Marine Safety Investigation Unit

Transport Malta

# SAFETY INVESTIGATION REPORT

**201711/034** REPORT NO.: 22/2018 November 2018



## MV *BOZDAG*
**Deck slewing crane failure in the port of Tallinn 28 November 2017**

## SUMMARY

A regular, five-yearly compulsory test on the ship's deck slewing crane was planned for 28 November 2017. Given that the safe working load (SWL) was 10 tonnes, the test was planned to 'overload' the crane by 25%. The test plan necessitated the lifting of two large bags, with about 10 tonnes of water.

During the course of the test, it was observed that the cable seemed to be slipping, with the weight dropping to between 1.5 m to 2.0 m. It was recalled that the crew operating the deck slewing crane attempted to lift the bags of water again but suddenly, at around 0958, the weight dropped by a further 1.5 m to 2.0 m. It was during this time that a very heavy impact noise was heard and the crane's jib collapsed and rested against the bulwark on the port side.

Two crew members, who were inside the deck slewing crane's cabin, were seriously injured.

The Marine Safety Investigation Unit has issued one recommendation to the Company designed to ensure adequate maintenance to deck slewing crane.


MT *Bozdag*

## FACTUAL INFORMATION

### Vessel

MT *Bozdag*, a 13,815 gt product tanker, was registered in Malta[1]. She was owned by Pal Shipping-6 Company Limited and managed by Palmali Gemicilik Ve Acentilik A.S. (Turkey). The vessel was built by Admiralteyskiy Sudostroitelnaya Zavod, Russia in 2002 and was classed by the Russian Maritime Register of Shipping (RMRS).

*Bozdag* had a length overall of 157.42 m, a moulded breadth of 24.5 m and moulded depth of 13.40 m. She had a summer draught of 9.8 m, corresponding to a summer deadweight of 19,800 tonnes. The vessel was fitted with a deck slewing crane, used to work hoses at the vessel's manifold area on the main deck.

Propulsive power was provided by a 6-cylinder 6S50MC-C, slow speed, direct drive diesel engine, producing 8,580 kW at 127 rpm. This drove a single fixed pitch propeller, to reach a service speed of 15 knots.

### Crew

*Bozdag*'s Minimum Safe Manning Certificate, issued by the flag State Administration, required a crew of 13. At the time of the accident, the vessel had a crew complement of 19, mostly Russian and Azerbaijani nationals. The crew compliment included the master, chief officer and chief engineer, two OOW (deck) and three engineers. The deck ratings included a bosun, a pumpman, three able seafarers (ABs), four motormen, a cook and a steward.

### Injured crew members

One of the injured crew members was the third engineer. At the time, he was 31 years old. The third engineer had joined the vessel one month before the accident happened. This was his fourth contract with the Company as a third engineer. In general, his duty was to carry out maintenance operations on the deck slewing crane.

The other injured crew member was the bosun, who was 50 years old.

### Deck slewing crane

The DK 160-10T-18M deck slewing crane was fitted with a hydraulic drive and was located in way of frame 50 on the vessel's centreline to reach both the port and starboard cargo manifolds.

Crew members reported that the deck slewing crane was seldom used and its main purpose was to lift the cargo hoses during STS operations and to shift the gangway. It was estimated that the maximum load during these operations would not exceed three metric tonnes.

Prior to the five-yearly mandatory test, the hydraulic motor had been ashore for repairs between 28 September and 13 November.

### Environment

At the time of the accident, the weather was cloudy with a Southeasterly moderate breeze. The air and sea temperature were recorded at 4 °C. During the test, no swinging of the load was observed as a result of the weather conditions.

### Narrative

*Bozdag* had arrived at Tallinn dry-docks on 21 August 2017.

A regular, five-yearly compulsory test[2] was planned for 28 November 2017. Given that the safe working load (SWL) of the deck

[1] The vessel was deleted from the Maltese Register of Ships on 07 November 2018.

[2] The last test prior to the accident was carried out in Riga, in 2012.

slewing crane was 10 tonnes, the test was planned to 'overload' the deck slewing crane by 25%, *i.e.*, a total load of about 12.5 tonnes.

The test plan necessitated the lifting of two large bags from the quay, which then had to be filled with 10 tonnes of water. The deck slewing crane was made ready for the operation, with the boom swung overboard and the hook lowered.

The deck slewing crane had to be tested in all operating modes. During the course of the testing period, two service engineers were on site. All personnel involved were briefed on the Company's deck slewing crane operating procedures. The necessary 'Lifting Gear Prior Use' checklist and an 'Inspection and Maintenance Report' were also compiled. Relevant crew members had VHF radios to communicate among each other during the tests.

Together with the chief officer, the necessary tests were discussed. The chief officer was responsible on deck and the bosun was designated to operate the deck slewing crane. The third engineer was also requested to start the hydraulic oil heating, operate the deck slewing crane without load and inspect and control the works on the deck slewing crane mechanism and hydraulic system.

At around 0950, testing was commenced without any load. The deck slewing crane's boom was lowered and slewed in all directions. No issues were noticed with regards to the movement of the deck slewing crane and its operation.

In order to satisfy one of the testing procedures, which necessitated that the suspended load is stopped in mid-air, the bags full of water were now lifted high above the quay. During this test, at one point in time, it was observed that the cable seemed to be slipping and the weight dropped between 1.5 m to 2.0 m. It was recalled that the crew operating the deck slewing crane made an attempt to lift the bags again but suddenly, at around 0958, the weight dropped further by a distance of between 1.5 m to 2.0 m.

By this time, the bags were about 2.0 m above the quay. It was during this time that a very heavy impact noise was heard and the deck slewing crane's jib just collapsed and rested against the bulwark on the port side (Figures 1 and 2). The test had not yet been carried out with 125% load but a 100% (10 tonnes) load test was being applied.

**Figure 1: The collapsed deck slewing crane**

**Figure 2: Parts of the slewing bearing following the collapse**

The accident was witnessed by a number of persons and medical assistance was immediately requested, fearing that crew members inside the deck slewing crane's cabin may have been seriously injured. A closer inspection revealed that this was the case, with the bosun being found on top of the third engineer.

It was immediately evident that the bosun was in injured and even complaining of chest pains. The third engineer was unconscious and on the medical team's instructions, he was not moved until further medical assistance arrived. Eventually, the bosun was pulled out of the cabin and at approximately 1012, medical assistance arrived on board. At 1115, both crew members were transferred to the local hospital for further assistance and medical care.

## ANALYSIS

### Aim

The purpose of a marine safety investigation is to determine the circumstances and safety factors of the accident as a basis for making recommendations, and to prevent further marine casualties or incidents from occurring in the future.

### Cooperation

During the course of this safety investigation, the MSIU received all the necessary assistance and cooperation from the Estonian Safety Investigation Bureau.

### Dynamic loading

When a load is applied to a structure, an unavoidable vibratory effect is generated on the structure itself. This phenomenon has a general nature, irrespective of the type of structure, which, however, would in turn affect the magnitude of the dynamic load. With lifting machinery (including cranes), such phenomenon is crucial, given that these loads may compromise the structural integrity, leading to catastrophic failure[3].

Dynamic loading can cause, in general, failure when phenomena such as, maximum stress, buckling, fatigue and equilibrium of the structure itself, are exceeded. One should also take into consideration the severity of these loads since they can easily exceed the safety factor of the lifting machinery. Dynamic loading also increases the number of stress cycles which are exerted on the deck slewing crane structure and which could, in turn, be detrimental and lead to premature structure failure (*i.e.*, reduced the life time of the structure due to internal structural stresses as a result of cyclic fatigue forces with limited amplitude[2].

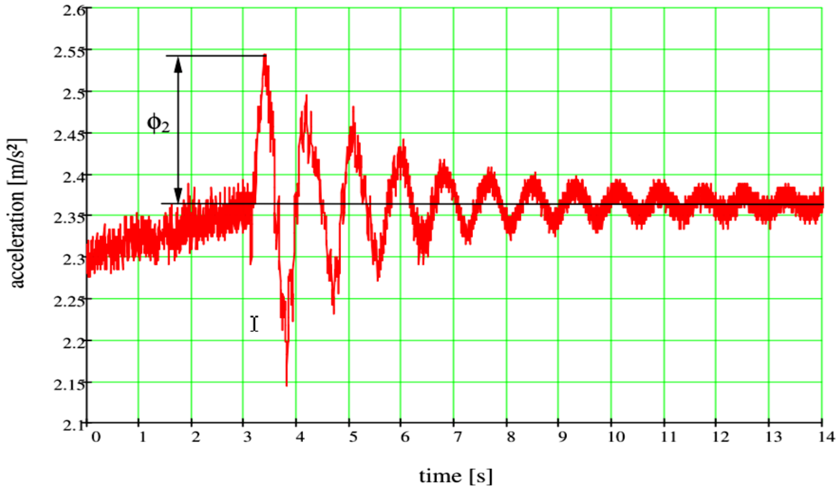


**Figure 3: An example of a typical dynamic overload due to stoppage of lifting motion**

[3] L. Solazzi, G. Incerti, and C. Petrogalli, 'Estimation of the dynamic effect in the lifting operations of a boom crane', *Proceedings - 28th European Conference on Modelling and Simulation, ECMS 2014*, vol. 8, 2014.

H. Pu, X. Xie, G. Liang, X. Yun, and H. Pan, 'Analysis for dynamic characteristics in load-lifting system of the crane', *Procedia Engineering*, vol. 16, pp. 586–593, 2011.

As it can be seen in Figure 3, most of the oscillations have peak amplitudes during the beginning or at the end of the motion of the load. Such oscillations, in turn, would increase the load, which acts on the structure itself. The symbol $\emptyset_2$ in the figure represents the dynamical overloading.

In order to further apprehend the forces and damping present in such conditions, a schematic diagram is shown in Figure 4.

Figure 4 indicates that two degrees of freedom are allowed due to the vertical motion of the load and rotational motion of the drum. The vertical motion of the load is represented by the linear displacement variable $x_1$. The displacement of the structure due to oscillations is represented by the linear displacement variable $x_2$.

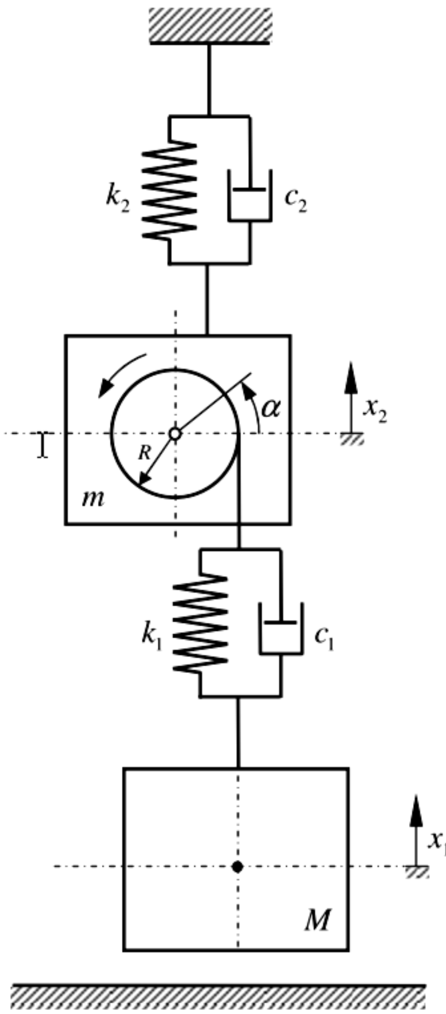


**Figure 4: Schematic diagram of loading during lifting/lowering motion**

'M' and 'm' represent the masses of the load and the lifting structure respectively. Due to the oscillatory motion, which causes vibrations in both the deck slewing crane structure and the cable, a damping and stiffness constant are added to both structure and cable, where 'k' represents the stiffness values and 'C' the damping factor. '$C_1$' and '$C_2$' represent the relevant values for the cable and the deck slewing crane structure respectively[4].

The rotational motion is represented by the rotational acceleration shown as $'\alpha'$ and the drum radius 'R'. The force due to gravity is omitted from the diagram since such force would be overcome by the elastic reactions due to static deformations of the system. With the sign convention as described in Figure 4, the following equations apply for the forces acting during such motion[3]:

$$M\ddot{x}_1 + c_1(\dot{x}_1 - \dot{x}_2) + k_1(x_1 - x_2) = R(k_1\alpha + c_1\dot{\alpha})$$

and

$$M\ddot{x}_1 + m\ddot{x}_2 + c_2\dot{x}_2 + k_2x_2 = 0$$

From both equations, it can be observed that the force exerted by the rotational motion of the drum (which in turn would have a stiffness and damping effect on the cable), is taken as the direct force acting against the forces due to the linear acceleration of the mass being lifted and the oscillatory damping and stiffness forces of the cable relative to the oscillation of the drum.

4 L. Solazzi, G. Incerti, and C. Petrogalli, 'Estimation of the dynamic effect in the lifting operations of a boom crane', *Proceedings - 28th European Conference on Modelling and Simulation, ECMS 2014*, vol. 8, 2014.

H. Pu, X. Xie, G. Liang, X. Yun, and H. Pan, 'Analysis for dynamic characteristics in load-lifting system of the crane', *Procedia Engineering*, vol. 16, pp. 586–593, 2011.

It may be further submitted that the second equation clearly points out that the forces of both masses and the oscillatory forces of the drum are repelling each other. To this effect, significant forces are generated, with the drum structure absorbing the oscillatory forces along with the linear force generated due to acceleration of the load[5].

**Sudden halt of a free falling object**

During the process of free falling, the potential energy (PE) due to gravity of an object is converted into kinetic energy (KE) (Figure 5)[6].

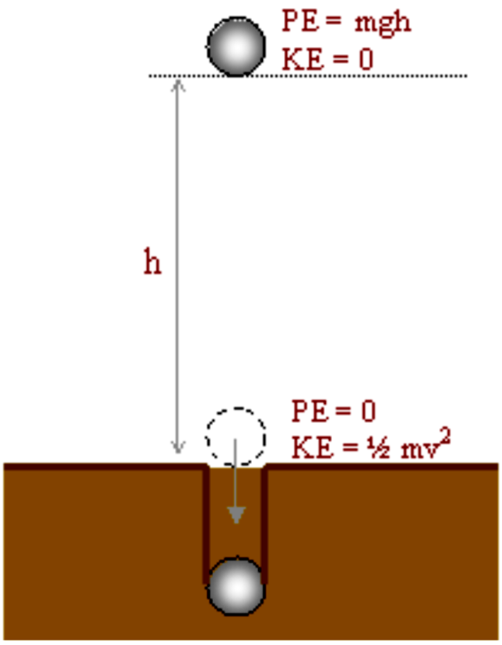


**Figure 5: Diagram showing the impact force from a free falling object [4]**

If a mass 'm', which is free falling vertically with only the gravitational pull acting on it, the initial kinetic energy is zero. As the object hits the ground, the potential energy is zero[5].

Therefore, assuming no loss of energy, the initial PE is equal to the final KE, or

$$mgh = \frac{1}{2}mv^2$$

and, the impact velocity just before the impact can be taken as:

$$v = \sqrt{2gh}$$

Applying the work and energy principle, the change which is occurring in the KE of an object is equal to the net work done on the object, *i.e.*,

$$W = \frac{1}{2}mv_f^2 - \frac{1}{2}mv_f^2$$

For a linear collision, the total work done is equivalent to the average force of impact multiplied by the amplitude of linear displacement during period of impact. Therefore,

$$Average\ impact\ force \times Distance\ traveled = Change\ in\ kinetic\ energy$$

which would lead to;

$$Total\ work\ done = Kinetic\ energy\ just\ prior\ hitting\ ground$$

The impact force can be calculated with

$$F_{avg} = \frac{{}^{1}/_{2}\, mv_f^2}{d}$$

where, 'd' is the distance travelled after the halting of the object.

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5 L. Solazzi, G. Incerti, and C. Petrogalli, 'Estimation of the dynamic effect in the lifting operations of a boom crane', *Proceedings - 28th European Conference on Modelling and Simulation, ECMS 2014*, vol. 8, 2014.

H. Pu, X. Xie, G. Liang, X. Yun, and H. Pan, 'Analysis for dynamic characteristics in load-lifting system of the crane', *Procedia Engineering*, vol. 16, pp. 586–593, 2011.

6 R. Nave, 'Impact Force from Falling Object', *HyperPhysics*, 2018. [Online]. Available: https://bit.ly/1sdGwUt. [Accessed: 27-Apr-2018].

LivePhysics, 'Solve problem related to impact force from falling object', 2018. [Online]. Available: https://bit.ly/1nlia9v.

Therefore, for a two metre drop, the estimated generated force is:

$$v_f = \sqrt{2 \times 9.81 \times 2} = 6.26 \text{ m/s}$$

and,

$$F_{avg} = \frac{{}^{1}/_{2}\,(10000)6.26^2}{0.05} = 3{,}918{,}760 \text{ N}$$

$$F_{avg} = 399.47 \text{ tonnes}$$

(the extra distance travelled after halting was taken as 0.05 m)[7].

**Multi-plate brake system**

The brake assembly for the deck slewing crane was received by the MSIU from the vessel. The extent of corrosion and wear on the bolts made it virtually impossible to disassemble the brake system unless destroying the casing. To this effect, the brake assembly had to be cut open so that the internal parts could be disassembled carefully and inspected.

The braking system, which was fitted on the deck slewing crane's hydraulic winch system, was a multiple disk brake system, represented schematically in Figure 6.

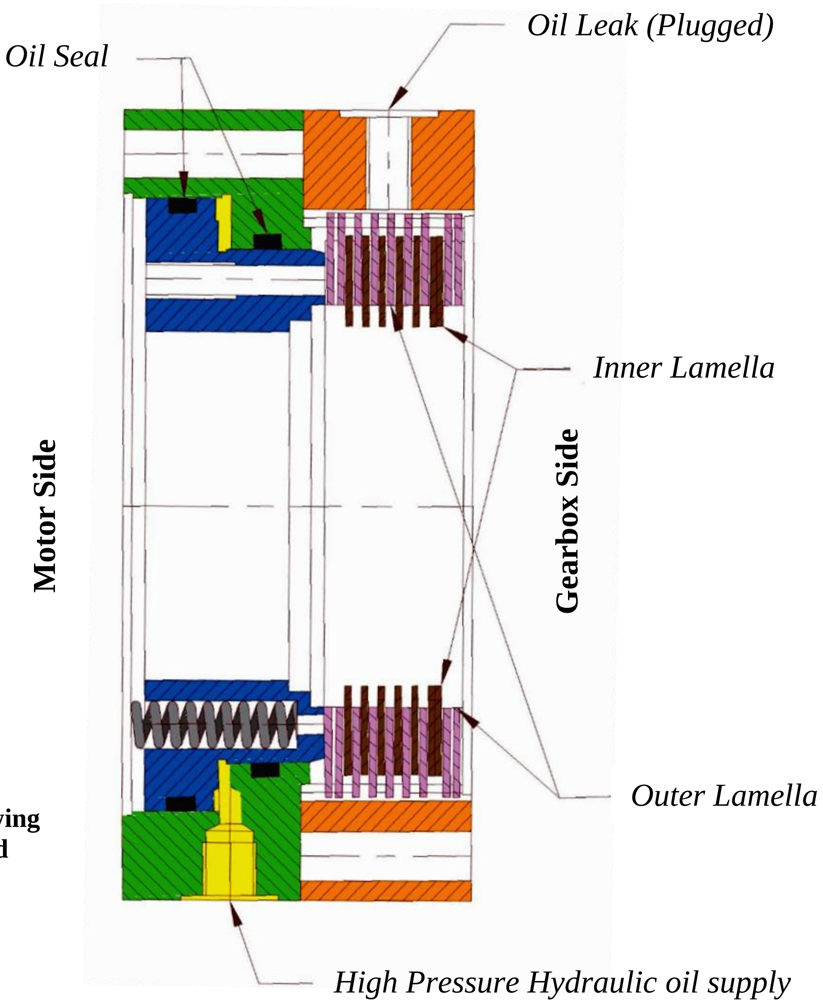


**Figure 6: Schematic drawing of the brake assembly and hydraulic supply**

[7] R. Nave, 'Impact Force from Falling Object', *HyperPhysics*, 2018. [Online]. Available: https://bit.ly/1sdGwUt. [Accessed: 27-Apr-2018].

LivePhysics, 'Solve problem related to impact force from falling object', 2018. [Online]. Available: https://bit.ly/1nlia9v.

The system comprised of a high pressure hydraulic actuating system (blue shading). The yellow shade shows the supply of hydraulic oil. The braking system was set to be normally engaged and by an increase in the hydraulic pressure, the actuator was pushed towards the motor side, compressing the springs (16 in total), which would return the actuator to its initial position when the pressure of the hydraulic was relieved.

Two oil seals on both ends of the actuator were fitted to hold any hydraulic oil from leaking either to the disk brakes' housing or the motor. A plug (marked as 'oil leak' in Figure 6 and shown in Figure 7), was also fitted inside the brake housing, serving as a tell-tale, should there be an oil leak as a result of an oil seal failure. The tell-tale was not open to atmosphere but plugged, thereby ensuring that dirt and water ingress inside the brake housing was prevented as this would have otherwise compromised the brake's performance.

**Figure 7: Plugged tell-tale (after cleaning of the brake housing)**

An initial inspection of the brake assembly when it was first received revealed multiple layers of paint. The housing itself had significant levels of corrosion with flakes of material chipping away during the cleaning process (Figure 8).

If the plug is not checked on a regular basis, any hydraulic fluid which may have leaked past the inner oil seal would go unnoticed from the exterior of the disk brake system.

**Figure 8: Layers of paint and corrosion**

Following disassembly, the internal inspection of the brake system revealed two main problems. It was immediately evident that the inner and outer lamellas (Figures 9 and 10) were 'smeared' with hydraulic fluid.

**Figure 9: Outer lamella (nine in total)**

**Figure 10: Inner lamella (eight in total)**

The safety investigation did not exclude that due to a warn oil seal on the high pressure hydraulic actuating system[8] (Figure 11) the brake system could have been contaminated with hydraulic fluid and when the winch was operated (and the hydraulic pressure is therefore increased and the brake discs are pushed apart from each other) the brake lining surfaces became exposed and contaminated with hydraulic fluid, hence compromising the braking capacity of the system, when the brake engaged again as soon as the hydraulic pressure was relieved.

As the hydraulic fluid contaminated the brake surfaces, it would have formed a coating which had a coefficient of friction that was extremely low when compared to the designed coefficient of the disk brake compound.

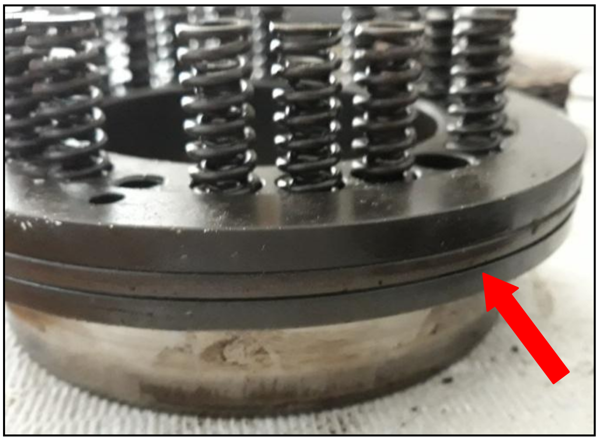

**Figure 11: Oil seal fitted on the high pressure hydraulic actuating system**

Once disassembled, it also appeared that the wear was uneven on at least three of the outer lamella sections. Figure 12 shows the innermost outer lamella. Whilst the safety investigation could not establish the reason behind the uneven wear, it was not excluded that the friction faces were not 100% in contact, thereby also compromising the braking capacity of the system.

[8] Shaded in blue in Figure 6.

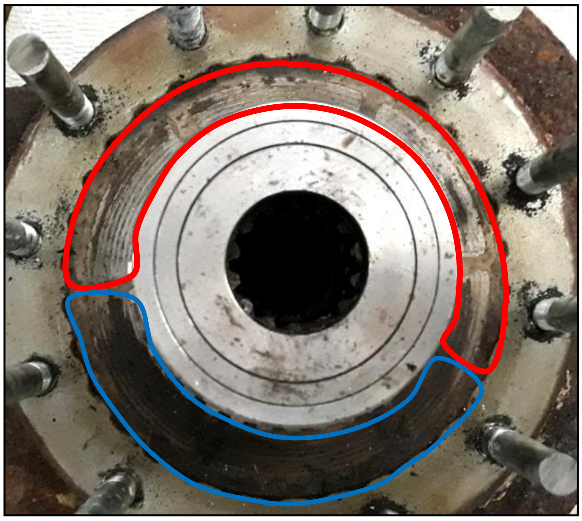

**Figure 12: Uneven pressure marks on the innermost outer lamella friction surface**

## Course of events hypothesis

The safety investigation was of the view that during the testing operation, both the dynamic loading and the sudden halt of the free falling loads were two major contributing factors to the failure of the deck slewing crane.

It was not excluded that the slipping of the brakes when the load was lifted could have had a direct correlation with the dynamic loads, the possible hydraulic leak and the uneven contact pressure as previously described. Since the load was halted in mid-air, unavoidable oscillation and forces (already described) could have overcome the frictional resistance of the (compromised) braking system, which in turn slipped but regained friction again[9].

Possible hydraulic fluid contamination on the brake liner could have further reduced the frictional coefficient between the disks and plates which, in combination with uneven pressure exerted on the outer lamella

[9] Leaking hydraulic oil, which would have sprayed onto the brake disks during rotation, would have caused slipping when the brake is engaged. Heat would burn out the hydraulic fluid, thus exposing fresh brake disk surfaces to regain braking friction. This would have taken few seconds to achieve.

material, would have further reduced the force that dynamic loads would need to overcome to force the load to drop.

As the load dropped by about two meters the first time, the initial free fall force would have acted as 'the load' which would suddenly be halted and which in turn, would have caused the sudden force along with continuous vibrations synonymous to sudden halting. This combination of sudden force and oscillation would have been enough to overcome the friction force in the brakes, taking into consideration the potential contamination with hydraulic fluid, causing a second drop of two meters.

During both the initial halting of the load in mid-air and the initial drop, the deck slewing crane structure would have absorbed part of the load as explained in the second equation of the dynamic loading and which could have potentially caused significant internal stresses on the slewing bearing assembly.

It is expected that since the load was transferred through the drum, the amplitude of the forces being exerted onto the deck slewing crane structure would have been slightly damped by the failing brakes. During the second drop, the amplitude of oscillation and dynamic loading would have been larger than those experienced during the first drop. In this case, the forces would have been superimposed, causing a degree of increase in amplitude of forces[10].

Finally, the combination of the increase in dynamic loading and the force due to sudden halting, would have been enough to cause the structure to fail before the braking system would fail again. This would cause the weakest load bearing point (which in this case was the slewing bearing) to fail, forcing the upper deck slewing crane structure to fall on the deck.

**Maintenance regime for the deck slewing crane**

A maintenance programme was available on board, which was over and above the five year compulsory testing. The programme, which totalled 19 items, was divided into three sections with maintenance tasks to be carried out:

1. every 100 hours of operation or two months;
2. every six months; and
3. every year.

The safety investigation noticed that the brake assembly inspection / testing had not been included in the maintenance programme. Moreover, there was no reference to the tell-tale opening on the assembly and any leaking hydraulic oil went undetected. The amount of flaking corrosion, layers of paint on the housing and rounded Allen bolt heads suggested that the brake assembly had not been opened for several years.

It is the view of the safety investigation that the checklist would have become an established work routine with respect to the maintenance of the deck slewing crane and therefore, one maintenance period after the other, the brake system was neither checked nor inspected.

It has to be acknowledged, however, that a thorough inspection of the brake system would necessitate the disassembling and boxing up again of the entire unit. It would have been therefore more probable that such a task is included in either the annual or the five-yearly maintenance programme.

In the absence of possible tests, which could be carried out on the slewing bearing, the

[10] During the first drop and braking, the structure and bearing undergo a shock load and start vibrating. When the load drops again and stops, another shock load and vibrating forces are transferred to the structure, through which there would have been already the effects of the previous shock load and vibrating forces caused by the first sudden drop.

safety investigation could not determine whether this was also contributory to the failure of the deck slewing crane structure. However, it was noticed that the vessel did not have a detailed procedure on how to carry out rocking tests and neither were there any dedicated record sheets for the results and comparative analysis of the readings over time.

## CONCLUSIONS

1. The dynamic loading and the sudden halt of the free falling loads were two major contributing factors to the failure of the deck slewing crane;
2. The estimated force, generated by the sudden drop of the weights, was in the region of 400 tonnes;
3. It was not excluded that the mechanical brake slipped as a result of hydraulic fluid leakage past the oil seals and less than optimal contact between the friction surfaces of the outer and inner lamella parts;
4. The brake assembly inspection / testing had not been included in the maintenance programme;
5. There was no reference to the tell-tale opening on the maintenance programme and any leaking hydraulic oil went undetected.

## RECOMMENDATIONS[11]

Palmali Gemicilik Ve Acentilik A.S. is recommended to:

***22/2018_R1*** Review its maintenance programme for the deck slewing crane and ensure that procedures are included for the rocking test of the slewing bearing and the inspection of the braking system assembly.

[11] **Recommendations shall not create a presumption of blame and / or liability.**

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|---|---|
| **SHIP PARTICULARS** | |
| Vessel Name: | *Bozdag* |
| Flag: | Malta* |
| Classification Society: | Russian Maritime Register of Shipping |
| IMO Number: | 9194012 |
| Type: | Product tanker |
| Registered Owner: | Pal Shipping-6 Company Limited |
| Managers: | Palmali Gemicilik Ve Acentilik A.S. |
| Construction: | Steel (Double hull) |
| Length Overall: | 157.42 m |
| Registered Length: | 149.20 m |
| Gross Tonnage: | 13,815 |
| Minimum Safe Manning: | 13 |
| Authorised Cargo: | Liquid in bulk |
| **VOYAGE PARTICULARS** | |
| Port of Departure: | Mongstad, Norway |
| Port of Arrival: | Tallinn, Estonia |
| Type of Voyage: | International |
| Cargo Information: | In ballast |
| Manning: | 19 |
| **MARINE OCCURRENCE INFORMATION** | |
| Date and Time: | 28 November 2017 at 09:58 (LT) |
| Classification of Occurrence: | Serious Marine Casualty |
| Location of Occurrence: | In port |
| Place on Board | Freeboard deck |
| Injuries / Fatalities: | Two serious injuries |
| Damage / Environmental Impact: | Damages to the deck slewing cargo, bulwark and railings |
| Ship Operation: | Alongside / moored / repairs |
| Voyage Segment: | Arrival |
| External & Internal Environment: | Cloudy with a Southeasterly moderate breeze. The air and sea temperature were recorded at 4 °C. |
| Persons on board: | 21 |

* Deleted on 07 November 2018.