Modeling, Identification and Control, Vol. 38, No. 2, 2017, pp. 51–58, ISSN 1890–1328

Sensitivity Analysis on the Reliability of an Offshore Winch Regarding Selected Gearbox Parameters

L. Wöll G. Jacobs A. Feldermann

[1] *RWTH Aachen University, Institute for Machine Elements and Machine Design, Schinkelstr. 10, DE-52062 Aachen, Germany. E-mail:* {lwoell,jacobs,feldermann}@ime.rwth-aachen.de

Abstract

To match the high expectations and demands of customers for long-lasting machines, the development of reliable products is crucial. Furthermore, for reasons of competitiveness, it is necessary to know the future product lifetime as accurately as possible to avoid over-dimensioning. Additionally, a more detailed system understanding enables the designer to influence the life expectancy of the product without performing an extensive amount of expensive and time-consuming tests.

In early development stages of new equipment only very basic information about the future system design, like the ratio or the system structure, is available. Nevertheless, a reliable lifetime prediction of the system components and subsequently of the system itself is necessary to evaluate possible design alternatives and to identify critical components beforehand. Lifetime predictions, however, require many parameters, which are often not known in these early stages.

Therefore, this paper performs a sensitivity analysis on the drivetrain of an offshore winch with active heave compensation for two typical load cases. The influences of the parameters *gear center distance* and *ambient temperature* are investigated by varying the parameters within typical ranges and evaluating the quantitative effect on the lifetime.

Keywords: Reliability, Sensitivity Analysis, Offshore Gearbox, Active Heave Compensation

1 Introduction

Thorough system knowledge enables the designer to accurately dimension a future system and its components to achieve the demanded product lifetime. If it becomes clear, throughout the development process, that the life expectancy will not match the demanded specifications, the designer can react accordingly and adjust the design of the system and its components systematically. In general, reliability evaluations of a final manufactured product promise the highest accuracy of predicted life expectancy but also cause the highest costs. Therefore, it would be beneficial, if the lifetime could already be calculated accurately in very early development stages. Basic gear parameters such as gear ratio and number of teeth are set at the very beginning of the development of new transmissions. Other parameters, such as *gear center distance* are set initially as well, but might be subject to minor adjustments throughout a development in iterative processes and can subsequently influence the gear durability. Therefore center distance is selected as an example for the sensitivity analysis regarding the effect of gear parameters on the final lifetime. For bearings, parameters such as lubricant temperature are often roughly estimated, disregarding influences like dissipation power and environmental conditions. In order to reveal unused potentials regarding economical bearing dimen-

doi:10.4173/mic.2017.2.1

sioning, a sensitivity analysis of the *ambient temperature* is performed to determine the necessary accuracy of temperature as input for bearing lifetime calculations for the investigated system, the drivetrain of an offshore winch.

For offshore operations, the availability of the deployed equipment is crucial, since failures on open sea have a much higher impact compared to breakdowns on land. The corrective maintenance is more elaborate because spare parts have to be procured from distant locations. Additionally, the induced downtime can cause a costly extension of the whole deployment.

The drivetrain subject to investigation in this paper is part of the winch of a vessel-mounted offshore crane for subsea operations with a lifting capacity of 150 t. The deeper the payload is lowered below sea level, the more the allowed payload has to be reduced to compensate the wire's own weight. An active heave compensation (AHC) mode utilizes the winch of the crane to compensate vertical movement of the crane due to wave motion (see Figure 1). When activated,

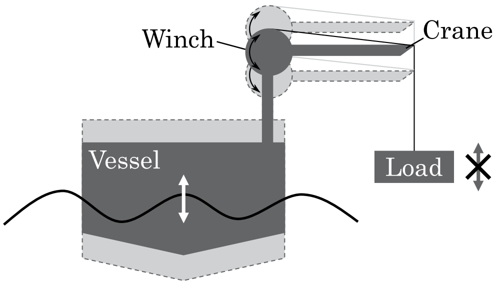

Figure 1: AHC mode

this mode induces oscillatory loads in the drivetrain. The winch drum is driven by 20 identical gearboxes, ten on each side. The gearbox itself consists of two planetary gear stages in one housing with the input connected to a hydraulic motor. The connection to the drum is implemented with pinions driving a cogwheel (see Figure 2).

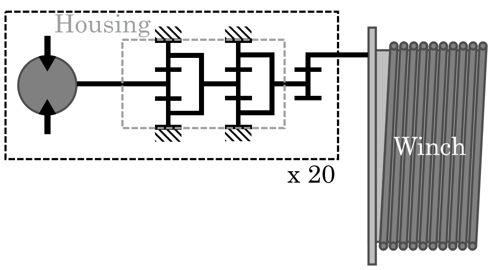

Figure 2: Structure of winch drivetrain

In the following, the reliability calculation approach will be presented, along with a short introduction into reliability basics. Thereafter, the sensitivity analyses are performed and the results are discussed.

2 Approach

The life expectancy of technical components is the result of the ratio of applied loads and load capacity. To assess the lifetime of a technical system, it is necessary to determine the stress and strength of the used components. The calculation of the components' lifetimes and the resulting system life expectancy as combination of the components' failure behavior has been performed following the approach shown in Figure 3. For further information it is referred to Neumann et al. (2016) and Wöll et al. (2017).

The lifetime of the components is calculated by comparing stress and strength of the gearbox components separately using a damage accumulation hypothesis. The strength or load capacity of the components is evaluated by available standards (Bearings: DIN ISO 281 (DIN ISO 281, 2010), Gears: DIN 3990 (DIN 3990, 1987)). The results of the load capacity calculation are usually Wöhler-curves. For the calculation of the stress, the external loads on the transmission need to known. The loads are a result of the weight of the payload and rope as well as the necessary torque to rotate the winch's inertia dynamically to compensate the wave motion. To determine the wave motion, the "Joint North Sea Observation Project" (JONSWAP) wave model has been used (Hasselmann et al., 1973). This model allows the calculation of the time functions of wave height, vertical velocity and acceleration for various wind speeds. It is based on measurements performed in the North Sea and only requires a characteristic wave height and peak frequency as an input. Both parameters are documented for different wind speeds. Based on the output, the torque, speed and angle of the winch drum can be determined, assuming the vessel with the mounted offshore crane experiences the exact vertical motions as the waves, which is an initial estimation. For converting the overall torque and speed to local loads and speeds of each component, the layout and dimensions of the gearbox and its elements are necessary, from which transfer functions can be derived. The results of this conversion are load-time and speed-time functions for each individual component. To further process this load information for the lifetime calculation, the histories are converted into load spectrums by means of counting methods. By applying a damage accumulation hypothesis on stress (load spectrums) and strength (Wöhler-curves), the damage sum for each investigated component and subsequently the expected lifetime for each component can be determined. To calculate the failure probability, the lifetime

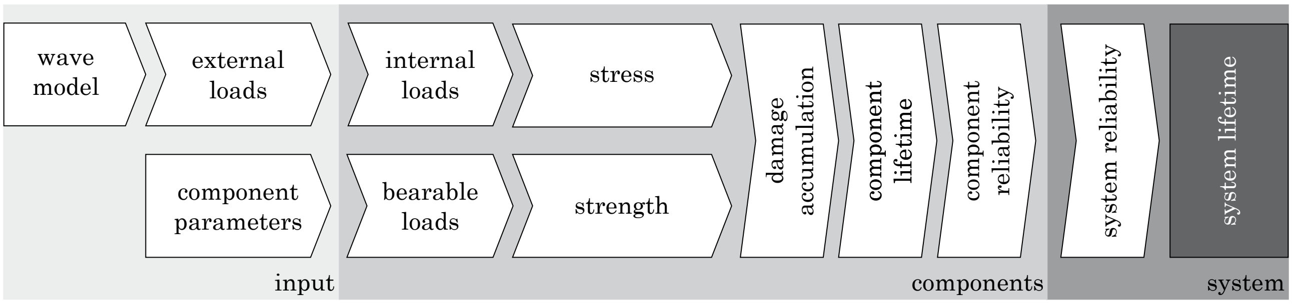

Figure 3: Reliability calculation approach

is combined with a failure distribution, obtained from tests with similar components, see Figure 4.

The failure distributions for mechanical components can be described by Weibull distributions, as given by Eq. 1.

$$F(t) = 1 - e^{-\left(\frac{t-t_0}{T-t_0}\right)^b} \tag{1}$$

The parameter t describes the characteristic lifetime, t_0 describes the failure free period at the beginning of service life and b describes the shape of the distribution.

Based on a suitable system theory, the overall reliability or failure performance of the entire system can be determined from the separate failure distribution of each component. The system theory used in this paper is the Boole System Theory. For systems in which a failure of a component means a failure of the entire system, the equation for the system failure probability is given by Eq. 2, where n stands for the number of system elements.

$$F_{sys}(t) = 1 - \prod_{i=1}^{n} (1 - F_i(t)) \tag{2}$$

Subject to the condition that the failure probability cannot exceed 100 % and is always positive, the resulting system failure probability is equal or higher than the failure probability of the weakest element. The resulting system $B10$-lifetime is the point of time after which a failure probability of 10 % is exceeded.

3 Application

The impact of changing gear dimensions might be known intuitively by tendency, but the extent is less obvious for some parameters. When designing a gearbox, first of all, the overall ratio and the structure is defined. Subsequently, the ratios of the individual gear stages are defined and a rough dimensioning of the gears is performed, based on estimations. In an iterative procedure the parameters are specified more accurately to match required specifications (Niemann and Winter, 2003).

The investigations in this paper have been performed with the assumption that the crane is operating in the North Sea with an operating depth of approximately 500 m. The ambient conditions for the wave model are set to a wind speed of Beaufort 5 or about 9.5 m/s, which is a mean value for the North Sea, according to Müller (2013). This corresponds to a significant wave height of 1.65 m and a peak frequency of 0.163 Hz (Journee and Massie, 2001). The calculations consider two different load cases, both with activated heave compensation but different payloads. In load case I a small payload (wire and payload combined: 30 t) is attached to the hook, in load case II the crane carries the maximum payload (wire and payload combined: 150 t). The respective load case is applied to the crane and its winch until the winch drivetrain fails.

An exemplary section of the resulting torque-time-history at the output of the gearbox of load case II is shown in Figure 5. The signal oscillates around the mean value which results from the static load (payload and own weight of the wire), whereas the oscillation is induced by the dynamic loads in the system due to acceleration of mass inertias.

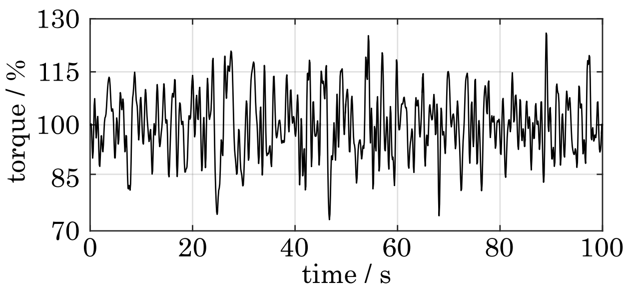

Figure 5: Torque-time history at gearbox output, Load: 150 t, Wind speed: Beaufort 5

The calculated load-time and speed-time-histories of the drum, based on the predicted wave motion is transferred to the investigated components as described before and transformed into load-revolution and speed-revolution histories to achieve the number of load alternation and their corresponding load and speed values, as described in Wöll et al. (2017). The histories are then simplified to load spectra with 64 classes for the load and 64 classes for the speed by means of the two parametric-counting method *level distribution counting*.

For gears, the failures "tooth base fracture" and "pitting" are considered and calculated individually, according to DIN 3990-2 and DIN 3990-3. Scuffing is not taken into account in this investigation since it is not researched well enough to provide a reliable lifetime estimation (Boog, 2011). Additionally, scuffing usually occurs only outside the predefined operating conditions (Renius, 1976). For the damage accumulation of pitting and tooth base fracture, the hypothesis "Miner modified (Corten-Dolan)" is used to include small stress amplitudes, since they also have a significant influence on the life expectancy. The necessary shape parameters of the failure distributions are chosen following Bertsche and Lechner (2004) and are listed in Table 1. The chosen shape parameters are referencing single teeth, since the load capacity calculation according to DIN 3990 is performed for a single tooth. Additionally, there exist failure free periods at the very beginning of the service life of some machine elements. Tests revealed that they are proportional to the $B10$-lifetime of the components (Bertsche, 1989). The available shape parameters for gears are widely scattered, since they depend on load levels and other operating condition. For the investigation conducted in this paper, a mean value, following the range provided in Bertsche and Lechner (2004) has been adapted. The shape parameters of bearings have been chosen according to Bertsche and Lechner (2004). Processing the

Table 1: Weibull parameters of failure distributions, following Bertsche and Lechner (2004)

component	cause of failure	$t_0/B10$	b
gear	tooth base fracture	0.9	1.7
	pitting	0.6	1.3
ball bearing	pitting	0.2	1.1
roller bearing			1.35

data and calculating the lifetime of the components as described above, the failure probability shown in Figure 6 can be derived for load case II. The probability of failures naturally increases for all components individually over time (solid lines). The dashed lines represent the combined failure probability of a component group. The longer the system is used and the longer the components are stressed, the higher is the probability of failure.

It can also be observed, that the failure performance of the gearbox mainly relies on the bearings which seem to be the most critical components in the investigated system.

3.1 Center Distance

The following investigation has been performed exemplary on the pinion/ring gear stage that connects the gearbox to the winch drum at the output of the second planetary gear stage, see Figure 2 and Figure 7. The center distance is varied within the range of geometric possibilities to investigate the influence of minor adjustments of the initially set center distance throughout the development process. Changing the center dis-

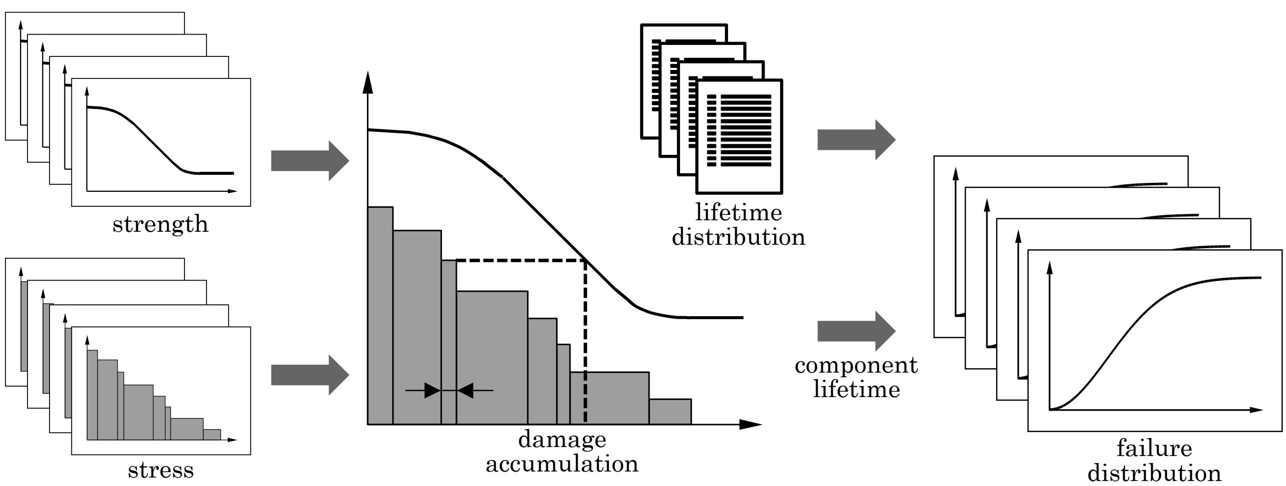

Figure 4: Process of damage accumulation for components

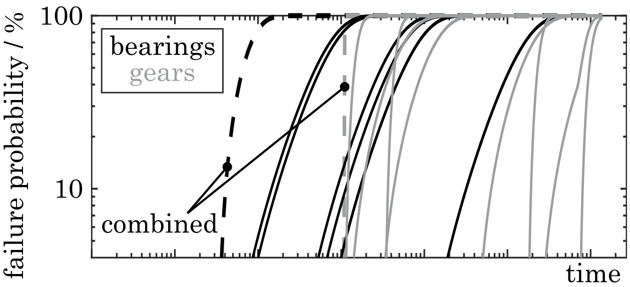

Figure 6: Failure probability of drivetrain, broken down by components,
Load: 150 t, Wind speed: Beaufort 5

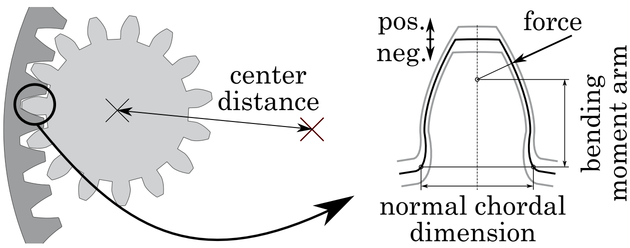

Figure 7: Center distance and profile shift

tance of a gear stage influences other gear parameters as well. However, in the course of this sensitivity analysis, the working angle, the number of teeth and the modulus are kept constant. The profile shift is adjusted to match the criteria of balanced specific sliding, which is a favorable state regarding durability (Roth, 2001). The range of investigated center distances is limited by the geometric restrictions of involute gear teeth. Due to the properties of the Wöhler-curves used for the lifetime calculation, even small changes of the load capacity induced by center distance modifications might have a large impact on the bearable load alternations and therefore on the life expectancy, see Figure 8.

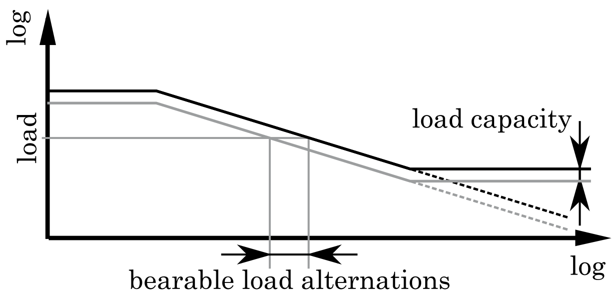

Figure 8: Sensitivity Wöhler-curve

In Figure 9, the change in the $B10$-lifetimes, calculated by the method illustrated in Figure 3, is presented for altering the center distance of pinion and ring gear (and consequently the profile shift) for load case II.

Changing the center distance affects both gears and the two causes of failure differently. Both gears are becoming less durable against pitting with an increasing center distance. The decrease in pitting lifetime arises from the increase of the Hertzian pressure in the contact due to the smaller contact radii as a result of the profile shift, causing higher stresses in the tooth flank, which leads to a facilitation of pitting. On the contrary, resistance of tooth base fracture of the pinion increases with a rising center distance, since the tooth base of external gears becomes larger for a positive profile shift (see Figure 7). For internal gears, however, the tooth base becomes smaller, leading to a decrease in lifetime for the ring gear. To reveal sensi-

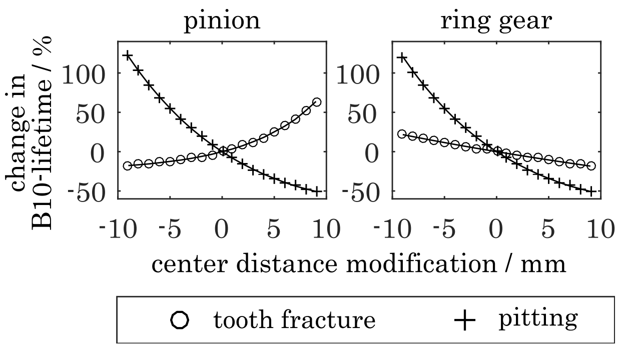

Figure 9: $B10$-lifetime changes for center distance modification,
Load: 150 t, Wind speed: Beaufort 5

tivity changes for different loads, the same simulation has been performed with a small payload (load case I). The results are illustrated in Figure 10.

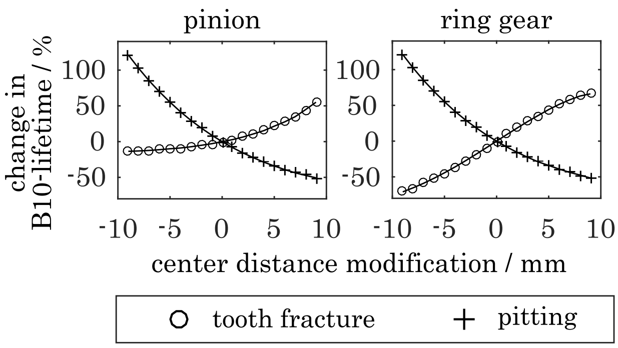

Figure 10: $B10$-lifetime changes for center distance modification,
Load: 30 t, Wind speed: Beaufort 5

It can be observed, that the sensitivity of the failure mode ”pitting” barely changes for both gears, compared to the other load case. Range and trend stay the same as in the results with 150 t load. The same observation can be made for the failure mode ”tooth base fracture" for the pinion. Regarding tooth base fracture of the ring gear, however, the trend reverses and the resistance increases for a larger center distance in the investigated example. Since the overall stresses on the gears are smaller during the load case with 30 t, there are load dependent factors in the standard DIN 3990-3, e.g. *KHalpha*, which surpass a set threshold which cause the trend of the $B10$-lifetime for ring gear tooth base fracture to invert. An explicit conclusion regarding the relation of center distance and lifetime sensitivity can therefore not be made for the gear stage subject to research.

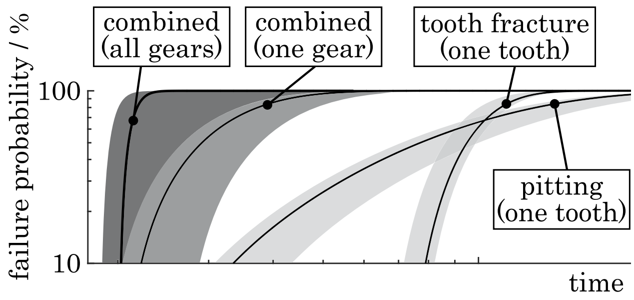

Figure 11: Failure performance variance of the pinion, broken down by failure modes,
Load: 150 t, Wind speed: Beaufort 5

To further investigate the effect of geometry modifications on the lifetime, the failure performance is shown exemplary for the reference center distance for the pinion gear in Figure 11, including variance due to modifications and how they influence the overall reliability. The reliability of a gear's tooth consists of its resistance against pitting and tooth base fracture. The failure probability of the gear's teeth due to both failure modes are displayed. It can be observed, that the probability of failure due to pitting is much higher compared to the likelihood of failure due to tooth base fracture for the investigated gearbox. The overall failure distribution per tooth results from all failure modes. As the failure of a single tooth causes the failure of the whole gear, the gear failure probability has to be calculated as a serial reliability system, with the number of teeth as number of serial elements, causing the resulting lifetime of the gear to be much lower than the life expectancy of a single tooth. For the combined failure probability of all pinions, the same procedure is applied, with the count of pinions as number of serial elements. Due to the properties of the Weibull distributions, a change of the $B10$-lifetime of one of the elements in a serial reliability structure will change the resulting system $B10$-lifetime by the same percentage.

In Figure 12 the sensitivity of combined $B10$-lifetime of all gears (pinions and ring gears) as a function of the center distance modifications is presented for both investigated load cases. The lifetime sensitivity for both load cases (30 t and 150 t) follows the same function.

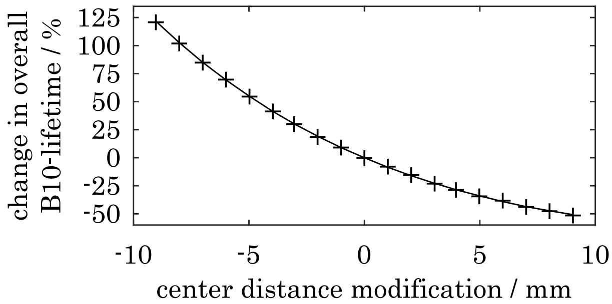

Figure 12: Combined lifetime of group "gears" for center distance modifications,
Load: 30 / 150 t, Wind speed: Beaufort 5

Hence, it can be concluded, that the lifetime sensitivity is not load dependent for the researched gearbox. In addition it can be seen, that changing the center distance can influence the lifetime by -50 % to +125 %, regardless of the applied load.

3.2 Temperature

As another critical parameter subject to investigation in this paper, the ambient temperature has been selected for analysis. Since DIN 3990 does not take temperature into account for viscosity calculation, only the impact on the lifetime of bearings according to DIN ISO 281 has been considered in this research. To determine the stationary temperature in the gearbox depending on the ambient temperature and wind speeds, a simple thermic model has been built up. For calculating the natural and forced convection, it has been assumed that the gearbox can be simplified as a horizontal cylinder in cross stream. Based on these boundary conditions, the calculations have been performed according to VDI e.V. (2013). Thermal conduction has been assumed to be negligible. The dissipation of the gears acts as the only source of heat. Only the losses of the two planetary gear stages have been considered, since the pinion/ring wheel stage is located outside the gearbox housing (see Figure 2).

The results of the calculation are shown in Table 2. The stationary gearbox temperatures (and lubricant temperature) are presented for different ambient temperatures. The combined efficiency of both planetary stages has been calculated according to Niemann and Winter (2003) to a value of 97 %, which is applied for both load cases.

For an ambient temperature range of $-20\,°C$ to $40\,°C$, which is a typical range for offshore cranes, the calculated stationary gearbox temperature, without oil

Table 2: Thermic gearbox model - results

load case	ambient temperature						
	$-20\,°C$	$-10\,°C$	$0\,°C$	$10\,°C$	$20\,°C$	$30\,°C$	$40\,°C$
I $(30\,t)$	$-6.7\,°C$	$3.1\,°C$	$12.9\,°C$	$22.7\,°C$	$32.3\,°C$	$41.9\,°C$	$51.4\,°C$
II $(150\,t)$	$19.0\,°C$	$29.2\,°C$	$39.3\,°C$	$49.4\,°C$	$59.3\,°C$	$69.1\,°C$	$78.9\,°C$

heating, is in the range of $19\,°C$ to $78.9\,°C$ for the load case with 150 t and in the range of $-6.7\,°C$ to $51.4\,°C$ for 30 t. Similar to the previous section, a sensitivity analysis has been performed to quantify the influence of changing ambient temperatures (and consequently the oil temperature) for the same two load cases.

The deviation of $B10$-lifetime from the reference temperature of $0\,°C$ with a load of 150 t is shown in Figure 13 for different ambient temperatures.

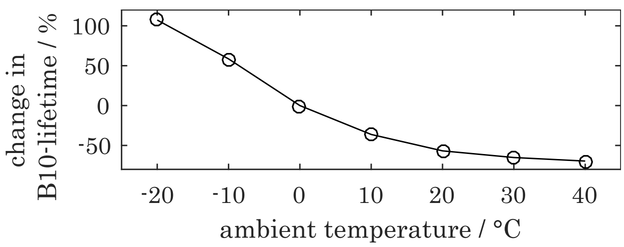

Figure 13: $B10$-lifetime changes of bearings for different ambient temperatures,
Load: 150 t, Wind speed: Beaufort 5

For this load case the $B10$-lifetime differs by -66 % to +102 % from the reference lifetime at an ambient temperature of $0\,°C$. The bearing lifetime for lower temperatures is naturally higher since the viscosity increases, causing a thicker lubricating film in the frictional contact of the bearing. It can also be observed that the influence of the temperature is significantly higher for low ambient temperatures.

As before, the investigation has also been repeated for the load case with a load of only 30 t. The results are shown in Figure 14. The $B10$-lifetimes differ from the reference lifetime by -90 % to +0 %. Compared to the load case with 150 t, the lifetime for the lower load case is slightly more sensitive for higher ambient temperatures. However, for low ambient temperatures there is no change in the calculated lifetime at all. Below ambient temperatures of $0\,°C$ fluid film lubrication with a sufficiently thick lubrication film occurs, for which the lifetime is not influenced by temperature changes anymore. The same effect will occur during the previous load case when the ambient temperatures are sufficiently low. Summarising, it can be observed, that for high ambient temperatures, the sensitivity regarding bearing lifetime is relatively small. Therefore, if the bearings were to be dimensioned for the investigated gearbox operating in a warm environment, the temperature would not have to be known very accurately since the deviation from the actual bearing lifetime would by relatively small. A similar effect can be observed for very low ambient temperatures, where the bearing lifetime reaches its maximum value, due to a sufficiently thick fluid film in the tribological contact. In between, however, the sensitivity is relatively high. Hence, for very accurate lifetime predictions, the temperature has to be known very precisely.

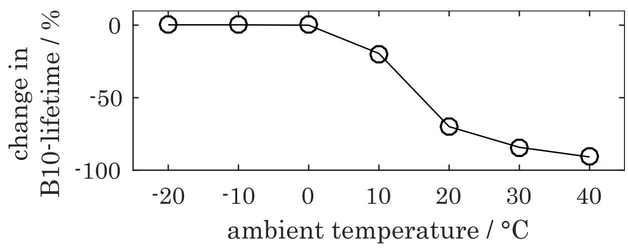

Figure 14: $B10$-lifetime changes of bearings for different ambient temperatures,
Load: 30 t, Wind speed: Beaufort 5

4 Summary

The present work investigated the sensitivity of the lifetime of a crane winch gearbox with Active Heave Compensation for offshore applications regarding selected parameters. The JONSWAP wave model has been used for load predictions, neglecting the vessel dynamics. The machine elements subject to research have been gears and bearings with the aim to estimate deviations in reliability calculations that might occur with only basic knowledge of the gearbox parameters.

For this investigation, sensitivity analyses have been performed exemplary for one selected gear parameter and one selected bearing parameter. Firstly, the center distance of a pinion/ring gear stage of the gearbox subject to research in this paper has been varied within the limits of possible tooth geometry, since the center distance might by subject to minor adjustments throughout a gear stage development process. Failure due to tooth fracture is subject to complex interdependencies which do not suggest a definite conclusion regarding the relation between lifetime sensitivity and center distance for the gear stage, investigated in this work. However, it has been shown that pitting is the primary cause of failure, whose sensitivity is also load independent. Furthermore, it has been revealed, that only by changing the center distance, the lifetime of the gear stage can be changed by -50 % to +125 %.

For bearings, the ambient temperature of the gearbox has been varied. For an estimation on how the gearbox temperature depends on the ambient temperature, a simple thermic model of the gearbox has been built up to calculate the stationary temperature induced by gear dissipation. It has been shown that the lifetime sensitivity is relatively small for very low and high ambient temperatures. In between, however, the temperature has to be known very accurately, since even small temperature changes might lead to large deviations in lifetime.

In future, the investigations regarding lifetime sensitivity need to be extended to other relevant parameters to receive a more comprehensive system knowledge and enable an accurate lifetime estimation in early development stages. Furthermore, the vessel dynamics will be considered for load applications, to investigate the influence of such models.

Acknowledgments

The research presented in this paper has received funding from the Norwegian Research Council, SFI Offshore Mechatronics, project number 237896.

References

Bertsche, B. *Zur Berechnung der System-Zuverlässigkeit von Maschinenbauprodukten*. Ph.D. thesis, Universität Stuttgart, Stuttgart, 1989.

Bertsche, B. and Lechner, G. *Zuverlässigkeit im Fahrzeug- und Maschinenbau: Ermittlung von Bauteil- und System-Zuverlässigkeiten*. VDI-Buch. Springer, Berlin, 2004. doi:10.1007/3-540-34996-0.

Boog, M. *Steigerung der Verfügbarkeit mobiler Arbeitsmaschinen durch Betriebslasterfassung und Fehleridentifikation an hydrostatischen Verdrängereinheiten*. Ph.D. thesis, KIT, Karlsruhe, 2011. doi:10.5445/KSP/1000021057.

DIN 3990. *Tragfähigkeitsberechnung von Stirnrädern*. Beuth, 1987.

DIN ISO 281. *Wälzlager – Dynamische Tragzahlen und nominelle Lebensdauer*. Beuth, 2010.

Hasselmann, K., Barnett, T. P., Bouws, E., Carlson, H., Cartwright, D. E., Enke, K., Ewing, J. A., Gienapp, H., Hasselmann, D. E., Kruseman, P., Meerburg, A., Müller, P., Olbers, D. J., Richter, K., Sell, W., and Walden, H. *Measurements of wind-wave growth and swell decay during the Joint north sea wave project (JONSWAP)*. Hamburg, 1973.

Journee, J. and Massie, W. *Offshore Hydromechanics*. Lecture Notes. Delft University of Technology, 2001.

Müller, S. *Wind measurements at FINO 2*. FINO Conference. Kiel, 2013.

Neumann, S., Wöll, L., Feldermann, A., Strassburger, F., and Jacobs, G. Modular System Modeling for Quantitative Reliability Evaluation of Technical Systems. *Modeling, Identification and Control*, 2016. 37(1):19–29. doi:10.4173/mic.2016.1.2.

Niemann, G. and Winter, H. *Maschinenelemente: Band 2: Getriebe allgemein, Zahnradgetriebe - Grundlagen, Stirnradgetriebe*, volume 2. Springer, Berlin, Heidelberg, 2003.

Renius, K. *Last- und Fahrgeschwindigkeitskollektive als Dimensionierungsgrundlagen für die Fahrgetriebe von Ackerschleppern*, volume 49 of *Fortschritt-Berichte VDI : Reihe 1, Konstruktionstechnik, Maschinenelemente*. VDI-Verl., Düsseldorf, 1976.

Roth, K. *Zahnradtechnik*. Springer, Berlin, 2 edition, 2001.

VDI e.V. *VDI-Wärmeatlas*. VDI-Buch. Springer, Berlin, Heidelberg, 2013.

Wöll, L., Jacobs, G., Feldermann, A., Neumann, S., and Strassburger, F. Influence of wave height on the reliability of an offshore winch. In G. Jacobs, editor, *Antriebstechnisches Kolloquium*, volume 17. Aachen, pages 59–70, 2017.