, pp. 1-9<br>4. Rettedal, W. (1993) "Design of concrete platforms after Sleipner A-1 sinking," Proceedings of the International Conference on Offshore Mechanics and Arctic Engineering – OMAE, pp. 309-316<br>5. Jakobsen, B. (1994) "The Sleipner Accident and its Causes," Engineering Failure Analysis, Vol. 1, No. 3, pp. 193-199<br>6. Ynnesdal, H., Berger, F. (1994) "The Sleipner Accident," Proceedings of the Second International Conference on Health, Safety & Environment in Oil & Gas Exploration & Production, pp. 715-716, 25-27 January |
| **Summary** |
| Condeep platforms are reinforced concrete structures meant to float in water up to 300 meters deep and are made up of several buoyancy cells that function as a floating mechanism. Water pumped into the buoyancy cells is then used to regulate the depth of the Gravity Base Structure in the sea. A number of the buoyancy cells have upward extensions, called "shafts", which serve as structural supports to the deck substructure.<br>This type of platforms undergo several cycles of submerging during the construction (deck-mating), ballast test and the voyage to its final destination. Due to the fact that the structure is made of concrete, extreme care has to be taken during the design phase.<br>During the second controlled ballast test which is an integral part of the deck-mating procedure, the platform began to take on water uncontrollably. The initial intake of water was denoted with a very "deep bang-like sound" as eyewitnesses described it.<br>The analysis of the accident concluded that the tri-cell walls and supports at the cell joints were the weakest points in the platform, and that the final failure was believed to take place as crushing of the concrete, presumably at the intersection between the tri-cell wall and the cell joint. This failure mechanism manifested because of several inconsistencies in the initial conditions defined in the design software as well as considerable complexity of the software itself. Additionally, the supports for the tri-cell walls in SLA-1 were designed to only resist lateral forces indirectly, which meant that the detailing for the tri-cell joints had to be very carefully designed and analysed. |
| **Legacy/Lessons learned** |
| • The need for extreme care and detail in design<br>• Importance of having experienced engineers verify computer-generated design work to ensure the proper use of analysis and design techniques<br>• Revised design philosophy with greater attention to construction details and numerical analysis results control |

**Figures**

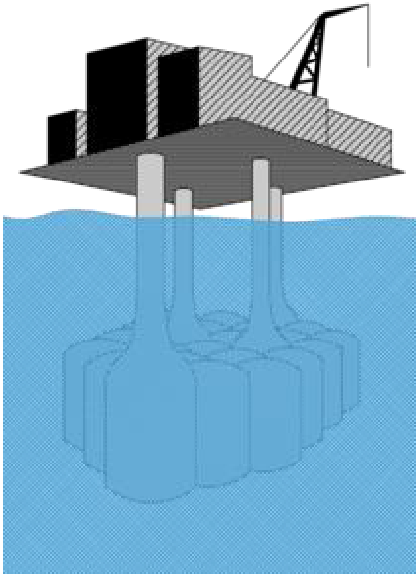


Fig. 1. Rendering of a typical Condeep platform

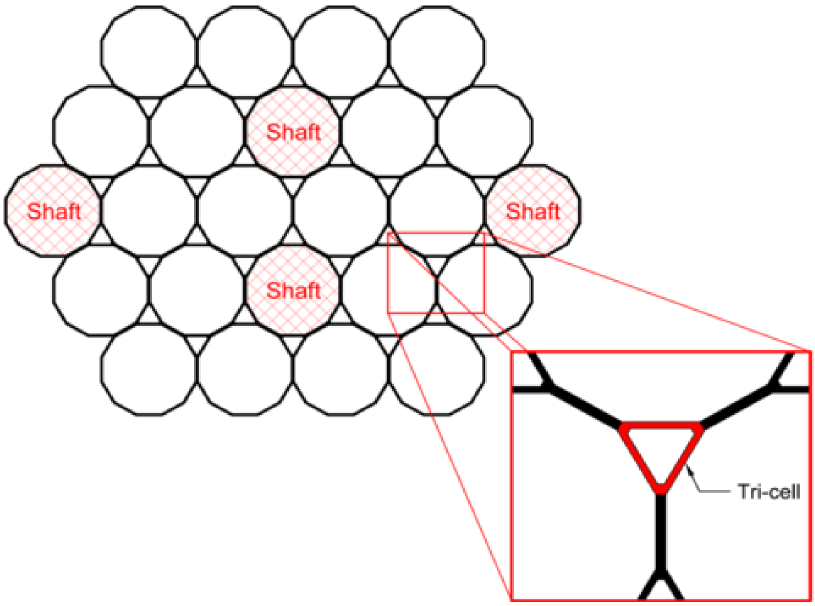



Fig. 2. Plan view of SLA-1 buoyancy cells

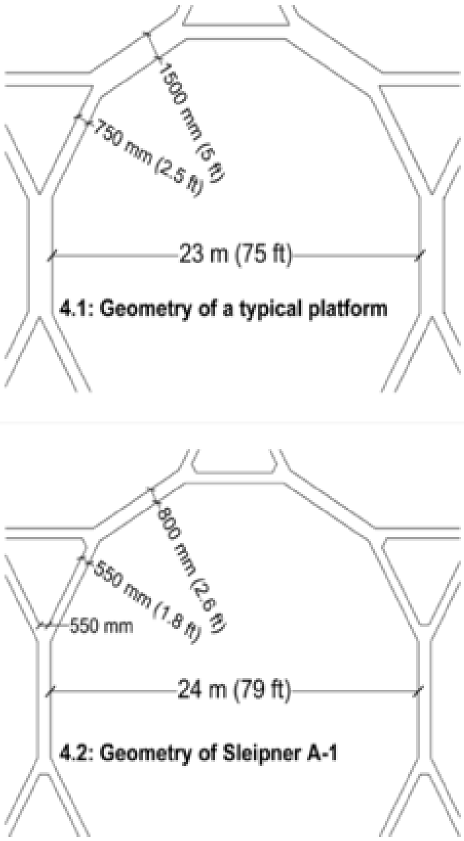



Fig.3 Tri-cell geometry of a typical platform compared to the geometry of SLA-1

**Further reading**

1. Arnold, D. (2009) "The sinking of the Sleipner A offshore platform," Institute for Mathematics and its Applications (IMA) at University of Minnesota, 7 September 2009, Web.
2. Jakobsen, B., Rosendahl, F. (1994) "The Sleipner Platform Accident," Structural Engineering International, IABSE, Vol. 3, pp. 190-193

## 5. Marine Equipment Failure

This section deals with failures of marine equipment such as port or dock cranes, cables and ropes, pressure vessels mounted on-board ships, underwater pipelines.

Cranes can be subject to unexpected sudden events which can be divided into accidents and emergencies. Catastrophic failure of a dockside crane jib [17] occurred in the proximity of the standing tower, near the connection of the jib’s three main tubes to the tower. Upon the visual inspection of the fracture surfaces the presence of a large preexisting crack was evident. The crack originated from a seam weld and propagated through one of the main pipes of the crane jib space frame. The failure occurred during maneuvering with no load attached. During the investigation crane material properties were obtained experimentally (tensile tests and Charpy impact tests) and the crane design was verified by FE analysis. Fatigue analysis was conducted according to standards (FEM

1.001, Eurocode 3) for the welding joints and the pipes. Failure mode analysis was done from fracture mechanics and plastic collapse approaches. All of the analysis and investigations brought to the conclusion that the fatigue design of the jib structure was not done according to standards and that the final failure was determined by plastic collapse, after a long stable propagation period of a dominant crack which originated at the edge of a seam weld.

As for the pressure vessel failures, there are two main reasons for failures, i.e. pressure part failure (safety valves failures, corrosion, low water level) or fuel/air explosions in the furnace (gas or liquid fuel leaks). Inadequate construction characteristics of high pressure tubes can cause failures. An investigation of a prematurely ruptured high-pressure oil tube has shown that inadequate pipe type (longitudinally welded instead of seamless) and material (design specified material replaced by a lower grade one) as well as inadequate installation procedures (not enough pipe clamps which allowed vibrations) resulted in vibration induced fatigue crack [18].

All equipment on marine structures is maintained and serviced continuously. In case of a malfunction in-situ repairs are often performed. The quality of workmanship and material choice do have a great importance in such cases. Bending stresses in equipment elements that should be subjected only to tensile stress (ropes, wires etc.) can cause failure of such elements. Numerical analysis of different wire rope cross section configurations is performed in order to determine remaining fatigue of operating wire ropes in dockside cranes [19].

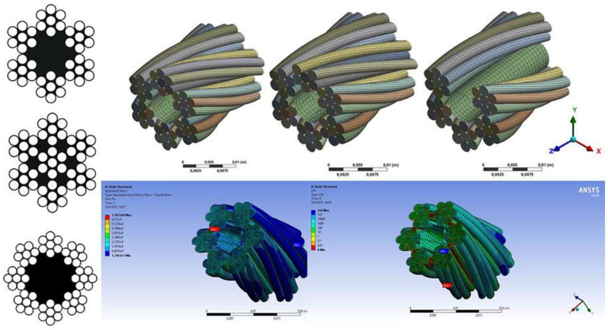


Fig. 5.1 Numerical analysis of remaining fatigue life of a wire rope

Subsea umbilicals are composite cable and small diameter tubular bundles deployed on the seabed in conjunction with offshore installations for oil or gas exploitation. These tubes are loaded by alternating internal pressure and exposed to sea currents, i.e. dynamic loading [20]. Cracks in this type of equipment result in leaks and loss off load carrying capacity. Umbilical tubes experience loss of circularity in shape (ovalization) and are subjected to re-rounding procedures by applying boost pressure prior to service which also translates in fatigue loading.

### *5.1 Case Study of Sea Angel Crane Failure*

Table 9. Data regarding Sea Angel crane failure

| **Technical data/general information** | |
|---|---|
| **Structure type:** | Hydraulic crane aboard MV Sea Angel, SWL 25t/22m |
| **Material:** | Steel |
| **Fate:** | The port jib arm of No.3 hydraulic crane detached from the crane’s heel pin |
| **Date of accident:** | October 31$^{st}$ 2005 |

| **Failure description** |
|---|
| **Failure mode:** |
| Fatigue crack |
| **Failure cause:** |
| Fatigue wear of bolts |
| **Load type/conditions:** |
| - |
| **Analysis data** |
| **Failure analysis tools and methods used:** |
| Visual inspection and examination<br>Non Destruction Crack Test |
| **Crack initiation:** |
| Three of the heel pin retaining bolts |
| **Crack propagation:** |
| |
| **Analysis results and conclusions** |
| During the cargo-loading process the port jib arm of No.3 hydraulic crane detached from the crane's heel pin, causing serious damage to the jib of the crane. A visual inspection of the internal jib, near the cut off sections, showed the steelwork of the jib to be in good condition. Close examination of the heel pin bearing cover confirmed that the paint coated on the cover's joints and bolts had not been disturbed or broken for some time, indicating that the heel pin retaining bolts had not been checked or inspected recently.<br>Further examination of the crane showed that only two of the retaining bolts were intact. The third bolt had fractured near the bolt head. The bolt also showed beach marks on the fracture surface, that are a characteristic of fatigue cracking over a period under cyclic loading. A Non Destruction Crack Test showed fatigue cracks on all three bolts that were found at the bottom of the second threads and near the bolt head. By reviewing maintenance records and crewmember depositions, as well as the investigation tests results it became obvious that the bolts were cracked even before the accident itself.<br>The investigation found various contributing factors that have caused the accident:<br>• Same type of cranes exhibited similar issues, resulting in the crane manufacturer issuing a Technical information bulletin<br>• Improper usage of the crane (dragging cargo with the crane represents an overload for the used 20mm diameter heel bolts)<br>• Poorly executed inspection & maintenance requirements/recommendations procedures by the crew<br>• The crane producer Surveyor also did not follow inspection requirements and recommendations fully<br>• The poor condition of the heel pin locking plate/device was considered one of the possible contributing factors for the heel bolts to work loose. However, the discovery of cracks on all the loose bolts found and all the detached bolts from the crane would make the condition of the heel bolt locking plate/device only a minor contributing factor to the eventual failure of the crane jib, because, if the bolts were properly locked up, they could still fracture and fail |

| References |
|---|
| 1. Maritime New Zealand, Accident Report No. 05 3888 – Crane Failure Sea Angel |
| **Summary** |
| On October 31$^{st}$ 2005 whilst Sea Angel was loading logs at the port jib arm of a hydraulic crane detached from the crane's heel pin, causing serious damage to the jib of the crane. There were no injuries. The subsequent investigation has found severe negligence during inspection and maintenance activities, as well as non-adequately dimensioned heel bolts that proven critical for certain crane operations. Possible pre-existing cracks were found on the bolts indicating that this particular accident has been caused by a combination of poor maintenance and fatigue. |

| Legacy/Lessons learned |
|---|
| • The importance of inspection & maintenance requirements/recommendations<br>• The importance of adequate information circulation and feedback information in and from exploitation<br>• The importance of crew education in recognizing and assessing equipment condition and behaviour during exploitation |
| **Figures** |

Fig. 1. MV Sea Angel

Fig. 2. Failure details

**Further reading**

-

### *5.2 Case study of speed boat steering wheel failure*

Table 10. Data regarding speed boat steering wheel failure

| Technical data/general information | |
|---|---|
| **Structure type:** | Steering wheel |
| **Material:** | aluminium alloy AA 6061 |
| **Fate:** | Structural failure |
| **Date of accident:** | 2014 |
| **Failure description** | |
| **Failure mode:** | |
| | |
| **Failure cause:** | |
| Excessive fastener torque moment, fretting between fastener and hole combined, poor machining process | |
| **Load type/conditions:** | |
| Torque on fasteners, arm force on two points of the rim | |
| **Analysis data** | |
| **Failure analysis tools and methods used:** | |
| Torque value test<br>Visual examination<br>Fractographic analysis<br>Scanning electron microscopy examination<br>Numerical analysis<br>Finite element analysis | |
| **Crack initiation:** | |
| Cracks emanating from one of the fastener holes | |
| **Crack propagation:** | |
| Through the thickness of the steering wheel, toward the outer edges | |
| **Analysis results and conclusions** | |
| During regular use of the steering wheel, a crack initiation zone was observed. The direction of the crack propagation is through the thickness of steering wheel, continuing toward the outer edges.<br>During investigation, machining or fretting damage was identified as a probable cause of the failure. Fracture area consists of dimple fracture and transgranular cleavage, separated by crack gaps. Fracture surface can be attributed mostly as transcrystal, with scarce areas of intercrystal fracture. Experimental study of fractured speedboat steering wheel revealed the material to be aluminum alloy AA6061-O, one of the most common aluminum alloys, widely used in marine industry, among others. | |

Visual examination of cracked steering wheel revealed machining and fretting marks on the surface of fastener hole from which cracks emanated. These marks served as initiation points for crack growth.
Measured torque values of fasteners showed that the fastener at hole from which cracks emanated had relatively high torque value comparing to others. This excessive load, combined with the load of driver's hand, speeded up crack propagation.
Detailed SEM examination of the fractured surface confirmed cracks growing from the mentioned marks and showed direction of crack propagation to be through the thickness of steering wheel and toward the outer edges. Fracture area consists of dimple fracture and transgranular cleavage separated by crack gaps near the fastener hole. Surface consists of some cleavage step pattern that reminds of Wallner lines. Cracks between flat surfaces and cleavage suggest possible fracture initiation point.
In addition, numerical analysis showed maximum stress level at the point of crack initiation on the outer edge of fastener hole. Same load produced higher stress level when the cracks were added to the FE model shifting them to the crack tips making way for propagation of the cracks. Joint stresses produced in the local stress concentration point at the fastener holes further enhance the fracture evolution.

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**Summary**

During regular use of the steering wheel, cracks started emanating from one of the six fixing holes by which the wheel was attached to column. After the final fracture, the wheel was detached from boat and subjected to fracture analysis. Results, obtained by experimental and numerical approach, suggest greater care should be taken in machining and mounting the wheel in order to avoid initial damage to the surface that could serve as a point of crack initiation. In addition, care should be taken when tightening the fasteners not to exceed the torque limits the additional load can improve crack growth.

**Legacy/Lessons learned**

- The importance of adequate maintenance procedure

**Figures**

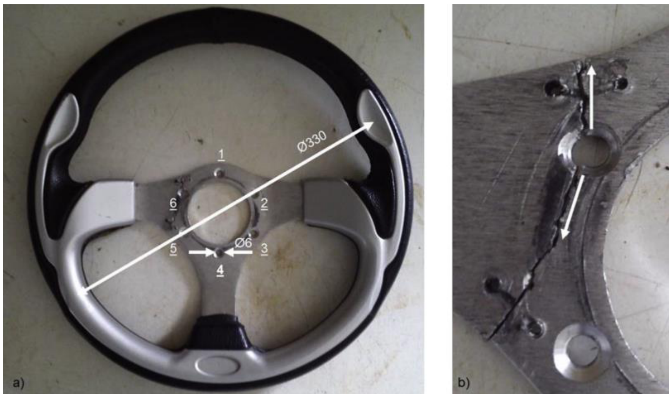

Fig. 1. Speed boat steering wheel

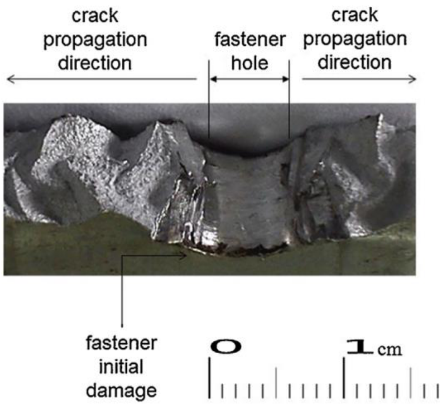

Fig. 2. Crack detail

**Further reading**

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## 6. Failure Causes and Mechanisms

The strength of a structure represents a limit state of loading conditions above which the structure loses ability to achieve its specified required function. As long as the actual strength of the structure is kept higher than the actual loading demands, a given marine structure can be deemed safe. Otherwise, structural failures will occur.

Structural failure can be defined as loss of the load-carrying capacity of a component or member within a structure or of the structure itself (including global failure modes like capsizing, sinking, positioning system failures etc.). The failure can result in catastrophic damage (i.e. complete loss of the structure itself) or partial structure damage when the structure can be repaired or recovered. Global failures can more often result in fatal casualties while smaller and localized structural damage may result in pollution and recoverable structural damage.

Structural failure is initiated when the material in a structure is stressed to its strength limit, thus causing fracture or excessive deformations. The structural integrity of a marine structure depends on load conditions, the strength of the structure itself, manufacturing and materials quality level, severity of service conditions, design quality as well as various human elements that have effects during exploitation of the structure.
There are two distinctive groups of failure causes. The first group is comprised of unforeseeable external or environmental effects which exert additional loading on the structure resulting in over-load. Such effects are extreme weather (overloads), accidental loads (collisions, explosions, fire, etc.) and operational errors. The second group comprises causes for failures that occur either during the design and construction phase (dimensioning errors, poor construction workmanship, material imperfections) or due to phenomena growing in time (fatigue), both resulting in reduced actual strength in respect to the design value. All of the listed causes can partially or completely be a result of human factor.
The process of fatigue failure itself is highly complex in nature and it is dependent on a large number of parameters. The factors are numerous and perhaps the most significant are mean stress (distribution), residual stresses, loading characteristics and sequence, structural dimensions, corrosion parameters, environmental temperature, design criteria fabrication methods and quality.
Failure mechanisms that usually occur in marine structures can be progressive (excessive yielding, buckling, excessive deformations) or sudden (brittle and fatigue fractures). Excessive yielding and brittle fractures occur when the load exceeds critical strength, whilst buckling and fatigue fractures depend on time and specific load conditions.

# 7. Failure Analysis Tools

The analysis methods can be grouped into methods that use nominal stresses (typical for standard codes) acting to a structure or part of a structure and then compare the stress amplitude to nominal S-N curves. This approach is appropriate for structures that are standardized and therefore well backed up with statistical experimental data that can be used as initial assumptions for fatigue analysis. The alternative is the evaluation of local stresses influence to fatigue (notch stress factors, N-SIF).
Some authors [21] divide fatigue analysis methods in two groups: S-N approach based on fatigue tests and fracture mechanics approach. The first method is used for fatigue design purpose using simplified fatigue analysis, spectral fatigue analysis or time domain fatigue analysis to determine fatigue loads. The second method is used for determination of acceptable flaw size, prediction of crack growth behaviour, planning maintenance of the structure and similar activities.
The latest trend in failure analysis development is the unification of analysis methods and procedures [22-24] in order to obtain a comprehensive procedure of structural failure analysis that would cover main failure modes and enable a safer and more efficient design, manufacture and maintenance processes.

## *7.1. Experimental tools*

Nondestructive testing and examination (NDT, NDE), as well as structural health monitoring (SHM), of structures play a significant role in fracture analysis and control procedures. Any method used must not alter, change or modify the failed condition but must survey the failure in a nondestructive mode so as to not impact, change or further degrade the failure zone. This kind of examination provides input values for fracture analysis which yields results that define inspection and maintenance intervals for the structure and represent input values for life prediction estimates. Structures are inspected at the beginning of their service life in order to document initial flaws which determine the starting point of

the structure fatigue life prediction. The most commonly used procedures for marine structures are optical microscopy, scanning electron microscopy (SEM), GDS and acoustic emission (AE) testing.
Optical microscopy is a common and most widely used NDT analysis method which enables rapid location and identification of most external material defects. This technique is often used in conjunction with micro-sectioning to broaden the application. One of the main disadvantages is the narrow depth-of-field, especially at higher magnifications.
Scanning electron microscopy is an extension of optical microscopy in failure analysis. The use of electrons instead of a light source provides much higher magnification (up to 100,000x) and much better depth of field, unique imaging, and the opportunity to perform elemental analysis and phase identification. The examined item is placed in a vacuum enclosure and exposed with a finely focused electron beam. The main advantage of this method is minimal specimen preparation activity due to the fact that the thickness of the specimen does not pose any influence to the analysis, ultra-high resolution and 3D resulting appearance of the test object. Various analysis of marine structures and equipment have been conducted using SEM [25-28], one of them being analysis of speed boat steering wheel fracture.
As it is well known, structural supporting members emit sounds prior to their collapse i.e. failure. This fact has been the basis of the development of scientific methods of monitoring and analysis of these sounds with the goal to detect and locate faults in mechanically loaded structures and components. AE provides comprehensive information on the origin of a discontinuity (flaw) in a stressed component and also provides information about the development of flaws in structures under dynamic loading. Discontinuities in stressed components release energy which travels in the form of high-frequency stress waves. Ultrasonic sensors (20 kHz – 1 MHz) receive these waves or oscillations and turn them in electronical signals which are in turn processed on a computer yielding data about the source location, intensity frequency spectrum and other parameters that are of interest for the analysis. This method is passive, i.e. no active source of energy is applied in order to create observable effects as in other NDT methods (ultrasonic, radiography etc.). Three sources of acoustic emissions are recognized, namely primary, secondary and noise. The primary sources have the greatest structural significance and originate in permanent defects in the material that manifest as local stresses, either on microstructural or macrostructural level. The amount of acoustic emission energy released, and the amplitude of the resulting wave, depends on the size and the speed of the source event. The main advantages of AE compared to other NDT methods that AE can be used in all stages of testing, lesser geometry sensitivity, the method is stress related, less intrusive method, it can be used for global monitoring, the scanning is remote and it gives a real-time evaluation [29]. The disadvantages are the sensitivity to signal attenuation in the structure, less repeatability do to the uniqueness of emissions for a specific stress/loading conditions and external noise influence on accuracy.

### *7.2. Analytical tools*

Although various analytical models have been proposed by a number of authors no comprehensive model exists. Analytical methods have been developed for prediction of progressive structural failures of marine structures [30]. The finite element modeling approach for prediction of the development of failures is accurate, but can be time consuming. Analytical procedures, based on spectral fatigue analysis, beam theory, fracture mechanics and structural factors, can provide solutions in considerably less time when needed.
The goal is to define approaches for computing the fracture driving force in structural components that contain cracks. The most appropriate analytical methodology for a given situation depends on geometry, loading, and material properties. The decisive choice factor is the character of stress. If the structure behavior is predominantly elastic, linear elastic fracture mechanics can yield acceptable results. On the other hand, when significant yielding precedes fracture, elastic-plastic methods such as referent stress approach (RSA) and failure assessment diagram (FAD) need to be used. Since a purely linear elastic fracture analysis can yield invalid and inaccurate results, the safest approach is to adopt

an analysis that spans the entire range from linear elastic to fully plastic behavior. One of the methodology that can be applied is the FAD approach.
The FAD approach has first been developed from the strip-yield model and it uses two parameters which are linearly dependent to the applied load. This method can be applied to analyze and model brittle fracture (from linear elastic to ductile overload), welded components fatigue behavior or ductile tearing. The stress intensity factors are defined on the basis of the structure collapse stress and the geometry dependence of the strip-yield model is eliminated [31, 32]. The result is a curve that represents a set of points of predicted failure points, hence the name failure assessment diagram. The failure assessment diagram is basically an alternative method for graphically representing the fracture driving force.
Depending on the type of the equation used to model the effective stress intensity factors the FAD approach can be sub-divided into the strip-yield based FAD, J-based FAD and approximated FAD. The J-based FAD includes the effects of hardening of the material, while the simplified approximations of the FAD curve are used to reduce the calculation times of the analysis. When stress-strain data are not available for the material of interest generic FAD expressions may be used [33], that assume that the FAD is independent of both geometry and material properties. The simplified curves proved adequate for most practical applications due to the fact that design stresses are usually below yield point. Fracture analysis in fully plastic regime require an elastic-plastic J analysis.
Marine structures are subjected to dynamic load that are characterized by exactly unpredictable, stochastic changes of value (environmental factors). Most fracture mechanics analyses are deterministic, therefore a need to view fracture probabilistically for real world conditions arise. The probabilistic fracture analysis overlaps the probability distributions of driving force in the structure and toughness distribution in the structure to obtain a finite probability of failure. Probabilistic methods can take into account time-dependent crack growth and stress corrosion cracking by applying appropriate distribution laws. Most practical situations exhibit randomness and uncertainty of the analysis variables so numerical algorithms for probabilistic analysis may be needed to apply. The well-known Mote Carlo method has been proven to be suited to accompany FAD models in cases of uncertainties.
Recently, normative institutions have been involved in projects and research, together with industry, in order to establish probabilistic methods for planning in-service inspection for fatigue cracks in offshore structures. DNV issued recommendations on how to use probabilistic methods for jacket structures, semisubmersibles and floating production ships [34]. Basically, the goal of probabilistic method is to replace inspection planning based on engineering assessment of fatigue and failure consequences with mathematical models for the influence of exploitation, fatigue causes and crack propagation characteristics on the lifetime of the structure to obtain a more reliable and secure assessment methodology independent of the engineers' level of expertise.
Li and Chow [35] have developed a fatigue damage model by formulating a set of damage coupled constitutive and evolution equations in order to write a computer software that could predict the behaviour of offshore structures under dynamic load. The fatigue damage model is based on sea wave's characteristics statistics. The model also includes historical damage data.
Cui [36] has focused his research on the requirement for accurate fracture growth predictions that preceding fatigue strength assessment methods, mainly based on cumulative fatigue damage theory using stress-endurance curves (S-N), have not taken into account. The effects of initial defects and load sequence are included in the prediction model. A fatigue crack propagation theory has been proposed as technically feasible and adoptable method for fatigue life prediction using commercial FEA/FEM software packages for the calculation processes. The need for a database of the size and distribution of initial defects for marine structures is emphasized.
Li et al. [37] have developed an improved procedure for creation of standardized load-time history for marine structures based on a short-term load measurement. The need for load-time history arises from the dependency fatigue crack growth behavior to load sequence effect.

It is known that small variations in the initial (basic) assumptions for a fatigue analysis can have significant influence for the predicted crack growth time. As mentioned above, the S-N based calculations are sensitive to input parameters values and definitions [38]. As the occurrence of a crack is not strictly deterministic, probabilistic methods for the prediction of crack behavior and sizes, based on fatigue crack propagation theory, can resolve accuracy problems. Probabilistic methods require extensive database of standardized load-time histories for marine structures, based on extensive experimental research, which can be used in analysis procedures.

### *7.3. Numerical tools*

The effective application of numerical methods in fracture mechanics and fatigue analysis begun with the development of computer science in the second half of the 20th century. Various methods were used (finite difference method, collocation methods, Fourier-transformations) but the finite elements method (FEM) has been established as a standard due to its universality and efficiency. FEM enables complicated crack configuration analysis under complex loads and non-linear material behavior [25].

Fig. 7.3.1. Numerical model of fractured speed boat steering wheel.

Recent years have brought a significant development and increase in accessibility of commercial computational software and hardware for finite element analysis applications, marine structures included. This enables more advanced and detailed fatigue and fracture analysis even for more complex large-scale structures. Furthermore, numerical tools can be used to complement or even substitute experimental analyses, as in the material selection stage in design process [39].
As the extent of scientific material published on this matter is very ample, here recently developed methods will be briefly described and referenced.
Extended FEM (X-FEM) is the most recent finite element method developed and is used mainly for fracture mechanics applications. Based on the finite element method and fracture mechanics theory, X-FEM can be applied to solve complicated discontinuity issues including fracture, interface, and damage problems with great potential for use in multi-scale computation and multi-phase coupling problems. The method has been introduced in 1999. [40], and since then further developed by various authors. The basic idea of the method is to reduce the re-meshing around the crack to a minimum. The improvements enabled the crack to be represented in the FE model independently from the mesh itself [40]. The solution for the problem of modeling curved cracks was developed by forming higher order elements [42]. Improved XFEM methods are continuously being developed by various researchers as the method has been proven as very valuable.
Various computer software packages for fatigue crack growth analysis have been developed by NASA. FASTRAN is a life-prediction code based on the crack-closure concept and is used to predict crack length against cycles from a specified initial crack size to failure for many common crack configurations found in structural components. NASA FLAGRO v2 fatigue crack growth computer program developed as an aid in predicting the growth of pre-existing flaws and cracks in structural

components using a two-dimensional model which predicts growth independently in two directions based on the calculation of stress intensity factors.
Recently, specific numerical automatic crack box technique (CBT) has been developed in order to enable to perform fine fracture mechanics calculations in various structures without global re-meshing [43]. The algorithm can be used for FEM calculations with ABAQUS code. The method represents an improvement as only the specific crack zone has to be re-meshed which results in simpler and time saving calculations. Also, the method allows the analysis of the influence of plastic material characteristics on the crack growth path.

## 8. Conclusions

This report provides an overview of common failures of marine structures taking into account failure mechanisms and tools used for failure analysis. As shown, the majority of employed failure analysis is comprised of visual, analytical and mechanical inspection methods in the attempt to identify failure causes. The working conditions in which marine structures operate are often stochastic in nature and strongly dependant on weather conditions at sea as well on loading conditions of the structure. The complexity of failure analysis accentuates the need for numerical simulation of possible catastrophic scenarios during the entire lifetime span of the structure. If the marine structures coupled with the relevant data collected during maintenance procedures are numerically modelled than a tool for failure prediction can be developed. Therefore, complete analysis comprising of analytical, experimental and numerical research is desirable to obtain satisfying results.
Throughout this report potential threats affecting marine structural integrity have been identified and, using experimental and numerical approach, various cases of failures have beey analyzed. Structural critical points that could serve as a root of failure have been assesed and, based on this findings, a database comprised of ten elaborated case studies dealing with marine structural failures has been established. This database, i.e. learning from actual examples from engineering practice, is to be used in subsequent education of marine engineers in order for them to understand causes of marine structural failures, structure's load response, failure process, possible consequences and methods to cope with and prevent failures.

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