# Reliability Evaluation of Drivetrains: Challenges for Off-Highway Machines

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**Abstract**

Downtime of mobile machinery used in fields like construction, earthmoving or mining usually leads to an instant halt of an entire process and can even endanger entire operations. To meet the customer's requirement for high availabilities of their equipment, safeguarding the reliability of overall systems and components is necessary. Since life expectancy of systems and its components strongly depends on the experienced load history, this information needs to be available as accurately as possible to allow reliable lifetime calculation results. Due to the wide range of machines and applications of off-highway machines, determining representative loads is especially challenging. The challenges, in determining both load cycles for the entire system and local component loads, are discussed in this work, along with approaches to face them. Additionally, a method is described, which allows users to quantitatively calculate life expectancy of technical systems in both the concept phase and the later stages of the product life cycle. In the end, two examples are presented in which exemplary challenges are faced.

**Keywords:** reliability, drivetrain, off-highway, load cycles, lifetime

## 1. Introduction

The global sales for off-highway machines are over 100 billion US dollars [1] annually with a wide range of different products, which are often used in logistic processes. Although there are methods available to plan and calculate these processes in the line of supply chain management, over 70% of construction projects cannot be completed within the proposed budget and time [2]. One cause can be the unexpected downtime of construction machinery since in logistic processes the failure of one machine may result in the stop of the entire process chain. These processes include the transport to and from the work site as well as production processes which are performed using mobile machinery. Similar processes can be found in mining operations, both above and below ground and other off-road activities. Even small defects can result in high costs for the operating company and delays due to repair time [3, 4]. Therefore, reliability and availability are important factors for the optimization of projects that rely on off-highway machines.

In order to increase the robustness of processes, the easiest way is to use redundancies on either the system or the component level. However, this option results in high costs and is therefore not preferable. Instead, a detailed knowledge of the systems reliability should be created, which can minimize the total cost of ownership and allow for more robust planning. Methods for the assessment of a system's reliability are often qualitative, such as failure mode and effects analysis (FMEA) and fault tree analysis [5, 6]. These methods do not yield quantitative information about the system's reliability over time, which is necessary for incorporating reliability information in the planning process or work scheduling. For these applications, quantitative reliability evaluation methods are necessary. They can be applied throughout the entire life cycle from the early concept phase to the use of reliability models as a basis for predictive maintenance.

The challenge in determining the quantitative reliability of mobile machinery is in the multitude of influences that determine the loads on the machine, as illustrated in **Figure 1**. The same machine can experience vastly different loads when used at different worksites. The environmental variables, such as temperature, humidity and exposure to sunlight may vary. The ground material and the slope of the path have an influence on the required traction forces. The main influence is the task that is performed by the machine, which can be defined by the loads and typical cycles of movement. The loads on the machine can be transformed into a *stress* on each component. Failures occur when the component stress exceeds the components *strength*, which depends on the components design [7].

This work describes a method to assess the reliability of mobile machinery drivetrains quantitatively, on both a component and a system level. The quantitative approach offers advantages both early in the development process, when different concepts can be compared with each other based on the quantitative reliability, and also in the later stages and in the exploitation phase, when the reliability assessment can be used to enable condition-based and predictive maintenance approaches.

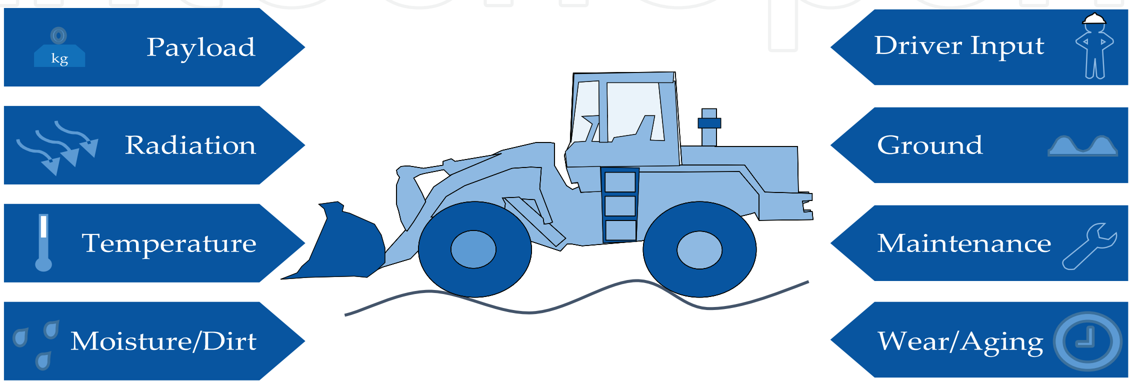


**Figure 1.** Influences on the system reliability of off-highway machines.

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The accuracy of the calculated lifetimes depends on the quality of the load assumptions, since the loads determine the damage of the components. The challenges that this crucial part of reliability evaluation creates for off-highway machines, which show a great variety in system concepts and applications, are discussed in this work. Two different systems, to which the method has been applied, will be used as examples for the application of the method and the possible ways to face the challenges in determining representative load cycles and local component loads.

## 2. Lifetime models

The system's reliability depends on the reliability of its machine elements. For many components, the lifetime can be determined through lifetime models, but not all components contribute equally to the overall system reliability. To identify relevant components in an investigated system, a qualitative reliability analysis has to be performed. The simple method *ABC analysis* [7] is well suited for this purpose, see **Table 1**.

Machine elements that are assigned to category A distinguish themselves by having a significant influence on the reliability of the system, to which they contribute. In addition to that, methods for lifetime calculation of these elements are available which provide reliable lifetime predictions. Typical components in common mechanical drivetrains belonging to category A are gears, bearings and shafts. Parts allocated to category B determine the overall life expectancy of the system as well, but their lifetime cannot be calculated as precisely compared to machine elements in category A. B components are, among others, friction clutches and seals. The third category C includes components which neither affect the system lifetime to any extent nor enable a lifetime calculation at all, for example, locking rings. The components in category C are rarely subject to failure, so they usually do not need to be considered.

Failures of components originate from an exceedance of their respective *load capacity* or *strength* by the occurring *loads* or *stresses*. Since both quantities are subject to statistical distribution, the lifetime, which is a result of the proportion of stress and strength, is statistically distributed as well. High volume testing of the investigated system and its components under realistic conditions would yield the most precise information. However, since such tests require an extensive amount of time and resources, this is rarely done during a development process. Therefore, calculation specifications to calculate the load capacity of machine elements have been developed, which are shown in **Table 2** for components in categories A and B.

| | A | B | C |
|---|---|---|---|
| Significance | High | High | Low |
| Lifetime prediction | Reliable | Unreliable | Unreliable |
| Machine elements | Gears, bearings, shafts | Friction clutches, seals, etc. | Locking, rings, etc. |

**Table 1.** Exemplary ABC analysis of mechanical drivetrain components [7, 8].

| | | | | | | | |
|---|---|---|---|---|---|---|---|
| Lifetime | Category A | Gears | DIN 3990 | ISO 6336 | AGMA 2001 | JGMA 6102 | DNV 41.2 |
| | | Shafts | FKM | DIN 743 | | | |
| | | Bearings | DIN ISO 281 basic | | | DIN ISO 281 expanded | |
| | Category B | Friction clutches | Wear | | | | |
| | | Seals | Wear | Temperature degradation | | | |

**Table 2.** Calculation specifications for machine elements.

Cylindrical gears, which are the most common gear type in drivetrains, can be designed and calculated by various available standards and norms [9–14]. These approaches consider three general kinds of failure for gears: pitting, tooth fracture and scuffing. Since scuffing prediction for gears is not advanced enough and furthermore only occurs outside of predefined operating conditions, it is currently not considered in reliability calculations [15, 16]. Flank failure due to pitting can occur on both tooth flanks on each gear tooth. While a tooth fracture ends the lifetime of a gear immediately, pitting does not. In gear-pitting testing, the threshold for pitting for hardened gears is usually 4% of the total flank area [9]. Thus, for each gear, three load capacities per tooth can be calculated and compared with the respective loads: Flanks 1 and 2 (for which the load capacity is usually the same) and the tooth base.

Roller bearings are one of the few machine elements, for which the lifetime calculation has been standardized instead of the load capacity calculation and has been made available in [17]. The basic approach only considers load and a basic load capacity determined by testing, whereas the expanded approach additionally takes lubricant viscosity (temperature adjusted) and particle contamination, as well as revolving speed into account. However, for complex motion behaviour like oscillation, the currently available methods lack accuracy and therefore cannot provide a reliable lifetime prediction [18, 19].

Calculation of load capacity of shafts and axles has been standardized and documented, for example [20, 21]. Shafts in drivetrains are exposed to oscillating stresses due to torque, axial forces and bending moments that depend on the rotation angle. While available methods are designed to prove the strength of shafts, lifetime calculation is possible but not considered to be accurate [22]. The accuracy would improve with finite-element method (FEM) calculations, but the modelling effort and computing time would increase as well.

Friction clutches are allocated in category B since there are many influences that cannot be considered for lifetime calculation yet (e.g. temperature, ageing, topology of disks). However, a rough estimation can be performed, calculating the wear of the clutch disks, proportional to the experienced frictional power and based on static friction coefficients. The clutch is considered as failed, when a predefined state of wear is reached [23].

Since the lifetime calculation of radial shaft seals is currently not possible, these machine elements belong to category B as well. The interactions of the seal with the surrounding elements are very

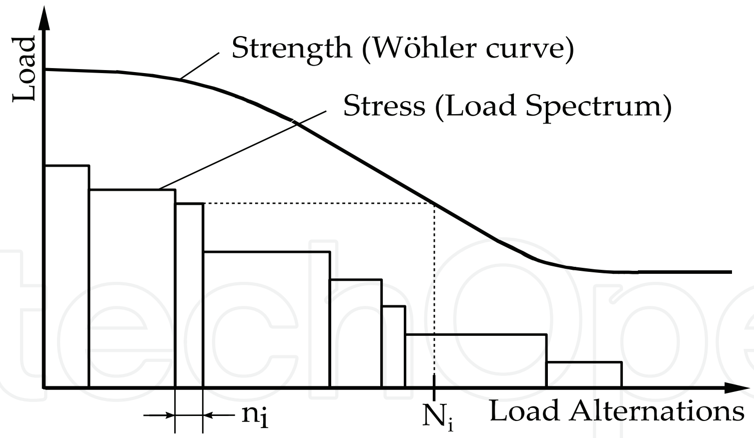


**Figure 2.** Comparison of stress and strength for damage accumulation [8].

complex and therefore hard to quantify. Radial shaft seals can fail not only due to mechanical failure modes (e.g. wear) but chemical failure modes as well (e.g. hardening) [24]. For a rough estimation of seal lifetimes, however, simple models that only consider wear or temperature degradation are available [25].

The previously described models for component strength define a Wöhler curve for each component, which represents the number of load alterations that can be endured at a certain load level. The component stress can be considered in the form of a load spectrum, which consists of the number of load alterations at a certain level the system has to endure, see **Figure 2**.

To assess the component damage caused by the loads in the load spectrum, for each load level the number of load alterations $n_i$ in the load spectrum has to be compared to the number of bearable load alterations $N_i$ from the Wöhler curve which leads to the component damage caused by the experienced loads [26]. To account for temporal distribution of failures, the common procedure is to combine the lifetime with mathematical distributions based on experience, which results in a failure probability function $F(t)$. To describe the failure probability mathematically, the Weibull distribution is commonly used. The shape of the function can be fitted to different distributions by changing the Weibull parameters $b$, $T$ and $t_0$ [7, 8, 25].

To assess the system's reliability, the component's reliabilities have to be combined. This can be done using a suitable system theory. The easiest one is Boole's system theory, which can be applied for non-repairable systems. According to Boole's theory, the system reliability for serial systems without redundancies is simply the product of all component reliabilities [27].

## 3. Method

The previously mentioned processes of component lifetime calculations and system theory are part of a superordinate method, which puts all steps into the context of a system analysis in its entirety [8]. This method is described briefly in the following, and aims to provide a guideline for the calculation of overall reliability of off-highway drive trains, see **Figure 3**.

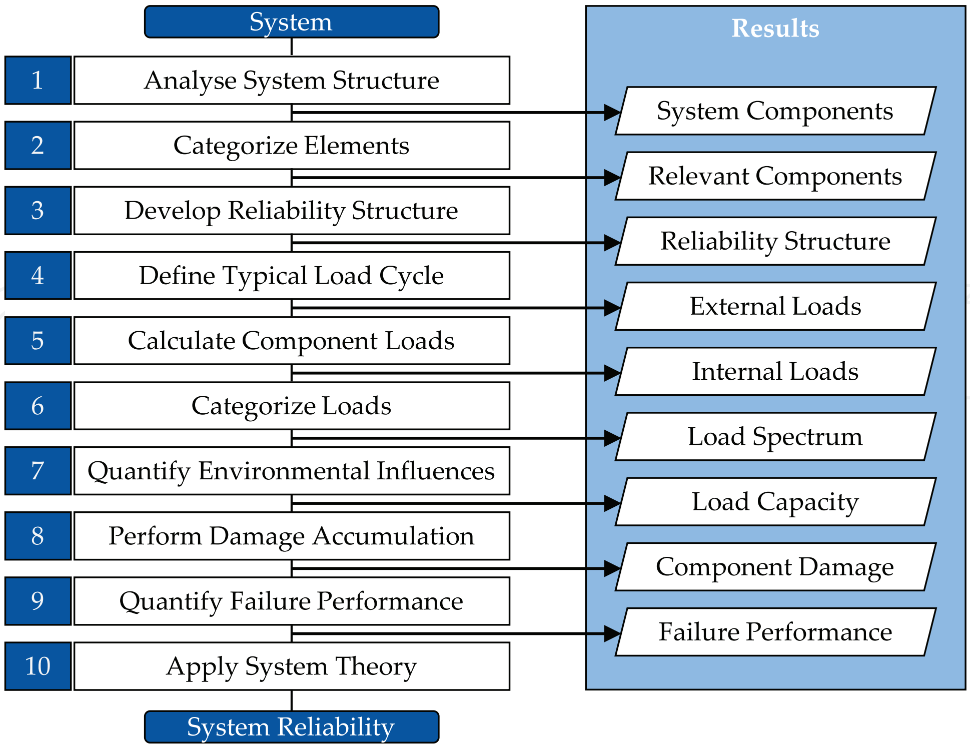


**Figure 3.** Method for quantitative reliability evaluation of drivetrains [8].

When performing a reliability evaluation of an existing system, firstly the system has to be analysed to identify all contained components and to understand how the system works (power flow, moving parts, etc.). The machine elements identified this way need to be categorized (ABC analysis). Since drivetrains usually contain similar components, this only has to be done if components are being used, which are not categorized yet. In preparation of the calculation of the system reliability, the system structure needs to be examined for redundancies. To determine the external loads on a drivetrain, a typical load cycle needs to be defined in order to allow representative calculation results. This topic, as well as the transfer to local component loads including environmental influences and load classification, is discussed in-depth in the next section. After calculating the components' lifetimes by comparing the locally occurring loads with the individuals load capacities, the lifetime and subsequently failure distributions can be obtained. Finally, the failure distributions are combined, taking into account the system reliability structure. The last step is the derivation of the overall system lifetime from the resulting system failure probability. More detailed information can be found in Ref. [8].

## 4. Challenges in load assumptions

In steps four and five of the described method, a representative load cycle and the resulting component loads have to be determined. This is especially challenging during the concept phase, when no measurements can be performed. The particular challenges for off-highway machines will be described in the following.

The term off-highway in drivetrain technology includes a wide range of fields. Typical areas are, for example, construction and mining applications, in which an equally wide range of mobile machines are utilized for various tasks with very different characteristics. Machines in open-pit mining, for example, bucket-wheel excavators or spreaders, are among the largest machines in the world and handle an enormous amount of material and are accordingly subject to immense loads. Other machinery in the general field of earthmoving includes among others excavators, loaders, graders, scrapers and dumpers [28]. Even for these types of vehicles, countless variants exist, for example, excavators, for which crawler undercarriages or wheeled types are common. Besides the technical realization of the drivetrains, other specifics, for example, boom variants (monoblock booms, adjustable booms, embankment booms, etc.), exist [29], which influence the operating behaviour and subsequently the occurring loads on structure and drivetrain. Furthermore, machines, even with same configurations, are utilized for vastly different tasks. Wheel loaders, for example, can be used for material-handling tasks (usually driving continuously in a Y-cycle), transportation of the material of larger distances, grading, material pushing or lifting work. With such a wide bandwidth of operating modes, it becomes clear that dimensioning drivetrains for mobile machines as well as determining life expectancies of such systems remains challenging due to the large variance of occurring loads, even on machines with identical configurations. Uncertain load assumptions are among the main reasons for unplanned downtime [30]. Therefore, representative load cycles as well as precise component loads as a result of the load cycles are necessary for accurate lifetime evaluations of mobile machines.

### 4.1. Load cycles

To build a representative load cycle as the basis for determining the loads on the components, a deployment scenario has to be created, including all possible tasks which the investigated machine can perform. For each task, a representative process pattern has to be defined. The approach supposed to yield most realistic results measures the workflow per task throughout an extended period of time, divided into sections for recurring activities. For the example of a wheel loader, a typical task is material handling in a Y-cycle. In this cycle, the operator fills the bucket by forwarding into a material heap, backs out of the heap, drives forward to an unloading zone, unloads, backs out of the unloading zone and restarts the cycle. Since the driving path differs each time, the path for each Y-cycle should be recorded over a sufficient period of time. Eventually, the sections are superposed and a relevant Y-cycle is created, see **Figure 4**.

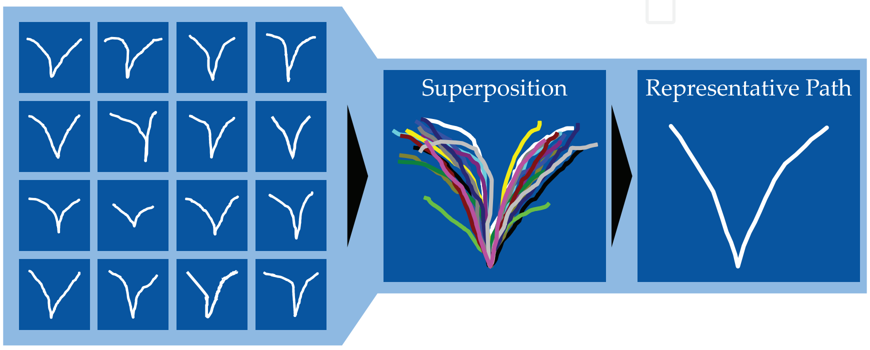


**Figure 4.** Schematic illustration of the determination of a representative driving path.

This procedure has to be performed for each identified task per machine, ideally for not one but multiple job sites, operators, the kind of handled material, ground conditions, etc. If a metrological approach is not possible due to unavailability of machines or if the investigation is performed in the product development process, the representative load cycle can be determined for a similar machine or by simulation without any measurements based on basic knowledge about the future tasks and job site, for example, topology of an open-pit mine.

With either method, a representative workflow is made available for all tasks, including necessary time-dependent load information. With knowledge about the time share of each task in the overall utilization, the representative time functions for each task can subsequently be compiled, as illustrated in **Figure 5**. The compiled load cycle serves as basis for the determination of local component loads in the system, which is discussed in the following section.

### 4.2. Component loads

Since the lifetime of the system depends on the components' lifetimes, defined by the experienced load history, the component loads have to be determined as accurately as possible. The loads can be determined either through measurements, through simulation or through a combination of both methods.

To determine component loads through measurements, either in the field or in a testing environment, extensive measuring equipment has to be applied to the components. For the lifetime calculation, values like forces, torques, rotational speeds, temperatures, etc. have to be known for each component, so that the necessary measuring equipment to capture these many values would be challenging to be incorporated into the system and also very expensive.

As an alternative to measurements, simulations can be performed to determine the component loads. Based on the global load cycle that is investigated, the component loads can be calculated by system models, which can be modelled on different levels of detail and complexity. A basic method is the torque plan, which can be used to determine torques and speeds for gearboxes without considering dynamic loads [31]. For a more detailed simulation, torsional simulations can be performed that consider dynamic effects in the rotational direction, which can be caused by acceleration, gear shift or the system's dynamic behaviour [32]. Through multibody simulations, all degrees of freedom can be considered in the load calculation.

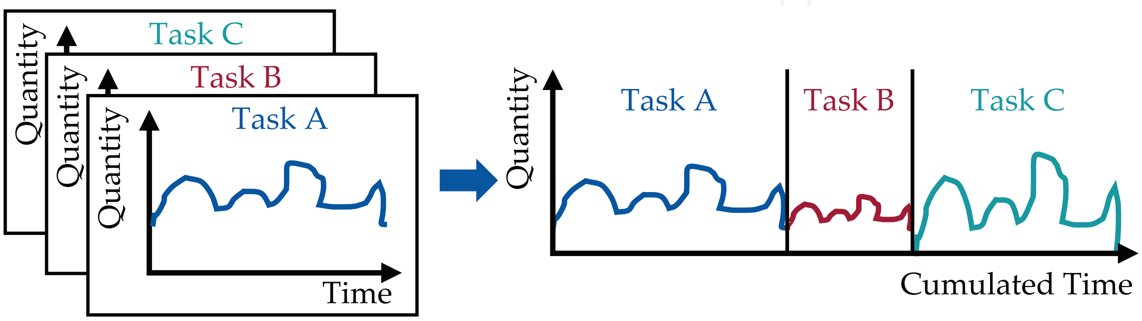


**Figure 5.** Schematic illustration of the compilation into representative load cycle.

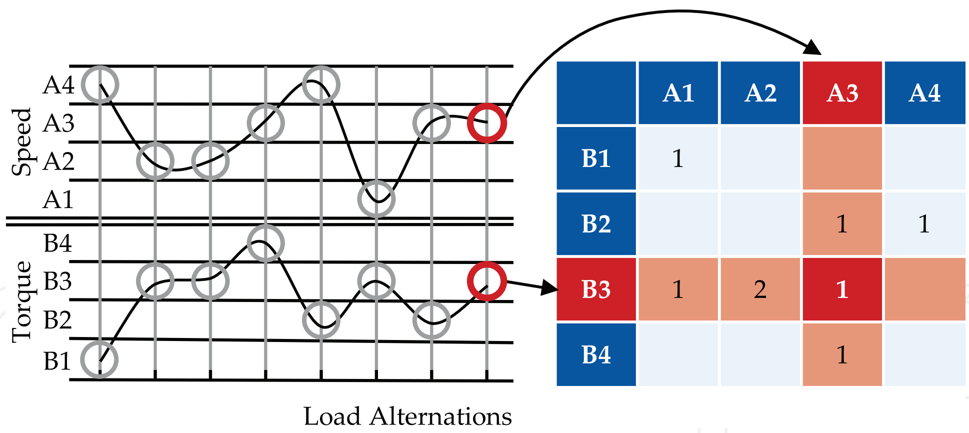


**Figure 6.** Schematic illustration of two parametric time-at-level counting.

The level of complexity that is chosen for the simulation depends on the load cycle that is investigated and on the information that is available about the system. Simulation models that have much detail also require detailed information about the system components. If the system is still in the concept phase, a detailed simulation model may require data that are not yet known. It also has to be considered that simulation models have to be validated to provide accurate and reliable results.

Many systems already have some sensors that provide measurements of some system states. These can be used to improve the quality of simulations by using the available measured data as an input or reference for the validation. This way, the advantages of both methods can be combined.

The measured or calculated loads usually have the form of a load-time function. For the next step, the component models require the data in the form of load per load alternation so that a transformation is necessary. To minimize the calculation effort for the reliability calculation, the load per load alternation can be transformed into a load spectrum using counting methods [33]. For many applications, such as gears and bearings, the combination of speed and torque determines the strain on the component. The time-at-level counting method can be used to define a load spectrum from the continuous data, as illustrated in **Figure 6**. The method defines classes for the values of two parameters, in this case torque and speed. For each load alternation, the combination of torque and speed classes is identified and counted in a matrix. The result of the counting method is the number of load alternations at each load and speed level in the examined time interval.

## 5. Examples of application

To illustrate the discussed challenges for load assumptions for off-highway machinery, two examples have been selected. In the following, the presented method is exemplarily applied to two off-highway drivetrains for different applications. The creation of load cycles and the determination of local component stresses are highlighted.

### 5.1. Power-shift transmission

The following exemplary system is a power-shift transmission designed for mobile machines, such as wheel loaders or dumpers with a power consumption up to 100 kW [8]. The transmission is able to switch between its three forward and three reverse speeds without tractive power interruption with friction clutches. In total, the transmission contains 10 gears, 18 bearings, 7 shafts and 2 radial shaft seals. The initial system analysis reveals gears, shafts and bearings as components allocated in category A and seals and clutches in category B. Since none of the components are designed redundantly, the reliability structure is serial. The representative load cycle in this example is built artificially rather than metrologically. The artificial driving cycle emulates a dumper in a quarry, consisting of five main phases, as illustrated in **Figure 7**. First, after loading material, the dumper drives out of the quarry for which a height profile of the road is being used. In the second section, the loaded dumper is assumed to drive straight ahead. In the third part, the loaded dumper backs up in an unloading zone, unloading and reversing back out. The following fourth segment is similar to the second one, where the empty dumper drives straight ahead, back to the quarry. In the fifth and last phase, the empty dumper drives down to the quarry, following the road profile in reverse. The whole driving cycle lasts 720 s over a distance of approximately 6 km and repeats itself over and over, whereas possible idling times are neglected [8].

The loads on the output of the transmission are calculated by determining the necessary traction force to move the loaded dumper up the quarry and the empty dumper down the quarry. In addition to the weight forces on the slopes, a rolling resistance is assumed, which is 2% of the overall weight force. The empty weight of the dumper is set to 10 t, with a payload capacity of another 10 t. The speed of the vehicle is taken from the maximal acceleration possible from the driving power, subtracting the power to overcome weight forces and rolling forces. The maximal vehicle speed is set to 10 m/s for driving straight ahead and to 7 m/s for uphill/downhill sections [8].

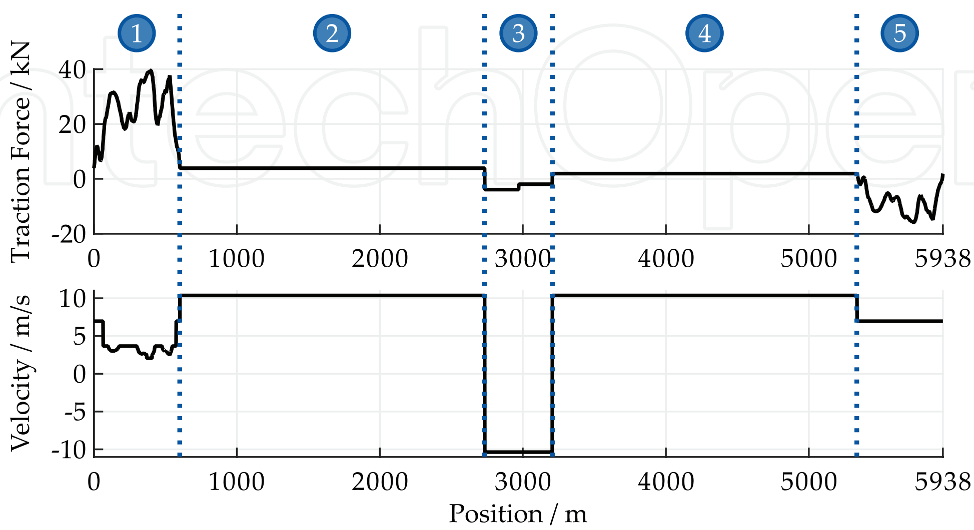


**Figure 7.** Artificial dumper driving cycle [8].

Having torque and speed at the transmission output available, the local component loads and speeds of gears, bearings and shafts can be determined. In this example, a torque plan of the transmission is used, based on the gear ratios of the individual stages, see **Figure 8**.

Loads on the bearings are the result of gear forces and shaft dimensions. For shafts, the critical cross sections and the respective stresses have to be determined according to Ref. [20], which are caused mainly by the gear forces. The friction power in the clutches, which causes the wear considered for its lifetime calculations, is obtained using shifting times, torque and speed differences per shift operation. For the seals, only temperature is considered as an influence on the lifetime. For the environmental influences in this example, a static oil temperature is assumed. The particle contamination factor, which influences the bearing lifetimes according to [17], is assumed to increase over time and drops every 500 operating hours when the oil filter is changed. The load information for the investigated components is categorized using time at level counting which results in load spectra serving as an input for the lifetime calculations. After combination with statistical failure distributions and applying Boole's theory as system theory, it yields a failure distribution plot, illustrated in **Figure 9**. The probability of failure for combined groups of components as well as for the entire transmission is plotted on the ordinate.

The reliability evaluation reveals that the shafts in this example have been designed to be fail-safe and therefore do not appear in the plot. Furthermore, it is highlighted that the gears seem to be designed considerably more durable compared to the other components, according to the load capacity calculation in Ref. [9]. The failure probabilities of the other components are in the same magnitude and define the life expectancy of the transmission. The overall B10 lifetime of the transmission can be derived from the plot with 2961 h which corresponds roughly to a distance of 90,000 km. If the lifetime of the transmission was to be increased, seals, bearings and friction clutches were the components that should be designed more durably [8].

The presented calculation, which yields the transmission lifetime, is repeated for two additional driving cycles, which are basically equal to the initial driving cycle, except the sections in which the dumper drives straight ahead are shortened, so that the total cycle only takes half

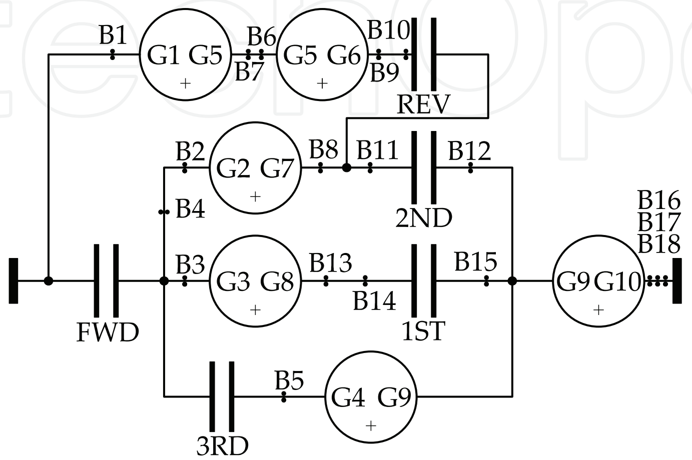


**Figure 8.** Torque plan of investigated power-shift transmission [8].

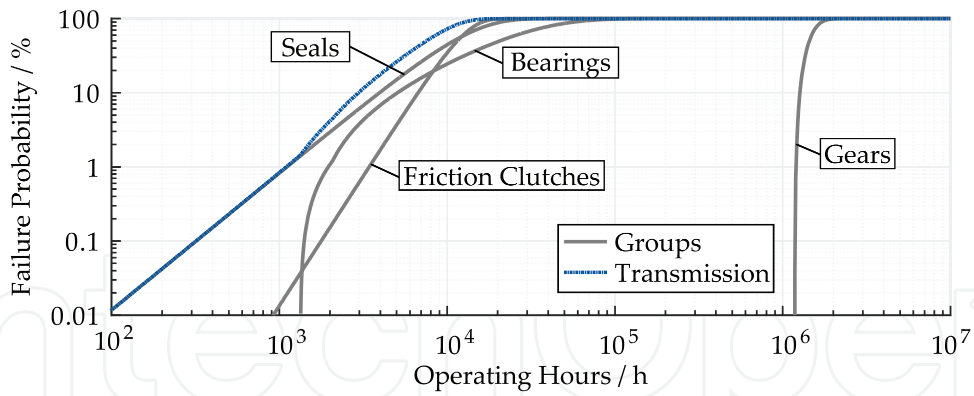


**Figure 9.** Failure probability power-shift transmission [8].

the time (variant 1: 360 s) and extended, so that the driving cycle period is twice as long as before (variant 2: 1440 s). The resulting lifetimes of all three driving cycles for different groups of machine elements as well as the entire system are illustrated in **Figure 10**.

For the short driving cycle, the lifetimes are significantly lower, compared to the initial normal driving cycle. For the long driving cycle, the inverse effect can be observed. Only the lifetime of the seals does not change due to the chosen lifetime model that only considered temperature degradation and is therefore independent of the occurring loads in the transmission. The change in lifetime of the transmission due to the different driving cycles arises from the changed percentages of phases with high loads (uphill/downhill phases) to the section with small loads (straight forward/backward). When executing the short driving cycle, the sections with high loads occur more often, compared to the longer cycles. Therefore, the components in the transmission experience more damage within the same time and subsequently fail sooner [8].

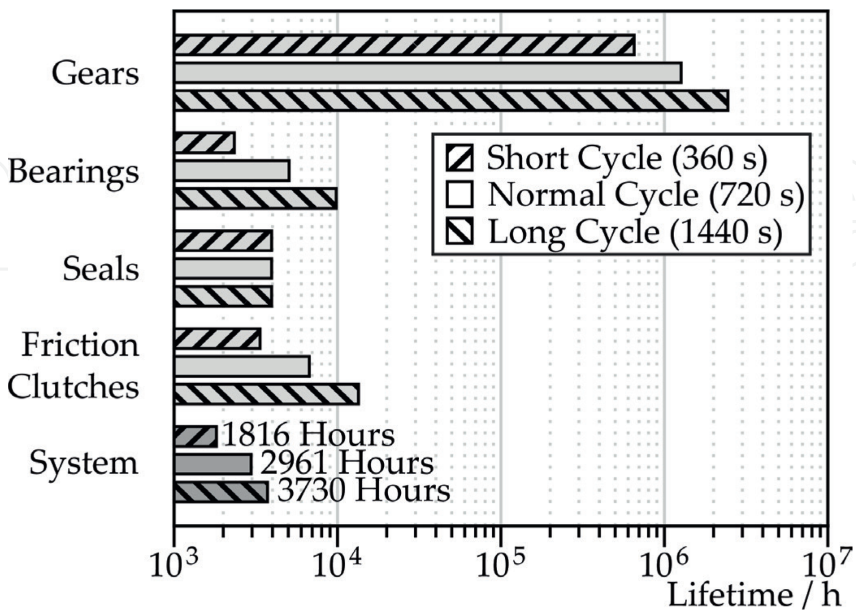


**Figure 10.** Lifetime of the power-shift transmission for different load cycles [8].

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This illustrates that the accuracy of the load assumptions has a large influence on the predicted lifetime of the off-highway drivetrain. The more realistic the investigated cycle is for the loads the system experiences during operation, the better the reliability can be calculated. One way to address this problem can be to use sensor data from the machine during operation to define more accurate load spectra. Using simulations that can calculate component loads from few input signals, the accuracy of the reliability evaluation could be increased without installing extensive metrology. Certainly, this would only be possible for existing systems.

### 5.2. Offshore winch

For offshore cranes, a high reliability is important due to the remote locations in which they are operated. In the offshore setting, spare parts and repair equipment are not easily accessible. The reliability of the drivetrain of an offshore winch can be assessed using the proposed method. In cranes for offshore applications, *Active Heave Compensation* (AHC) systems are used to compensate the vertical vessel movement that is induced by the waves. This allows maintaining the payload at a steady position in spite of the wave movement, which is useful when loads are lifted from a ship to a steady platform or the sea floor, as illustrated in **Figure 11**. The compensating motion is generated by winding and unwinding the crane's winch, which creates dynamic loads on the drivetrain.

The drivetrain consists of the main drum with cogwheels on each side that are driven by 10 pinion wheels each. Each pinion is connected to a hydraulic motor by a gearbox, which offers the necessary torque due to two planetary gear stages.

To create a typical load cycle for the AHC mode, all influences on the winch loads have to be identified. The necessary winch movement depends on the wave height over the course of time, so that the vertical wave movement has to be calculated for the chosen operating point. This can be done using the JONSWAP spectrum [34] which yields the wave movement for a chosen wind speed on the Beaufort scale. As a rough estimate, it is assumed that the vessel, and therefore the payload, follows the wave movement directly. This means that the movement to compensate the waves is the opposite of the wave height at each point in time. From this movement, the required acceleration of the winch can be calculated. To calculate the required dynamic torque due to acceleration and deceleration, the winch's moment of inertia has to be known. It is composed of the drum's inertia, the rope's inertia both on the drum and uncoiled and the inertia of the payload on the rope [35].

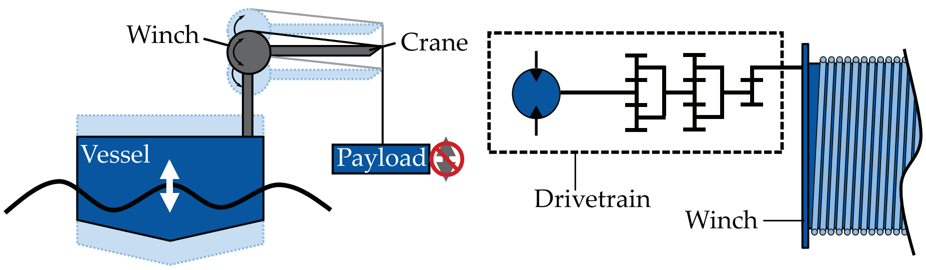


**Figure 11.** Winch movement and offshore winch drivetrain [35].

From the total required torque, the load on all components can be calculated. In this example, only gears and bearings are considered. The gear loads can be calculated using a torque plan, which includes the gear ratios in all gear stages. Since all of the gearboxes are identical and the loads are distributed symmetrically, only one of them is analysed. The torque plan also yields the rotational speeds of all shafts. The bearing loads result from the gear forces and the weight of the shafts. They can be obtained using transfer functions, which can be generated by applying external loads and observing the resulting internal loads that depend on the dimensions of the gears and shafts. The component loads are then categorized and transferred into component stresses using standardized approaches [9, 17].

In the case of the winch, the categorization of the load-time data poses a challenge due to the oscillating motion of the winch. In typical drivetrains, the system performs full revolutions which can then be counted as load alternations. When operating in AHC mode, the drum and the shafts in the drivetrain do not always perform full revolutions so that a modified counting method is used. A new load alternation is counted each time the zero angle position (or multiples of 360°) is crossed, see **Figure 12**. The lifetime models also require the speed and load corresponding to each load alternation. For this application, the mean speed and the mean load between two turning points are considered. This approach offers a conservative estimation of the lifetime.

With the categorized loads, the lifetime of the gears and bearings can be calculated and then used to create a failure probability function. **Figure 13** shows the failure probabilities for the gears and bearings in one gearbox and the combined failure probabilities for the gears and bearings in all gearboxes. The bearings in the system have shorter lifetimes than the gears, which means that they determine the system's reliability. The system's failure probability in this example is equal to the failure probability of the combined bearings. The information about the system can then be used to identify critical components and derive the overall lifetime.

This example shows that due to special circumstances of operation, off-highway drivetrains can pose challenges for the reliability calculation. In the case of the winch, the non-typical oscillation movement creates difficulties for the load classification and the lifetime models of the components. Although the system can be handled by modifying existing methods, it shows that additional research is still required.

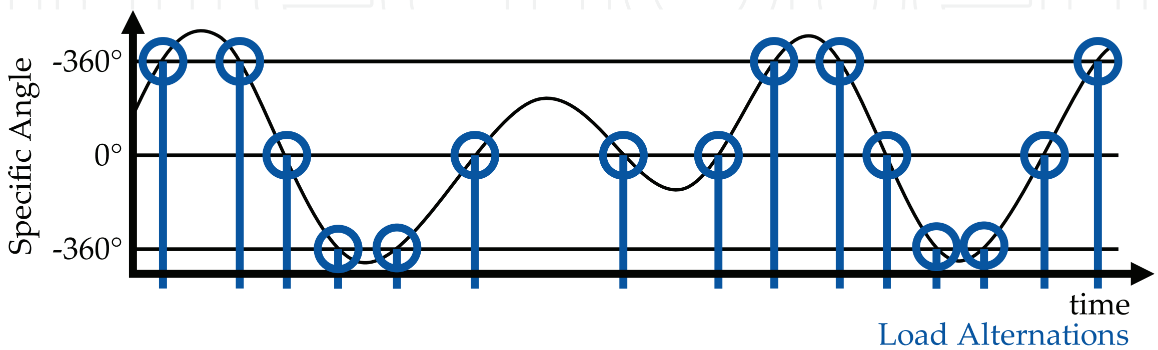


**Figure 12.** Schematic illustration of load alternation counting.

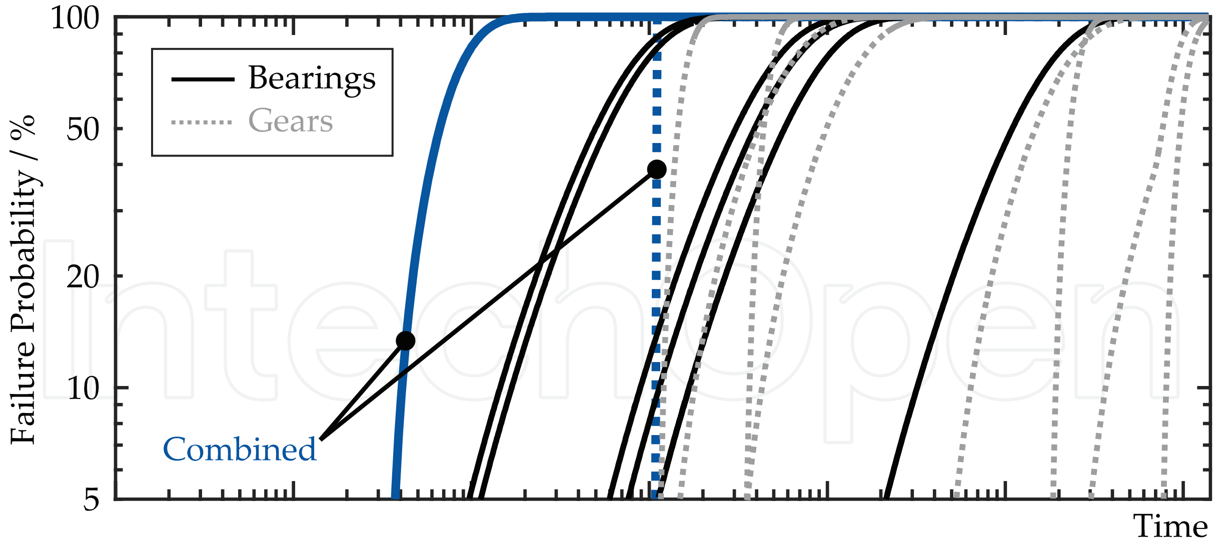


**Figure 13.** Failure distribution of gears and bearings [35].

## 6. Conclusion

Off-highway machines often perform tasks that are embedded into a process chain so that machine downtime causes high costs and delays in the process. Therefore, approaches to assess the reliability of each machine quantitatively offer a possibility for a lower total cost of ownership due to better maintenance planning.

The challenge in the quantitative reliability evaluation of off-highway drivetrains lies in the determination of a representative load cycle, since the accuracy of the lifetime calculation depends on precise load assumptions. Due to the wide range of off-highway machines, their applications with different tasks and varying loads, even for machines with identical configurations, the loads are machine specific. The most precise information could be generated through elaborate measurement campaigns of the machine in the field. Since such approaches are sometimes not possible, especially in early development stages, simulations offer a possible alternative.

The described method for the reliability evaluation is applied exemplarily to two off-highway drivetrains with different fields of applications. The example of a power-shift transmission illustrates that the calculated lifetime depends greatly on the assumed load cycle. Therefore, methods have to be found to address this issue, possibly through an approach that combines measurements with simulations to calculate component loads from few measured signals.

The example of the offshore crane winch shows that the method can be applied to a wide range of systems. The challenge for the reliability calculation for the winch drivetrain lies in handling the oscillating motion during AHC operation. Additional research for better counting methods and more extensive lifetime models is required.

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