University of Rijeka, Faculty of Maritime studies

PROJECT ACRONYM AND TITLE: Marine Structural Failures Database (MarStruFail)

FUNDING PROGRAMME: International Association of Maritime Universities (IAMU)

PERSON RESPONSIBLE: Goran Vukelić

FINANCIAL DATA

Project total cost	Overall funding assigned to PFRI
57.600 USD	46.080 USD

SUMMARY

Goals of "Marine Structural Failures Database" research project include:
- identifying potential threats affecting marine structural integrity,
- analyzing various cases of failures using experimental and numerical approach,
- assessing structural critical points that could serve as a root of failure,
- forming a database comprised of elaborated case studies that can be used in the education worldwide,
- dissemination of results and promoting open access to database.

Start date	End date
01.05.2017.	31.05.2018.

PARTNERSHIP

Br.	Partner organization	Country	Role
1.	Faculty of Maritime Studies Rijeka	Croatia	Lead partner
2.	Gdynia Maritime University	Poland	Partner
3.	EPF-Ecoles d'Ingénieurs Sceaux	France	Partner

WEBSITE: -

Marine Structural Failures Database

Goran Vukelic
Associate Professor, Faculty of Maritime Studies, gvukelic@pfri.hr
Goran Vizentin
Assistant, Faculty of Maritime Studies
Lech Murawski
Professor, Gdynia Maritime University
Naman Recho
Professor, Blaise Pascal University-Clermont II and EPF-Ecoles d'Ingénieurs

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 paper provides an overview of most common case studies of marine structural failures taking into account failure mechanisms, tools used for failure analysis and critical review of possible improvements in failure analysis techniques.

1. Ship Structural Failures

1.1 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 15th 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 Visual crack inspection Pellini (drop weight test) NDT crack tip opening displacement (CTOD) tests fracture mechanics calculations performed using PD6493 (1980) procedures

Crack initiation:
The initial fracture through the bottom and side shell plates was brittle 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.

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.

The subsequent breaking of the ship in two was inevitable due to the extensive structural damage caused by the fatigue crack propagation.

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.

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

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
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.

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.

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.
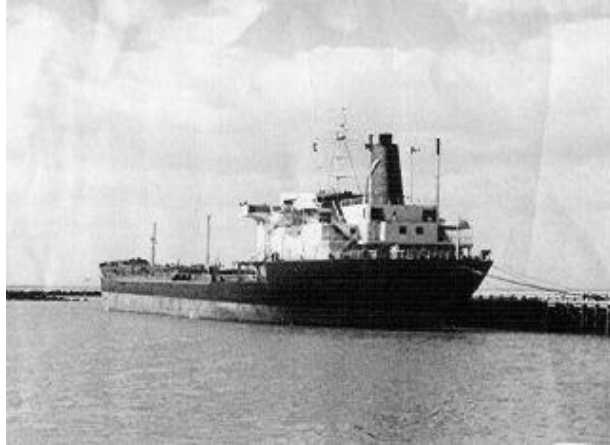
Figures

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/ 2. https://www.wrecksite.eu/wreck.aspx?34872

1.2 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 3-hold model elasto-plastic analyses Probabilistic load estimation 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:

- 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

1.3 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 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
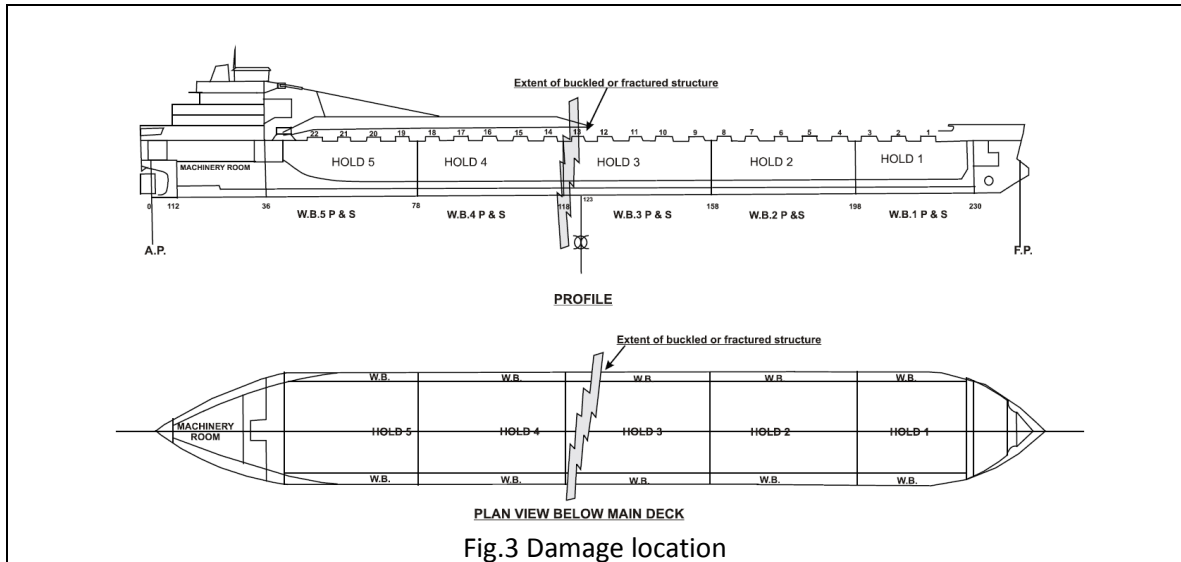

Fig. 2. Damage detail

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

2. Propulsion System Failures

2.1 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:	-
Date of accident:	-
Failure description	
Failure mode:	
Bending-torsional fatigue crack	
Failure cause:	
Material imperfections Fatigue stresses	
Load type/conditions:	
combination of cyclic bending and steady torsion	
Analysis data	

Failure analysis tools and methods used:

Microscopy (eye seen) observation
Linear elastic fracture mechanics
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.

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

2.2 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 Root Cause Analysis (RCA) Fault Tree Analysis (FTA) method Chemical composition analysis Micro-structural characterization Fractography Hardness measurements Finite element simulation	
Crack initiation:	
Ends of keyways (stress concentration factor) Filets, tapers and chamfers in the shaft geometry Shaft spline joints Bolted connections 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

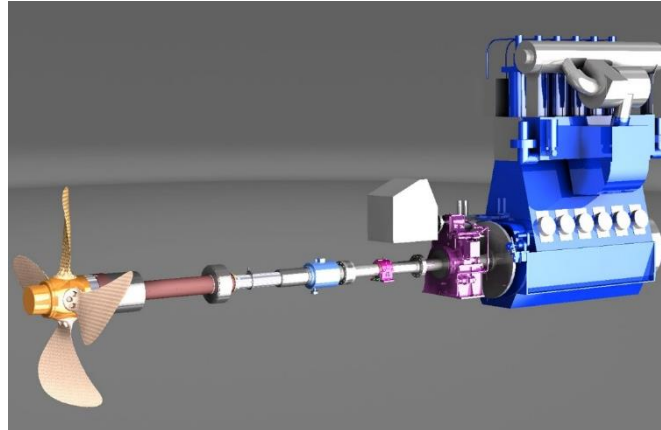- The importance of geometry details in the design phase

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

2.3 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:	
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 Laser alignment measurements Crankshaft springing test 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. 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. 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

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

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

3. Offshore Structures Failure

3.1 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 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.

The investigation of the disaster concluded that the structural failure had occurred in three stages:

- Fatigue crack growth in brace D6 initiating from pre-existing cracks in the fillet welds between a hydrophone support and the brace
- Final, mainly ductile, fracture of brace D6
- 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)
2. https://officerofthewatch.com/2013/04/29/alexander-l-kielland-platform-capsize-accident/
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.

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. 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.
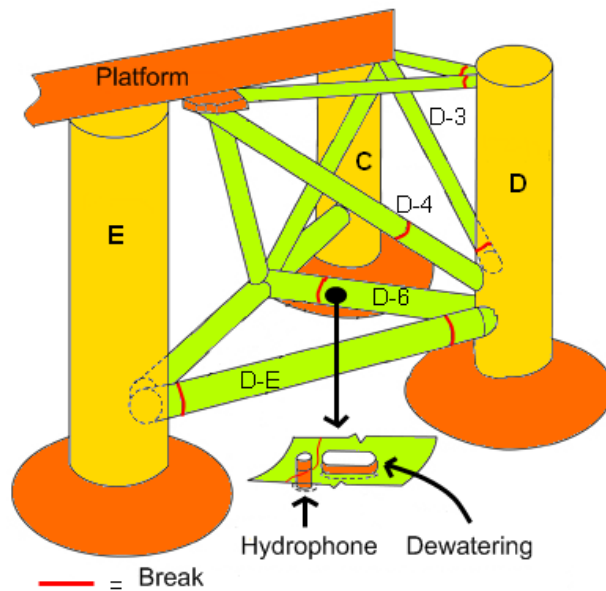
Figures

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

3.2 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 Analytical calculations 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 2. Collins, M., Vecchio, F., Selby, R., Gupta P (2000) "Failure of an Offshore Platform," Canadian Consulting Engineer - Structures, pp 43-48, March/April

3. Jakobsen, B. (1992) "Loss of the Sleipner A platform," The proceedings of the International Offshore and Polar Engineering Conference, pp. 1-9

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

5. Jakobsen, B. (1994) "The Sleipner Accident and its Causes," Engineering Failure Analysis, Vol. 1, No. 3, pp. 193-199

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.

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.

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.

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
- Importance of having experienced engineers verify computer-generated design work to ensure the proper use of analysis and design techniques
- Revised design philosophy with greater attention to construction details and numerical analysis results control
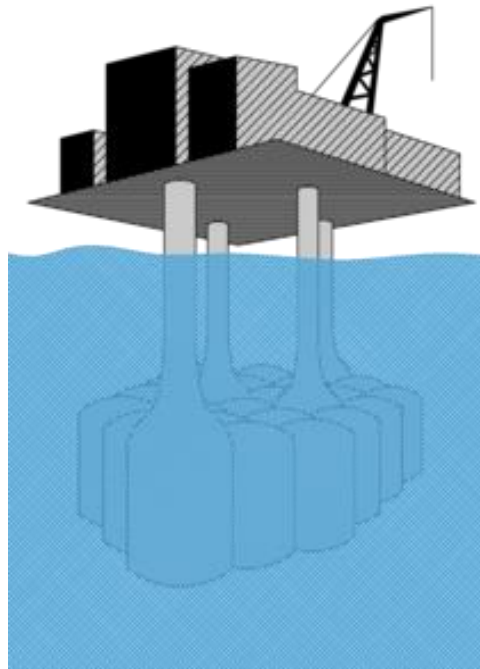
Figures

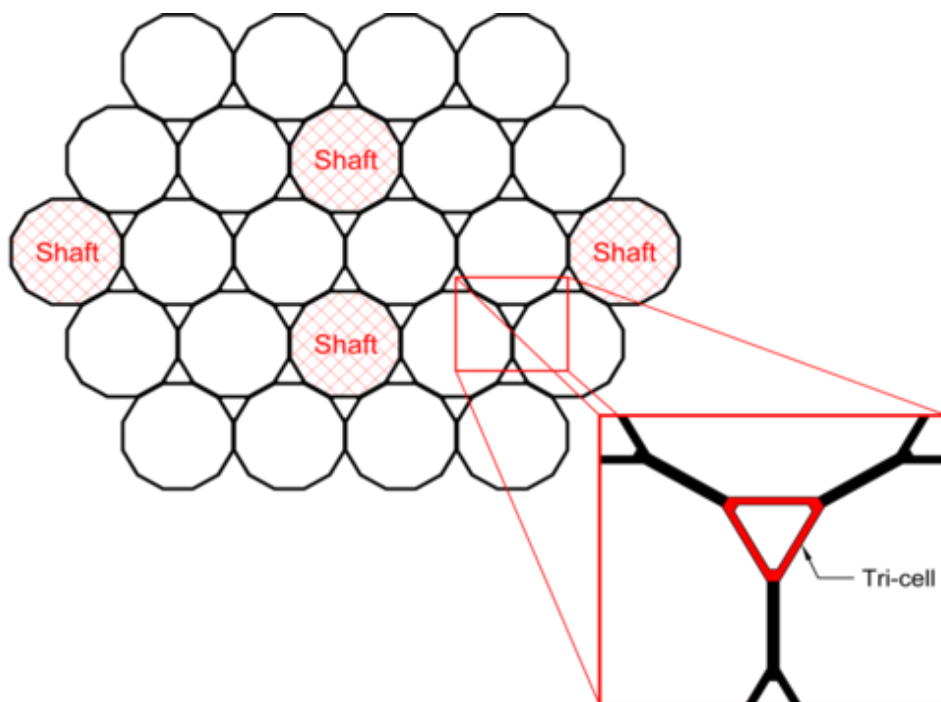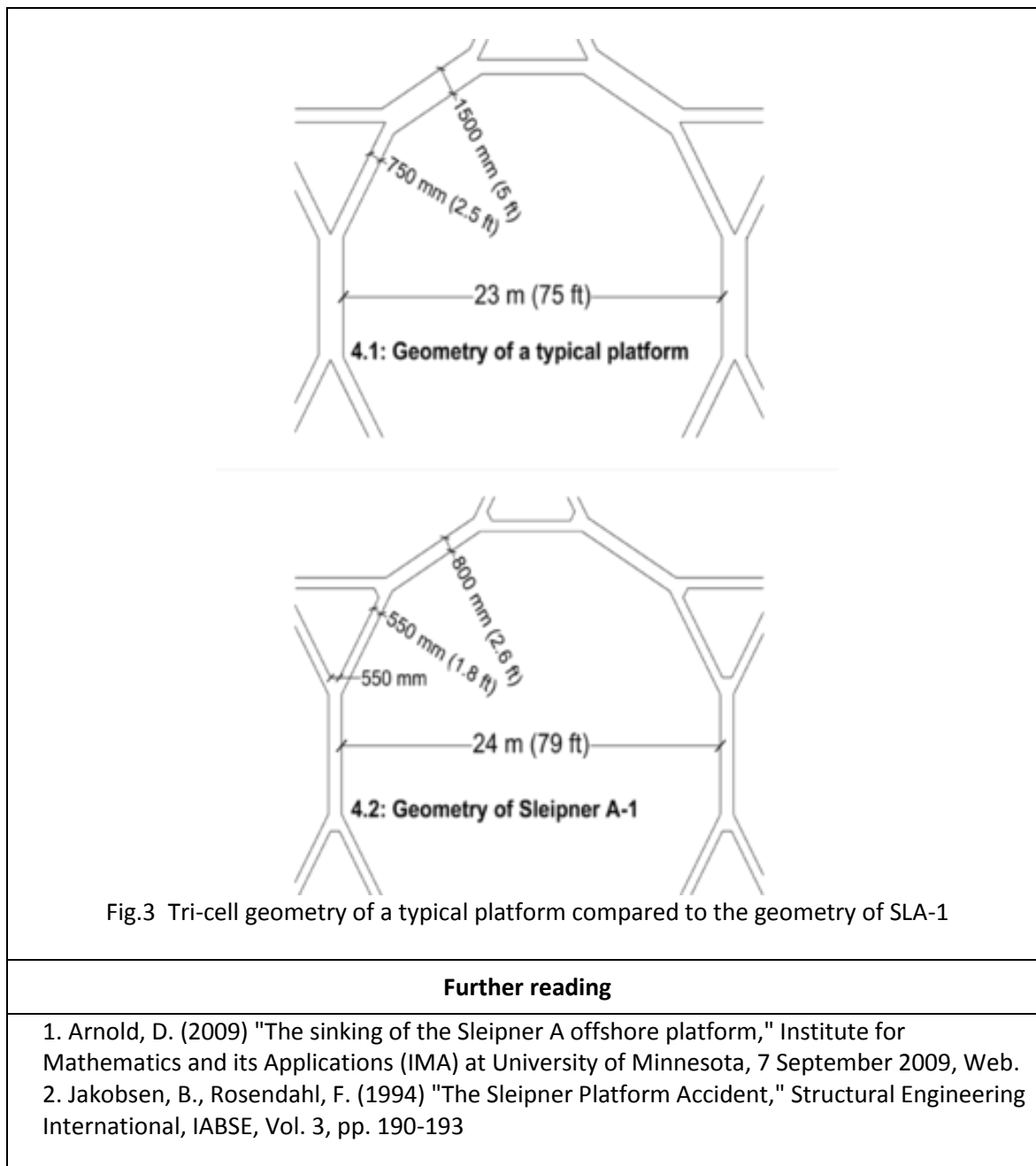Fig. 1. Rendering of a typical Condeep platform

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

4. Marine Equipment Failure

4.1 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 31st 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 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.

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.

The investigation found various contributing factors that have caused the accident:

- Same type of cranes exhibited similar issues, resulting in the crane manufacturer issuing a Technical information bulletin
- Improper usage of the crane (dragging cargo with the crane represents an overload for the used 20mm diameter heel bolts)
- Poorly executed inspection & maintenance requirements/recommendations procedures by the crew
- The crane producer Surveyor also did not follow inspection requirements and recommendations fully
- 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 31st 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   - The importance of adequate information circulation and feedback information in and from exploitation   - 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

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4.2 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:	-
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 Visual examination Fractographic analysis Scanning electron microscopy examination Numerical analysis 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. 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.

References

1. Goran Vukelić, Failure study of a cracked speed boat steering wheel, Case Studies in Engineering Failure Analysis 4 (2015) 76–82

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

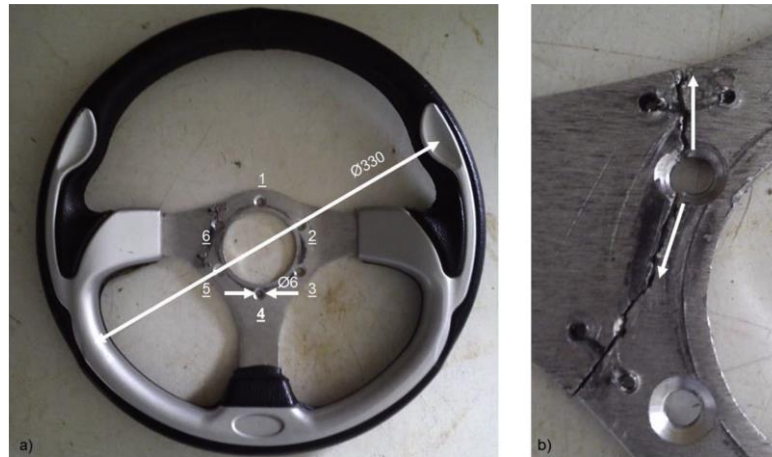- The importance of adequate maintenance procedure
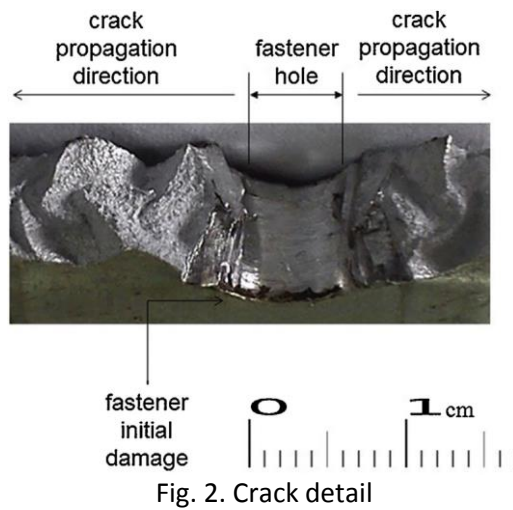
Figures

Fig. 1. Speed boat steering wheel

Fig. 2. Crack detail

Further reading

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