

FAILURES IN GEARBOXES DUE TO MATERIAL PROBLEMS

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Abstract

Mechanical failures in general can appear because of weak design, wrong handling, and problems during the manufacturing process, material problems etc.

This paper is dealing with failures in lifting machinery gearboxes which appear because of material stresses and tribological problems. The study conducted is based on theoretical analysis and analysis of inspection results for failures that have appeared in cranes and other lifting machinery in southern part of Sweden.

Investigation shows that besides the surface damages like abrasive and adhesive failures also cracks in the material in different parts appear.

Since these types of gearboxes are made in short series, gearbox housings and sometimes also gears are made of welded steel. Hence welding process makes the weld joints brittle; cracks do start and developed until the part fails. Another problem that happens due to welding is warping which leads to changes in the geometrical tolerances and that does affect the parallelism of the shafts and the functionality of the gearbox.

Key words: mechanical failures, material stresses, tribology, lifting machinery, failures in gearboxes

1. INTRODUCTION

Lifting equipment is any work equipment for lifting and lowering loads. Lifting operations are inherent to many occupations in the industry. They can be performed manually or using lifting machinery.

1.1. Overview of the different types of lifting machinery

Different types of lifting machineries are used in construction, transportation, industries, ports and other fields. Those are machines which use one or more simple mechanisms to create mechanical advantage and thus move heavy loads.

Two types of material lifting equipment are defined - Hoisting Equipment and Conveying Equipment. Hoisting Equipment is a group of machines with lifting transmission intended for moving loads mainly in batches. There are three types of hoisting equipment: Hoisting machinery, Cranes, and Elevators. [allengineering-info]

- A hoist: is a device used for lifting or lowering a load by means of a drum or lift-wheel around which rope or chain wraps. It may be manually operated, electrically or pneumatically driven and may use chain, fibre or wire rope as its lifting medium.
- A crane is a machine that is capable of raising and lowering heavy object and, while holding them suspended, transporting them through limited lateral distance. Cranes are a type of machinery, generally equipped with a hoist, wire ropes or chains, and sheaves, that can be used to lift and lower heavy materials and to move them horizontally. Cranes are distinguished from hoists, which can lift objects but that cannot move them sideways. Cranes are also distinguished from conveyors that lift and move bulk materials, such as grain and coal, in a continuous process.

Cranes are classified according utilization and their form. Many factors are taken into consideration while selecting a crane. These factors include lifting capacity, crane use and application and the number of work cycles that the crane is required to undergo.

1.2. Gearboxes in lifting machinery

Different types of gearboxes are used in lifting machinery: gearboxes for conveyor, vertical gearboxes, forward-reverse gearboxes, electric motor gearboxes, high rpm gear speed reducers and etc.

The main product objectives for gearboxes used in cranes are: wide range of reduction ratio, high mechanical transmission efficiency, smooth operation, low noise, long service life, high carrying capacity, easy to overhaul, and easy to install.

The most common for different types of lifting machineries is the reduction gearbox for hoisting. The range of the used gearboxes varies very much. From larger hoisting speed reducers and large crane gearboxes to medium hoist lifting gearbox reducers and small hoisting reduction gearboxes.

The principal structure of a hoist contains: motor, brake pad, coupling, gear reducer, bearing house, steel cable, hook, and limitation stopper (see Fig.1.1.).

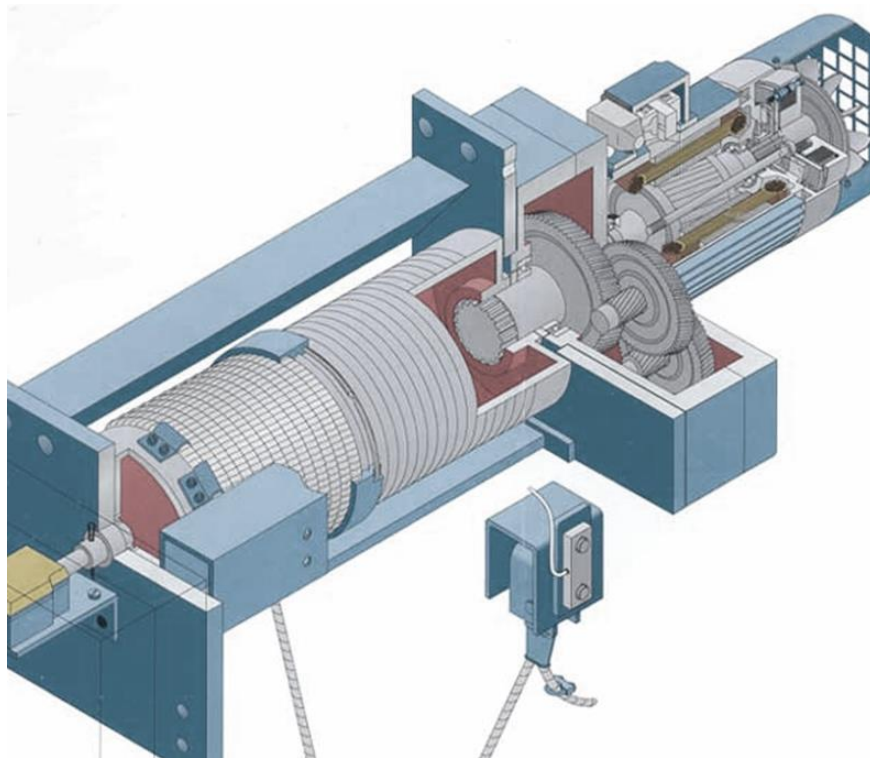


Figure 1.1. *Principal structure of a hoist*

1.3. Failures in lifting machinery

Because of working in heavy conditions, and in changing environment failures are natural to appear in lifting machineries. Failures in the sub-systems of the lifting equipment add high risks of fatal collapse, and unexpected consequences, such as economical lose and even people injuries. Any undesirable change in the size, shape or material properties of a structure, machine or machine part that affects the satisfactory performing machine's intended function is called mechanical failure. Mechanical failures in lifting machineries could appear due to causes from different type and nature (see Fig.1.2.).

To analyse a mechanical failure there are four main categories to be defined:

- The equipment (for example gearbox in a hoist)
- The failure defined in time and logical sequence (for example catastrophic breakage of teeth in a gear)
- The failure mode (by definition is what the equipment or component failed from, e.g. Fatigue)

- The root cause (by definition, what caused the failure mode to occur and what can be changed to prevent re-occurrence, it could be more than one, e.g. coupling misalignment, gear misalignment, input torque changes, pinion and gear eccentricity, machining errors, torsional vibration and resonances). [Mahto, 2008]

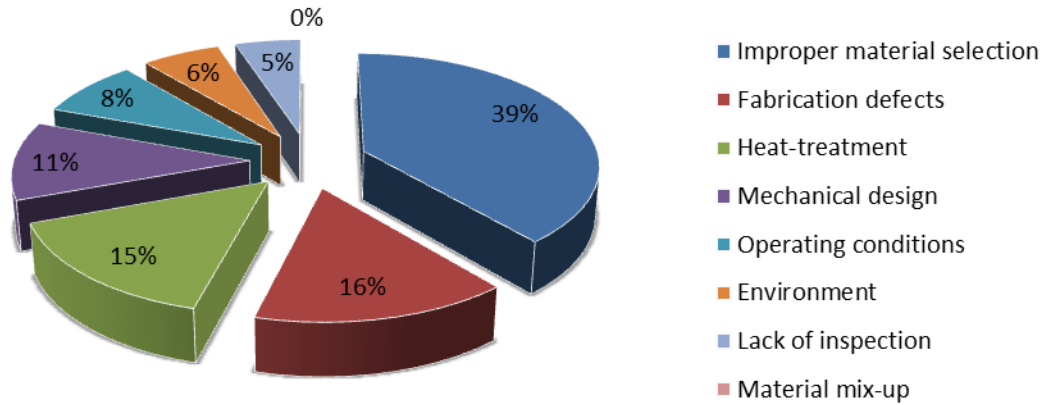


Figure 1.2. Frequency of Different Reasons for Mechanical Failures, [Neitzel, 2001]

Failure modes, which are the physical processes causing failure, could be defined based on three categories: manifestation of failure, failure-inducing agents and location of failure. [Collins, 1993]

- The manifestation of failure could be: elastic deformation, plastic deformation, ruptures or fractures and material change (metallurgical, chemical or nuclear).
- The failure-inducing agents are also divided in four types – Force, Temperature, Time, Reactive environment.
- The two failure locations could be: body type and surface type.

Each specific failure mode is defined as a combination of one or more manifestation of failure with one or more failure-inducing agents and a failure location.

Some typical failure mechanisms in lifting equipment are due to fatigue, wear, corrosion, and ductile or brittle fracture. Insuring the lifting equipment is not overloaded, in addition to appropriate inspections, maintenance, and repair can help eliminate many failures.

The leading cause of lifting equipment failures are due to improper operation. Unfortunately, overloading of a lifting mechanism is not an uncommon practice. Wear is the most readily recognized condition by operators and maintenance personnel. Excessive wear is usually a relatively easy condition to correct.

However, complete elimination of wear in components used in lifting equipment is not feasible.

Fatigue is a more insidious type of failure mechanism as it is more difficult to detect. Periodic inspection by properly trained inspection and maintenance personnel can prevent fatigue failures of critical components. [Dubey 2015]

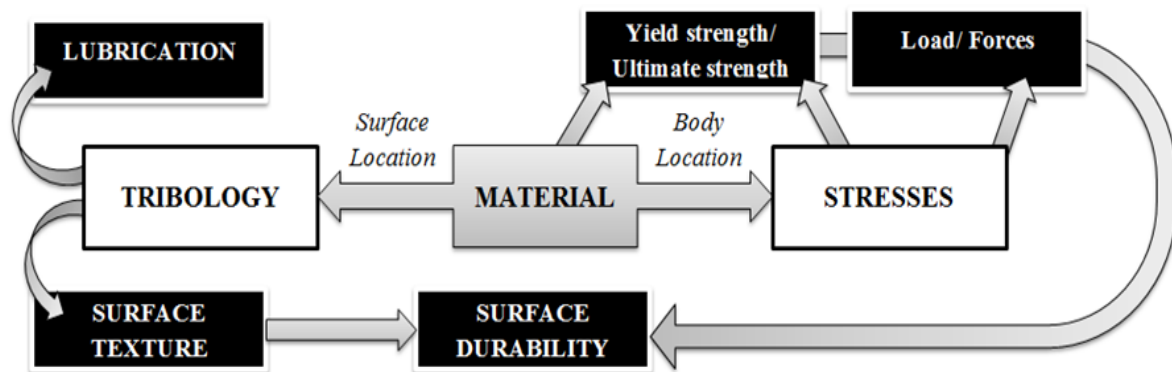


Figure 1.3. Mapped context of the presented problem in this work

1.4. Mapping the content of the presented work

The focus in this paper is on failure due to the material of gear transmission parts. But the material cannot be treated separately from other factors, which are part of the problem causes, like contact stresses in gears and rolling bearings. In addition to the material problems, also tribological issues, like surface texture in the contact areas, have to be treated. There are loads in all the machine elements of the gearbox and these have to be dimensioned for the type and size of the stresses. Therefore load and force analysis must be presented, when dealing with material issues. The content of the presented work is mapped in figure 1.3.

2. MACHIN ELEMENTS AND MATERIALS IN GEAR TRANSMISSIONS

In this paper only parallel gear drives or gear transmissions (transmissions with parallel shafts), containing spur or helical gears, will be treated.

According to [Jelaska, 2012], “A Mechanical power transmission or a mechanical drive consists of units which transfer power from the prime mover (driving machinery) to the actuator (driven machinery) with the assistance of rotary motion”. Transmissions or mechanical drives are situated between the prime mover and the actuator (see Fig. 2.1.)

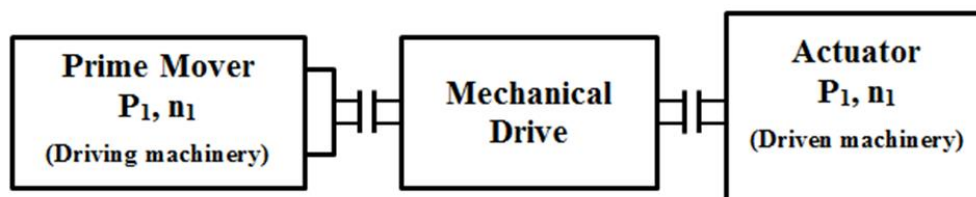


Figure 2.1. Sketch of a mechanical drive application [Jelaska, 2012]

The mechanical power transmissions can be with constant or variable transmission ratio. In the case of lifting machinery, the drives are with constant transmission ratio, which means the constant rotational speed of the driving shaft results in constant rotational speed of the driven shaft.

2.1. Gears and gear materials

Gears are the “heart” of the gear transmission and the parameter relation between them will decide the transmission ratio. Spur gears are the simplest gears and they make the base of other type of gears. The only difference between the spur gears and helical gears is that the direction of the gear teeth in the helical gears is not parallel with the axis of the gear.

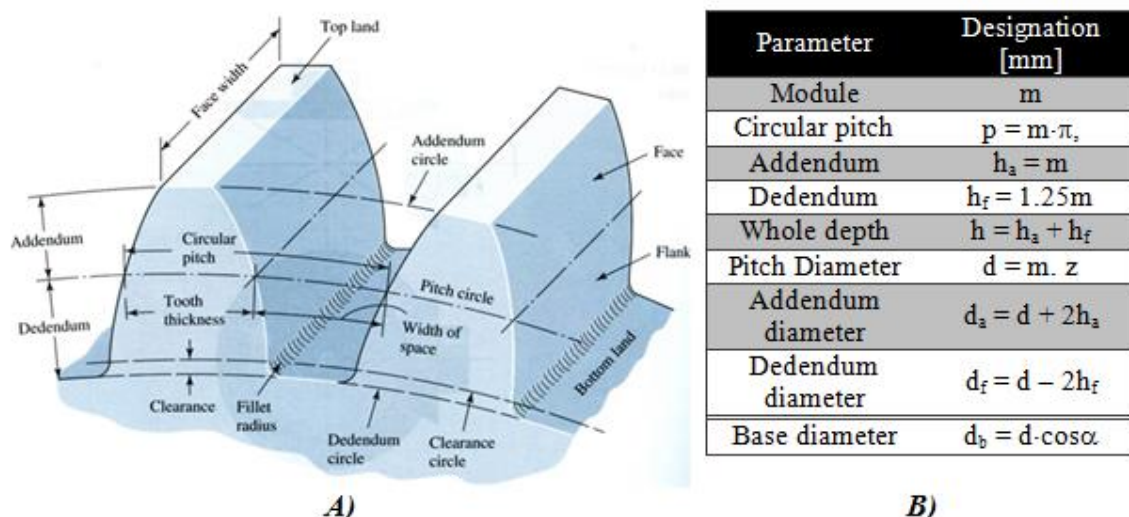


Figure 2.2. A) Nomenclature of spur-gear geometry [Shigley, 2001]; B) Relation between gear parameters and module

The geometry of the gears is based on the size of the module which is standardized. The parameters for spur gears are shown in Fig.2.2.A) and B) almost all the other geometries are connected to the module.

Gear materials are selected depending on the required properties, lowest cost, and to satisfy other functional and technical requirements. Fatigue strength, surface hardness, and wear resistance are few of the most required material properties.

Gear materials in lifting machineries are made of steel, cast iron, or cast steel. Since the gear teeth will be subjected to contact stresses, the gear material must be heat-treatable, for example possible to be hardened. The best physical property with respect to the gear strength is that the gear has a hard surface and a ductile core. In many cases the gear transmission series for lifting machineries are too short; therefore, gears (and housings) are made of welded plates and parts. In this case, the material must be weld-able and heat-treatable. Table 2.1. is showing surface hardness and surface durability ($\sigma_{H \lim}$) for some material used in gears.

Table.2.1. Allowable surface hardness and surface durability for some gear materials

Material	Notation (example)	Hardness	$\sigma_{H \lim}$ [MPa]
Lamellar cast iron	GJL-200	180 HBW	300
Black malleable cast iron	GJMB-350	150 HBW	320
Nodular cast iron	GJS-400	180 HBW	370
Unalloyed cast steels	GS 52.1	160 HBW	320
Construction steels	E335	190 HBW	430
Through hardening steels	34CrNiMo16QT	350 HBW	750...830
Carburizing steels	17 CrNiMo8	58...62 HRC	1300...1500

2.2. Shaft and shaft materials

A shaft is a rotating or stationary machine part, usually with circular cross-sections. Shafts in gearboxes are usually rotating and they are prepared for having mounted on gears, pulleys, sprockets, bearings, etc. It is normally subjected to various combinations of axial, bending and torsional loads and these loads may be static or fluctuating. Typically, a rotating shaft transmitting power is subjected to a constant torque [Juinall, 2012]. These loads result in static and variable stresses in the shaft material. In addition, for to satisfy strength requirements, shafts must be designed so that deflections are within acceptable limits.

Many shaft-design situations include the problem of transmitting torque from one element to another on a shaft, for example from shaft to hub and in opposite. Common torque transfer elements are keys, splines, setscrews, pins, press or shrink fits, and tapered fits. Press- and shrink-fits for securing hubs to shafts are used both for torque transfer and for preserving axial locations. The resulting stress-concentration factor is quite small [Shigley, 2001]. But there are other torque transferring means, like keys, which cannot solve the axial location problems, therefore additional locational devices must be used, like sleeve, ring and groove, setscrew, shaft shoulder, etc.

To minimize deflections, shaft materials are generally cold drawn or machined from hot-rolled, plain carbon steel. Cold drawing improves the physical properties, it improves the values of ultimate tensile and yield strengths of steel [Ugural, 2015]. Alloy steels are used when toughness, shock resistance or greater strength are needed. These materials can be heat treated to give the desired properties. Where wear resistance is needed rather than extreme strength, it is usual to harden only the surface of the shaft requiring it, the reminding of the shaft is left in its original conditions.

All changes in the shaft cross sections are sources for stress rising, therefore stress risers in already high stressed parts of a shaft should be avoided. All the stress concentration factors must be taken into concentration in the dimensioning and design process. Keys are used to enable the remission of torque from the shaft to the hub or in opposite. There are many types of keys for different requirements, the square and flat keys are the most used in machine assemblies. To use keys, it requires corresponding keyways in the shafts as well as in the hubs. Keys are, in the most of the cases, made of low-carbon, cold-drawn steel. For example, from the American Standard, steel AISI 1020 CD having an ultimate strength of 420 MPa, yield strength of 352 MPa, and a 15% elongation [Mott, 2004]. The strength and ductility of this material is adequate for the most applications.

2.3. Bearings

Bearings used in gearboxes in lifting machineries are rolling-element bearings also known as rolling bearings. Rolling bearings are standardized machine elements with dimensions and all other important data. All the data about the bearings are published in the rolling bearings' manufacturers product catalogues and/or on their websites to be available for machine designers. Rolling bearings can carry radial, axial, or a combination of radial and axial loads.

These bearings provide coefficient of friction between 0,001 and 0,002. The designer must deal with such matters as fatigue, friction, heat, lubrication, kinematic problems, material properties, machining tolerances, assembly, use, and cost [Ugural, 2015]. There are of course many different types of rolling bearings designed for different load type, size and other conditions. The rolling bearings are divided in two major groups depending on rolling element, i.e. roller bearings and ball bearings. A rolling bearing is made of an outer ring (race), rolling elements, inner ring (race), and separator (see Fig.2.3).

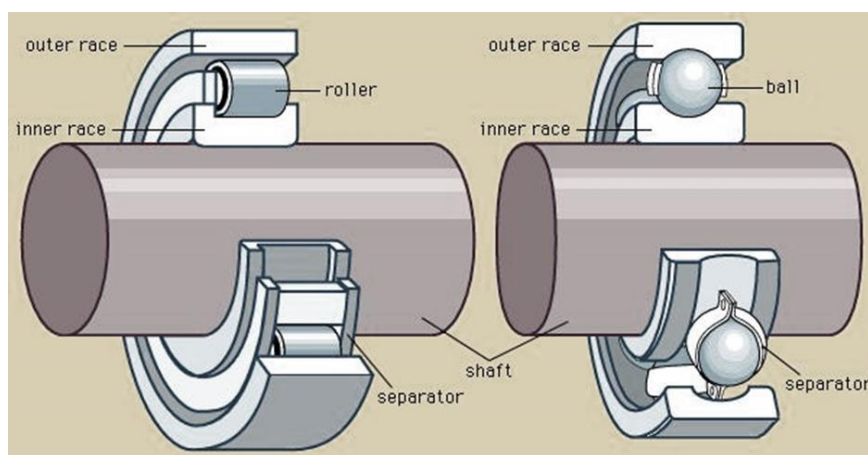


Figure 2.3. Structure of Roller and Ball Bearings,

Most balls and rings (races) are made of high-carbon chromium steel and heat-treated to high strength and hardness, and the surfaces are ground and polished. Separators are usually made of low-carbon

steel and copper alloy, such as bronze. Roller bearings (but not ball bearings) are often made of case-hardened steel alloy. Nowadays there are bearing steels with reduced level of impurities.

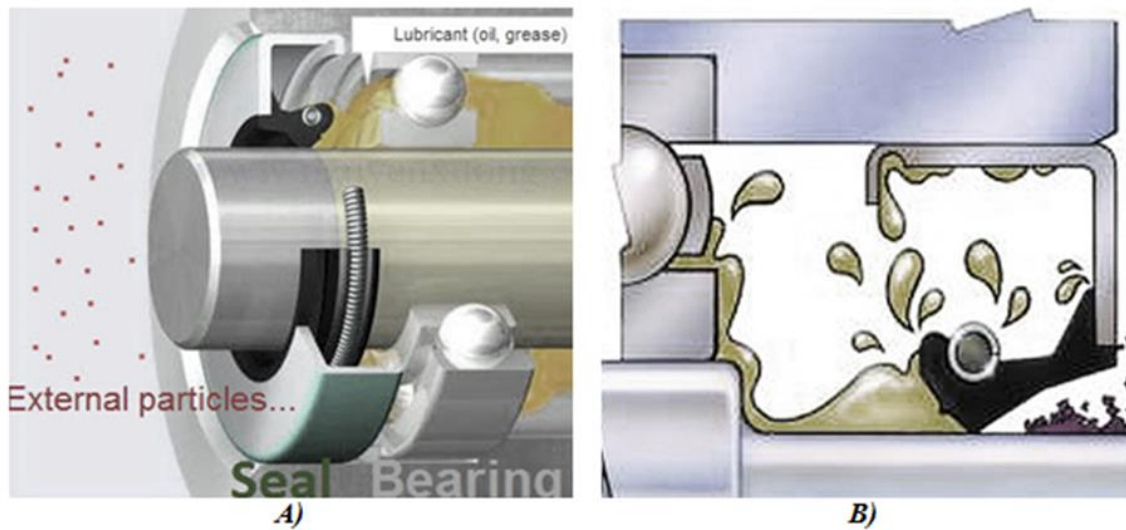


Figure 2.4. A) Seal on shaft in gearbox; B) Seal in use

2.4. Seals

Seals are very important parts in machines where lubrication is used, the seals will keep the lubricants inside a space at same time it will keep the contaminants, water, and dust outside the machine, like the case of a gearbox. There are different types of seals used in different machines, some parameters which will affect the seal selections are: type of fluid to be contained, pressure, relative motion between the seals and other surfaces, temperature, tolerance for leakage, service, etc. For sealing around rotating shafts such as ingoing and outgoing shafts in gearboxes, engines, etc. radial shaft seals must be selected (see Fig.2.4).

According to [Mott, 2004], when radial lip seals are required around shafts, the shafts are typically steel. They should be hardened to HRC 30 to resist scoring of the surface. Tolerance on the diameter of the shaft on which the seal bears should confirm the recommendations presented in Fig.2.5.B).

To insure that the seal lip can follow the variations. The surface of the shaft should be free of burrs to protect against tearing the seal. A surface finish of 254 to 508 μm with adequate lubrication is recommended, to ensure full contact and to reduce friction between the seals and the shaft surface.

Most of seal materials are resilient (elastic) to permit the sealing point to follow minor variations in the geometry of mating surface. Radial shaft seals require rigid, hard materials that can withstand constant sliding motion. Resilient seals are made from synthetic elastomers such as Neoprene, Fluorocarbon, Butadiene, Polysulfide, Polyacrylate etc. Design requirements and material properties will de will decide which of the material will suite in any case. Table 2.2. is showing some material properties that meet requirements, like weather resistance, acid resistance etc.

Table.2.2. Seal material properties with respect to requirements

Requirements	Material (with properties which meet the requirements)
Weather resistance	Silicone, Fluor silicone, fluorocarbon, ethylene, etc.
Petroleum fluid resistance	Polyacrylate, polyester, PNF, nitrile, polysulfide, etc.
Acid resistance	Fluorocarbon
High temperature operation	Ethylene propylene, fluorocarbon, polyacrylate, silicone, etc.
Cold-temperature operation	Silicone, fluorsilicone, ethylene propylene, PNF
Tensile strength	Butadiene, polyester, and polyurethane
Abrasive resistance	Butadiene, polyester, and polyurethane
Impermeability	Butyl, polyacrylate, polysulfide, and polyurethane

2.5. Lubrication system

It is not possible to run a gearbox without lubricant. All contact areas which are moving with respect to each other must include a type of lubricant to prevent solid surface contact (like metal to metal contact). In the case of gearboxes, lubricant is needed for the bearings, gears, and seals. Besides the separation of solid surfaces, lubricant has other advantages like heat distributing and dissipating, corrosion prevention, and transportation of broken surface peaks in the machine elements. Depending on the load, speed, and temperature requirements, bearing lubricants are either greases or oils. For high speed and/or higher load demands, oil is recommended. Closed gearboxes are usually lubricated by oils.

As mentioned, no gearbox, unless it is designed for very light loads and low speeds, will manage to run without lubrication. These gearboxes must have a well-planned lubrication system to insure lubrications of all the roller bearings, gears, seals and other solid surfaces in contact, and with relative movement. The duties of lubricants are to minimize the contact friction, cooling down, and cleaning the contact surfaces. It is very important that the lubricant is fresh and clean in order to accomplish its duties to carry loads, to separate the hard surfaces for minimizing or preventing metal to metal contacts. The simplest method of oil lubrication is the oil bath. The oil, which is picked up by the rotating components of the gearbox, is distributed within the gearbox and then flows back to the sump in the housing.

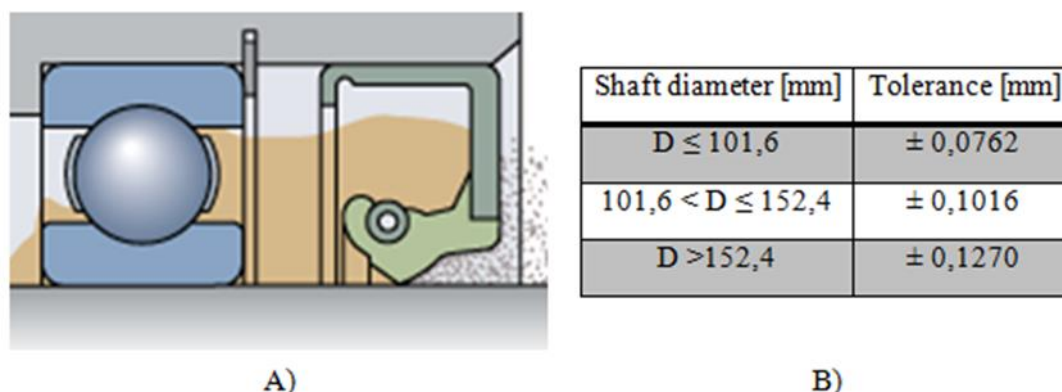


Figure 2.5. A) Lubrication system; B) Tolerances in relation to the shaft diameter

The oils (lubricants) are classified in systems, like ISO-Grading system based on the lubricant viscosity. Viscosity is the most significant lubricant property, but the viscosity is depending on the temperature of the lubricant. Therefore, the viscosity for oils is represented in diagrams where one axis is showing the viscosity and the perpendicular axis is showing the temperature. The relation viscosity-temperature is used to be expressed in the Viscosity Index (VI).

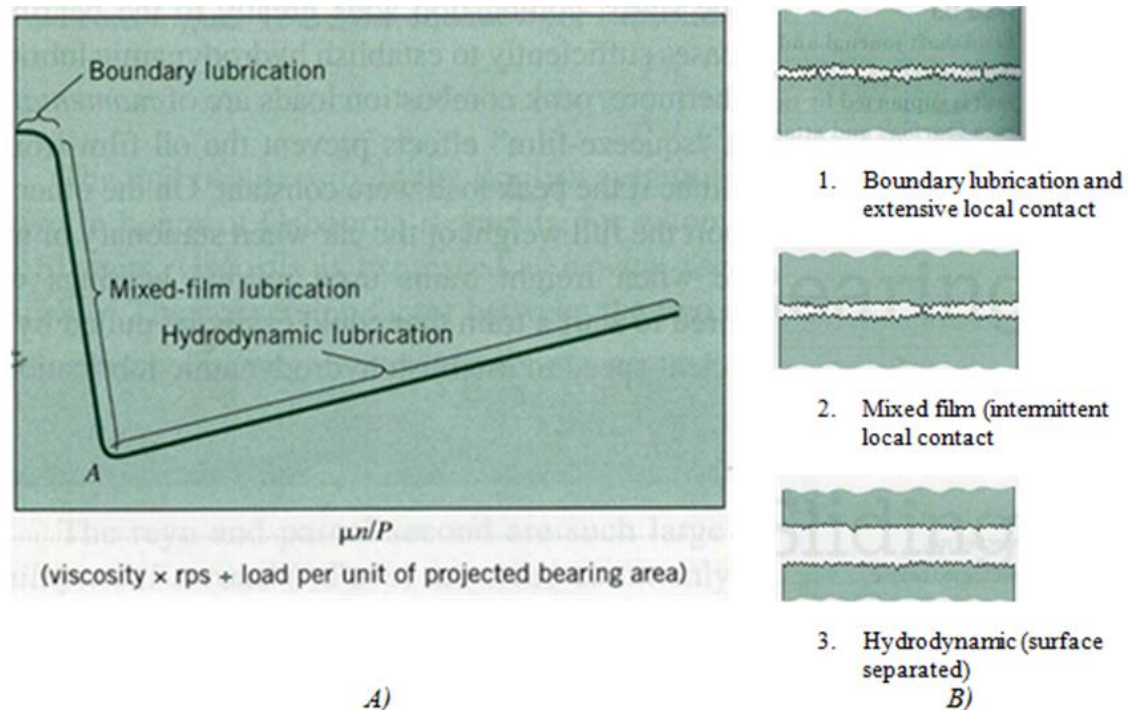


Figure 2.6. A) Friction in relation to speed; B) Contact zone for different type of lubrication

Lubrication is commonly classified according to degree with which the lubricant separate the sliding surfaces as shown in Fig.2.6. [Juvinall, 2012]

1. Boundary lubrication: surface contact is continuous and extensive, but the lubricant is continuously “smeared” over the surface. This reduces the friction and wear.
2. Mixed-film lubrication: The surface peaks are intermittently in contact, and there is partial hydrodynamic support
3. Hydrodynamic lubrication: The surfaces are completely separated by the lubricant film.

Lubricant additives are mainly added to the base oils to enhance the viscosity index, the lubricity under boundary lubrication conditions, and the lubricant life, which in turn enhances the lifetime performance of the machine [Van Beek, 2009]. Lubricity concerns the friction and wear in the boundary lubrication regime.

To improve lubricity, anti-wear (AW), and extreme pressure (EP) additives are added to the lubricants. Other additive like anti-oxidant, rust and oxidation, detergents, anti-foam etc. are also added to extend the lubricants life resulting in longer oil change intervals.

For hydrodynamic lubrication, when the solid surfaces are totally separated, there will be a minimum oil film thickness h_{min} (usually from 0,1 to 0,5 μm) which is very important for the function, friction and heat in the machine element. For example, in a journal bearing, the minimum oil film thickness will be in the narrow gape, see Fig. 2.7.A). Although the solid surfaces look to be polished, like the case in the journal bearings, a closer look will show that the surface have roughness and waviness. The surface roughness is usually given in different standards, but the most common used is the arithmetical mean deviation R_a . Other representations of surface roughness are the standards deviation or variance σ . According to [Van Beek, 2009], if the height of the roughness is disturbed normally (Gaussian) then:

$$\sigma_{Gaus} \approx 1,25 R_a \quad (2.1)$$

As shown in Fig.2.7.B) the variance will be calculated for each of the two surfaces when R_a values are known ($\sigma_1 \approx 1,25 R_{a1}$ and $\sigma_2 \approx 1,25 R_{a2}$), then the common variance will be calculated as:

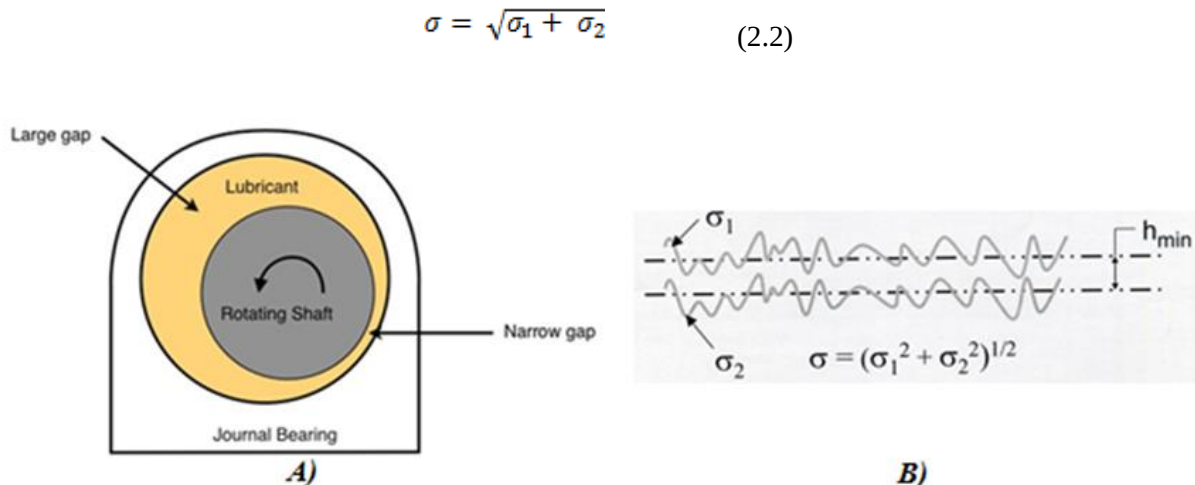


Figure 2.7. A) Lubrication of a journal bearing; B) Specific oil-film thickness,

The specific oil-film thickness λ can be calculated

$$\lambda = \frac{h_{min}}{\sigma} \quad (2.3)$$

The specific film thickness defines the lubrication classification according the following table.

Table.2.3. Lubrication classification

Specific film thickness:	$\lambda < 1$	$\lambda = 1 - 2$	$\lambda > 2$
Lubrication:	Boundary	Mixed Fil	Hydrodynamic

3. FORCE ANALYSES AND STRESSES

The materials in different structures and machine parts behave different depending on the type of loads and forces. Material behaviour in such situations cannot be treated isolated from the stresses appearing in the material due to the different types and sizes of loads. This part is showing the process of analysing the different types of forces and loads in vital parts of a simple gearbox, and then connects the forces to the different types of stresses which appear in the material.

A gearbox is usually connected to driving machinery through an ingoing shaft, and to driven machinery through an outgoing shaft, as is shown in Fig 2.1. The driving machinery can be an electric motor, an engine, or another mechanical transmission. In the case of electric motor or engine, the Power P and the rotational speed n are known, and then the torque T can be calculated. As seen in figure 3.4. an one-step helical gearbox driven by a belt transmission and driving a chain transmission will be analysed. This combination drive will be used to illustrate the force analyses and stresses in the different parts of the gearbox. In this case, the torque T_1 and the rotational speed n_1 of the ingoing shaft (shaft 1) must be known.

3.1 Gear forces and stresses

When the ingoing shaft is rotating, the pinion which is connected to that shaft will also rotate and it will force the gear and the outgoing shaft to rotate with opposite direction. The gear is rotating because of the pressure on its flank from the contact with the pinion flank. The torque of the ingoing shaft T_1 in combination with the pitch diameter of the pinion d_1 will produce a contact force between the gear flanks; this force is called normal force F . Because of the geometry of helical gears with helix angle β and pressure angle α , the normal force is nor perpendicular with the gear pitch diameter. To make the force analysing easier, the gear normal force is divided in three force components; i.e. radial force F_r , axial force F_a , and tangential force F_t , as shown in figure 3.1.B). The latest force component can easily be calculated by the equation below.

$$F_t = \frac{2 \times T_1}{d_1} \quad (3.1)$$

Figure 3.1.B) is showing the relation between the normal force and its components together with the pressure and helix angles. The loads together with the geometry of the gears will produce different types of stresses. As shown in Fig 3.2. a contact stress is appearing when the two mating gears are in contact while transmitting a torque. This situation is very similar to the one when two cylinders are in contact; therefore, a modified Hertzian contact stress equation is used to calculate the contact stress. The movement of the gear teeth flanks at the contact area is partly rolling and partly sliding.

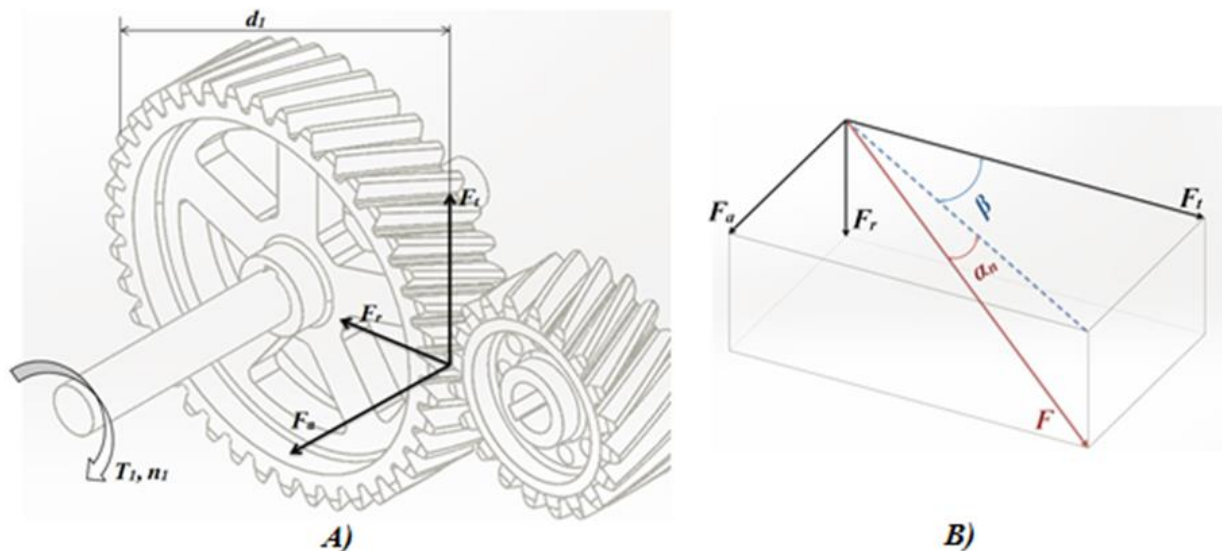


Figure 3.1. A) Helical gear pair with forces in the contact area; B) Force components in the contact point of helical gears

The teeth roots of the gears are subjected to bending stresses, where one side is subjected to tension and the other side to compression stresses. This situation can be compared with a beam with a free end loaded by a perpendicular load to the beam length. The bending stress in the gears is calculated in the same way as for a beam when modification factors are taken into consideration. The gear teeth are loaded during a short period of each cycle (rotation), therefore the loads and the stresses are cyclic. In this case both the contact stress and the bending stress are repeated from zero to the maximum values during each rotation.

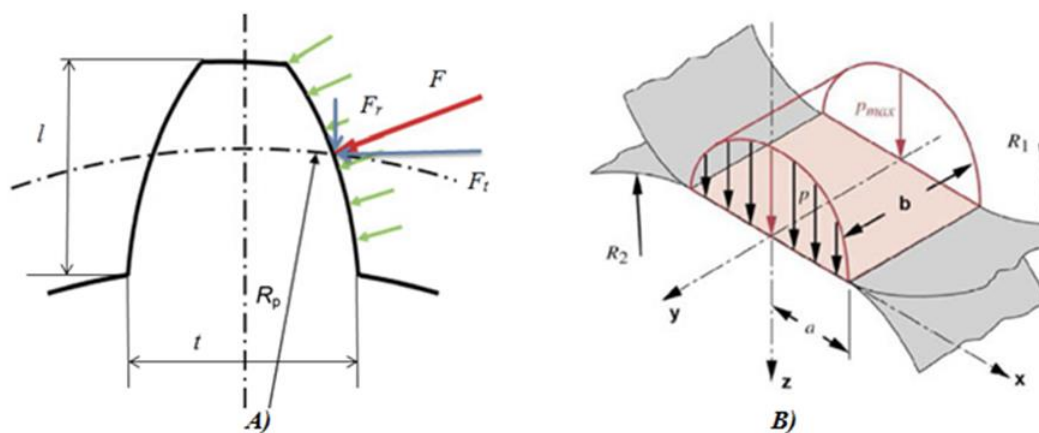


Figure 3.2. A) Gear tooth force and pressure; B) Contact stress between two mating gears [Norton, 2000]

3.2 Shaft forces and stresses

Gearbox shafts are usually designed in that way that it is prepared to be connected to different hubs like gears, sprockets, pulleys etc. These preparations include keyways, grooves, holes, shoulders etc., as shown in figure 3.3.

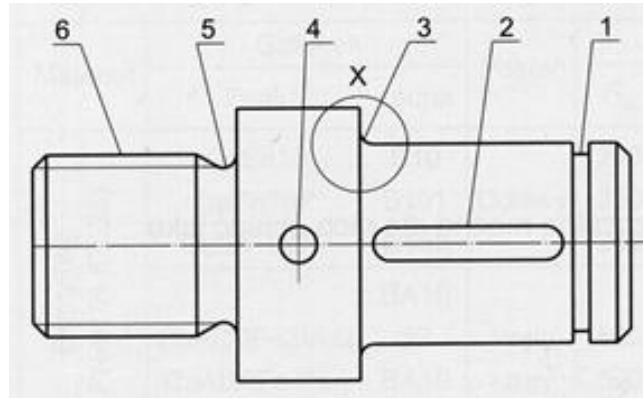


Figure 3.3. Stress raisers in a shaft: 1. Snap-ring groove, 2. Keyway, 3. Fillet radius, 4. Hole, 5. Thread-end groove, 6. Threads. [Czesc 2, 2011]

The power and the torque are transmitted from one hub to another through a shaft at same time as the shaft must be supported by bearings. Therefore, the shafts in gearboxes are usually loaded by a combination of loads producing a combination of stresses. Figure 3.4. is showing an ingoing shaft transmitting a torque from the pulley outside the gearbox to the pinion inside the gearbox. It is visible that the shaft is subjected to bending moment in two directions (in the vertical and horizontal plane), torque between the pulley and the pinion, and axial “compression load” between bearing A and the pinion.

When the shear force and bending diagrams for the both planes are made, the maximum bending moment and its position along x-axis will be calculated. The bending moment is static (constant), but because the shaft is rotating, it will produce “rotating bending” fatigue (or cyclic stress) in the shaft. The bending stresses are normal stresses (σ). The torque is normally constant which leads to static shear stresses in the “torque” loaded parts of the shaft. The “static” axial loads will produce axial “normal” stresses in the loaded parts of the shaft.

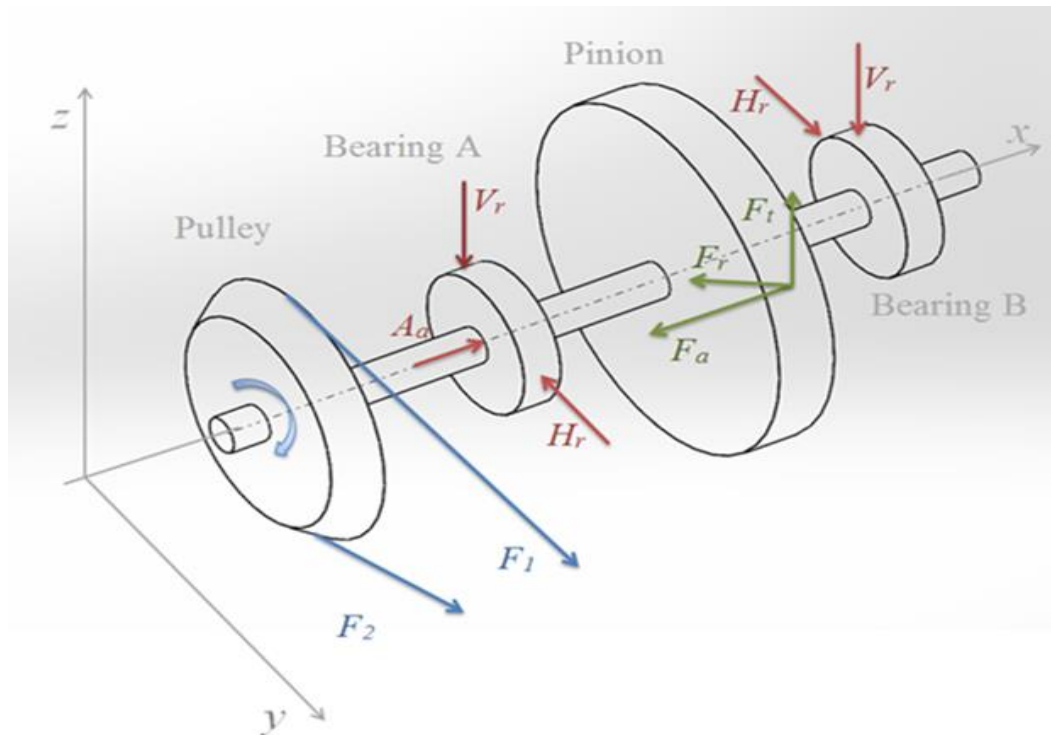


Figure 3.4. Ingoing Shaft force analysis

These three different types of stresses will be used to find the equivalent Alternating bending stress σ_{ea} and the equivalent mean bending stress σ_{em} :

$$\sigma_{ea} = \sqrt{\sigma_a^2 + 3\tau_a^2} \quad (3.2)$$

$$\sigma_{em} = \frac{\sigma_m}{2} + \sqrt{\tau_m^2 + \left(\frac{\sigma_m}{2}\right)^2} \quad (3.3)$$

In this case,

$\sigma_{ea} = \sigma_a = \text{Bending stress}$ (Alternating bending stress)

$\tau_a = 0$ (No alternating shear stress)

$\sigma_m = \text{Axial stress (mean axial stress)}$

$\tau_m = \text{Torsional stress (mean shear stress)}$

Stress concentration factors must also be taken into the account to find the most stressed area, or the weakest point of the shaft. The size of the “static” stress concentration factors is depending of the type and the size of changes in the cross section area. Figure 3.5.A) for example is showing a diagram to appreciate stress concentration factor at a shoulder of a shaft subjected to bending. Depending on the relation of the shaft diameters and the relation between the fillet radius and the smallest diameter, it is seen that such factor can be between 1, 1 and 2, 5. From figure 3.5.B) stress concentration factor in a standard keyway in shaft subjected to torque can be found. As seen here, the factor can be between 2,8 and 5,5. The stress concentration K_t taken from the diagrams is for machine members subjected to static stress. Fatigue stress concentration factor K_f will be calculated from the equation below:

$$K_f = 1 + (K_t - 1) \cdot q \quad (3.4)$$

Where q is a notch sensitivity factor.

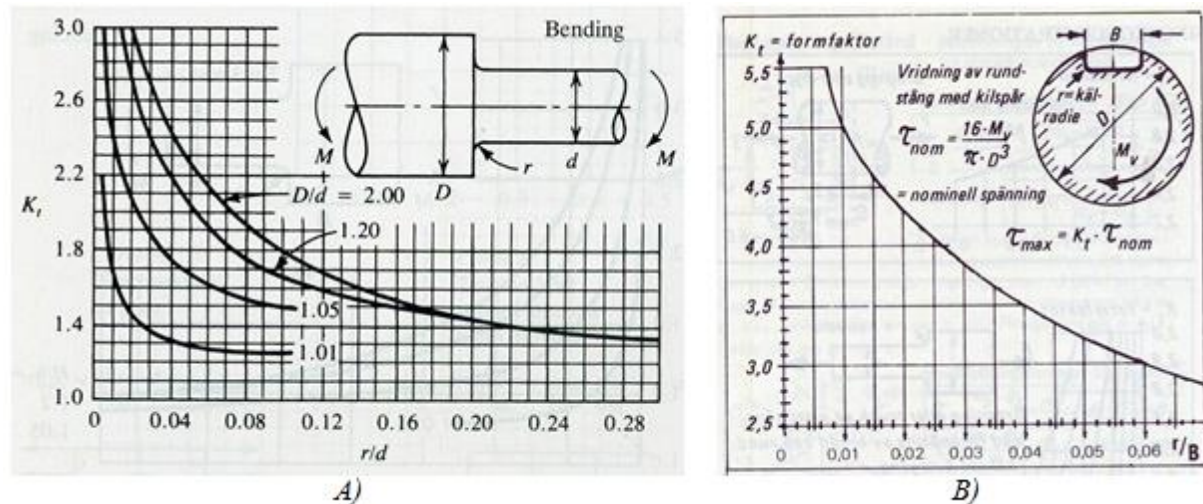


Figure 3.5. A) Static stress concentration factors in a shaft shoulder subjected to bending [Björk, 1991]; B) Static stress concentration factors in a keyway [Björk, 1991]

3.3 Rolling bearing forces and bearing life length

The force analyses of the shaft are including forces and loads on bearings which are carrying the shafts. Figure 3.4. shows the radial loads in bearing A and bearing B in two directions, from which a resultant radial force on each bearing will be calculated. The bearings on a shaft are fixed in axial directions by shaft shoulders or other arrangement; this is to decide which of the two bearings will carry the axial load, (see Fig. 3.6). Life length of the rolling bearings is depended on the size of the bearing, material, lubrication, cleanliness, loads, etc. According to SKF Rolling-bearing catalogue:

where: L_{nm} = SKF rating life (at 100-n% reliability), millions of revolutions

L_{nmh} = SKF rating life (at 100-n% reliability), operating hours

p = exponent of the life equation

$p = 3$ for ball bearings

$p = 10/3$ for roller bearings

n = rotational speed, rpm

C = basic dynamic load rating, kN.

a_1 = life adjustment factor for reliability

P = equivalent dynamic bearing load, kN

$P = XFr + YFa$

Fr = actual radial bearing load, N

Fa = actual axial bearing load, N

X = radial load factor

Y = axial load factor

$$L_{nm} = a_1 \times a_{SKF} \left(\frac{C}{P} \right)^p \quad (3.5)$$

or

$$L_{nmh} = a_1 \times a_{SKF} \frac{1000000}{60 \cdot n} \left(\frac{C}{P} \right)^p \quad (3.6)$$

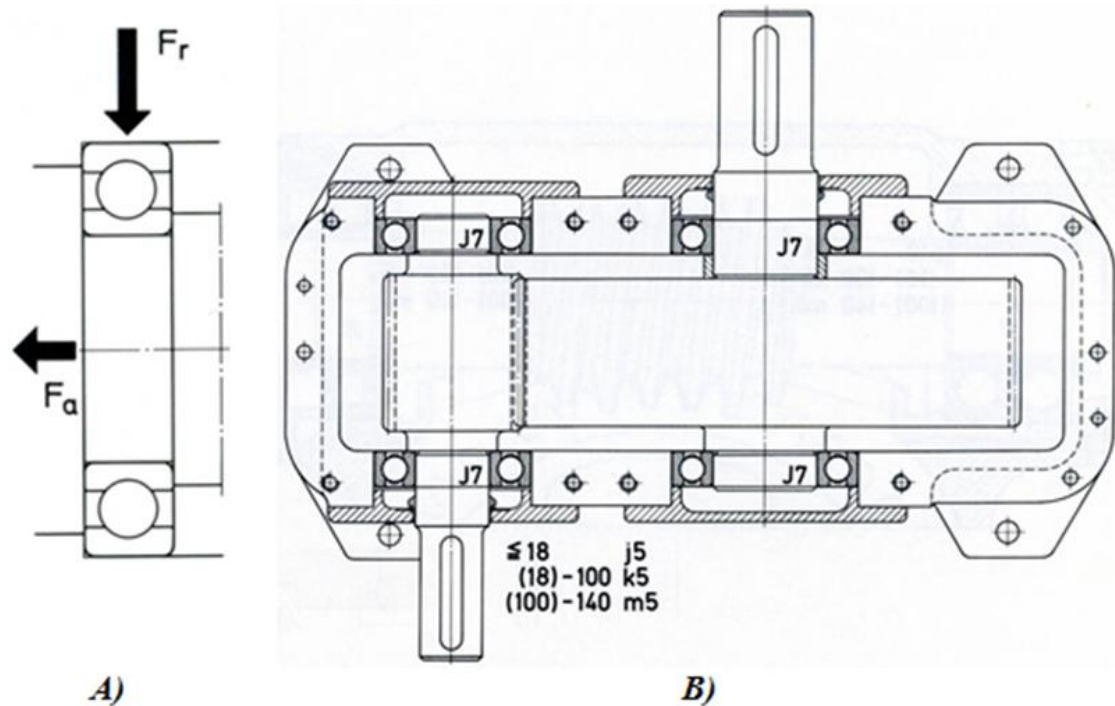


Figure 3.6. A) Loads in a bearing; B) Requirements to the bearing assembly, [SKF. 1975]

As seen above, the rolling bearings are dimensioned through calculating a life length in millions of revolutions or in operating hours. In this case, it is assumed that all the requirements like tolerances (Fig. 3.6. B.), surface finish, form, lubricant, correct mounting etc. are fulfilled. The contact stresses between the rolling elements and the rings, in combination with the material strength is limiting the life length of rolling bearings. These contacts are subjected to variable pressure, hence fatigue stresses (see in Fig. 3.7.).

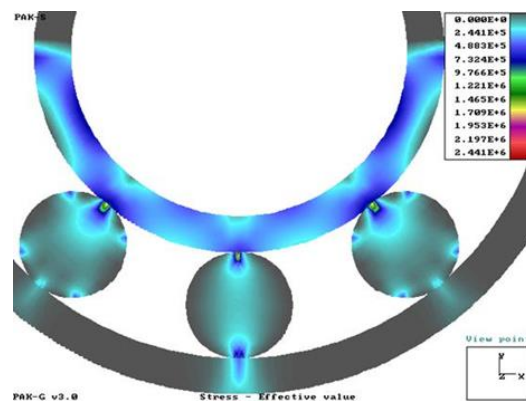


Figure 3.7. Hertz contact stress in roller bearings

3.4 Forces and stresses in seals

The radial seals are usually pressed in the gearbox housing and placed in correct position with the “soft” lips in contact with the shaft. There must be a certain pressure between the seal lips and the shaft surface to prevent leakage of lubricant, and penetration of contaminations, water, etc. from outside. In this case, the only force acting on the seal will be the friction force between the lips and the shaft. This force is of course a product of the normal force (because of the pressure between the lips and the shaft) and the friction coefficient. The size of friction coefficient is depending on material combinations and surface finish. Lubricants has also huge impact on the friction coefficients, it will act like a slide bearing with separated surfaces by an oil film thickness when right conditions are fulfilled.

4. FAILURES AND FAILURE ANALYSES

Machine elements/parts are designed by analyzing kinematics, load, and stresses, together with proper material selection to enable stress-strain-strength evaluation in term of safety factor. One of the first questions in designing these parts is if it will fail in service. A machine element/part is considered to have failed [Hamrock, 1999]:

1. When it becomes impossible to operate
2. When it is still possible to operate but is unable to perform its intended function satisfactory
3. When serious deterioration has made it unreliable or unsafe for continued use, and must be immediately removed to be replacement or repaired

There are many types of failures and failure modes in machine parts. Some of these failures are because of deformation like: yielding, bending-wrapping-twisting, buckling, brinelling, and melting. Some others failures are due to fracture (separation) like: ductile fracture, brittle fracture, fatigue fracture and cracking. There are failure modes due to the surface/material change as well like: ageing, corrosion, stress corrosion, wear. Many types of wear (pitting, spalling, etc.) are occurring in gears and bearings. According to [Polak, 1999] failure in gearboxes are in the most of the cases 49% caused by bearings, while 41% by gears, and 10% by other parts of the gearboxes (see Fig. 4.1.).

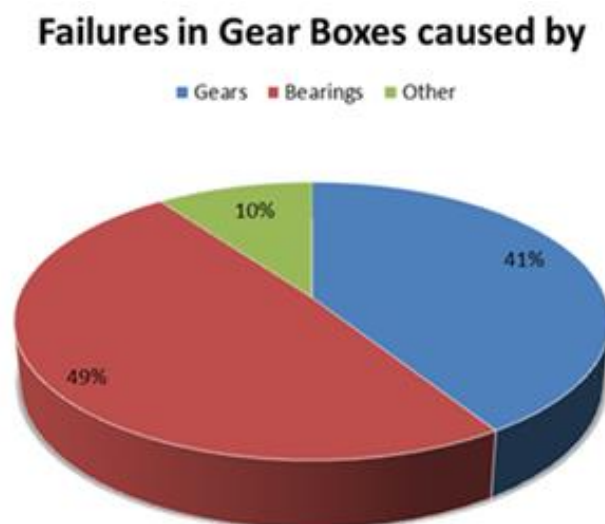


Figure 4.1. Failure in gearboxes caused by gears, bearings and other parts

Failure in one part of a gearbox can be caused by another part, therefore it is very important for example when analyzing gear failure firstly to pay attention to gears, shafts, bearings, lubricant system, seals, and housing.

4.1 Gear failures

Gear failures are divided in two groups, i.e. tooth breakage (overload breakage or fatigue breakage) and surface damages like surface fatigue (pitting, spalling), wear (running-in wear, abrasive wear), and scuffing (adhesive wear). We can also say that the first group is strength-related failure, and the second one is lubricant-related failure. Gear failure modes are classified according to ISO 10825 and are presented in table 4.1. [Van Beek, 2009].

Table 4.2. shows that general failure causes by design where material selection is included, is reason for fatigue in the tooth and corrosion film wear and pitting on the flank. It also causes plastic

deformation, brittle fracture, ductile fracture and electrochemical corrosion in whole gear (including tooth and flank).

Table.4.1. Gear failure modes classified according to ISO 10825

1. Indications of surface disturbances	1.1	Sliding wear: running-in wear, moderate wear, polishing, abrasive wear, excessive wear, moderate scoring, severe scratching, and interference wear.
	1.2	Corrosion (chemical corrosion, fretting corrosion, and scaling.
	1.3	Overheating
	1.4	Erosion (cavitation erosion and hydraulic erosion)
	1.5	Electric erosion
2. Scuffing		
3. Permanent deformations	3.1	Indentation
	3.2	Plastic deformation (by rolling, by tooth hammer)
	3.3	Rippling
	3.4	Ridging
	3.5	Burrs
4. Surface fatigue phenomena	4.1	Pitting (initial-, progressive-, and micro pitting)
	4.2	Flake pitting
	4.3	Spalling
	4.4	Case crushing
5. Fissures and cracks	5.1	Hardening cracks
	5.2	Grinding cracks
	5.3	Fatigue cracks
6. Tooth breakage	6.1	Overload breakage (brittle-, ductile-, and semi brittle fracture)
	6.2	Tooth shear
	6.3	Breakage after plastic deformation
	6.4	Fatigue breakage

According to [Geitner, 1999], stationary gear failure causes are distributed as following: Vendor problems 36%, operating problems 47%, and extraneous influences 17%. Failures caused by materials of construction are making 7% of the total causes. But if we will relate these causes only to the vendor problems, that will give us a rate of about 20%. This means material selection and material property will affect the life of the gears in 20% of the cases. A very important aspect of a failure investigation is through metallurgical analysis. It shows that some gears are cut and finish-machined with much more attention than is given to them during heat treatment. Hence, many failures have been caused by improper metallurgy.

Figure 4.2. shows that 19% of the gear system failures are because of misalignment. Misalignment in the system will cause both gear and bearing problems (except for self-aligning ball bearings and spherical roller bearings). When dimensioning and designing gears, it is assumed that whole the contact area between the gear flanks is used, but in the case of misalignment, shaft deflection, improper form and position tolerances, improper housing dimensions, etc. will reduce the actual flank contact area and will increase the contact pressure which can lead to different types of failures.

Misalignment is probably the most common single cause of failure. In gears it is exhibited as premature pitting at one end of the tooth [Polak, 1999].

Table 4.2. The relationship between failure modes and general causes in gears [Heinz, 2012]

General Failure Causes	Failure modes										
	Tooth	Flank				Whole the gear					
	Fatigue	Scuffing (Adhesion)	Abrasion	Corrosion Wear	Pitting	Plastic Deformation	Brittle Fracture	Ductile Fracture	Cycle fatigue	Hardening	Grinding
Design and material selection	*			*	*	*	*	*			
Manufacturing process										*	*
Assembly	*										
Working conditions		*	*	*	*	*	*	*	*		
Improper operation	*					*	*	*			
Maintenance	*	*	*	*				*			

Investigations in gearboxes of cranes and lifting machineries in southern Sweden show that the gear problems in many cases are due to misalignment (see Fig.4.3.). Since gearboxes for lifting machineries are of short series, the housings and sometimes the gears are manufactured by welding.

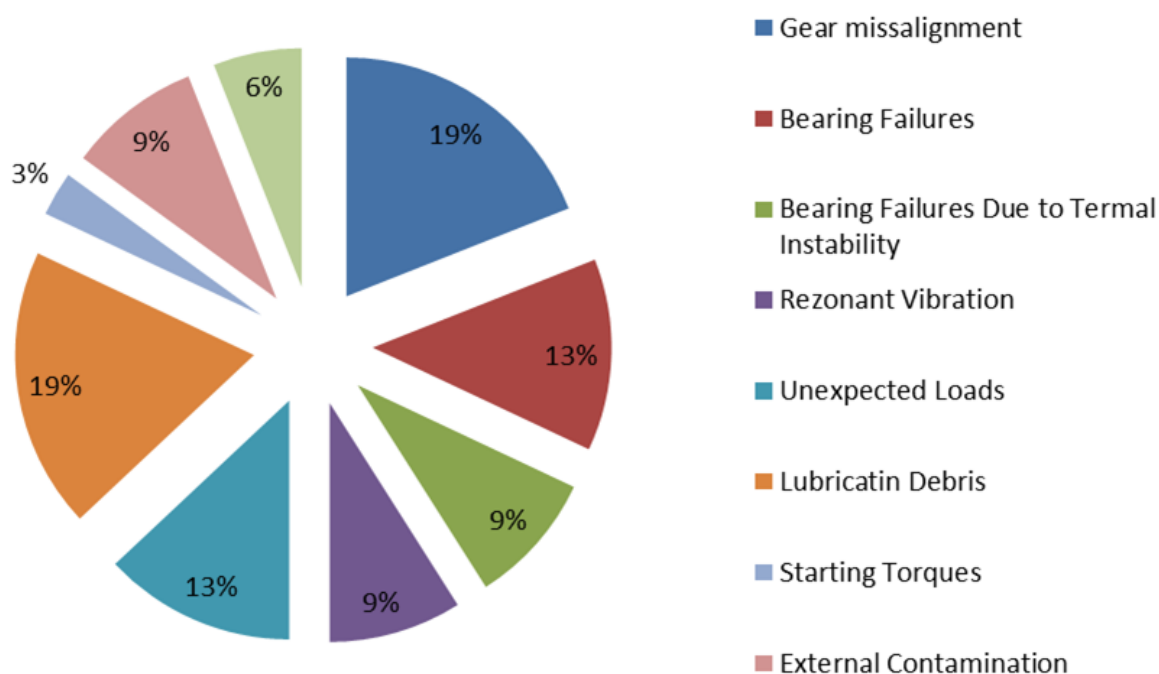


Figure 4.2. Detailed frequencies of failure reasons, [Polak, 1999]

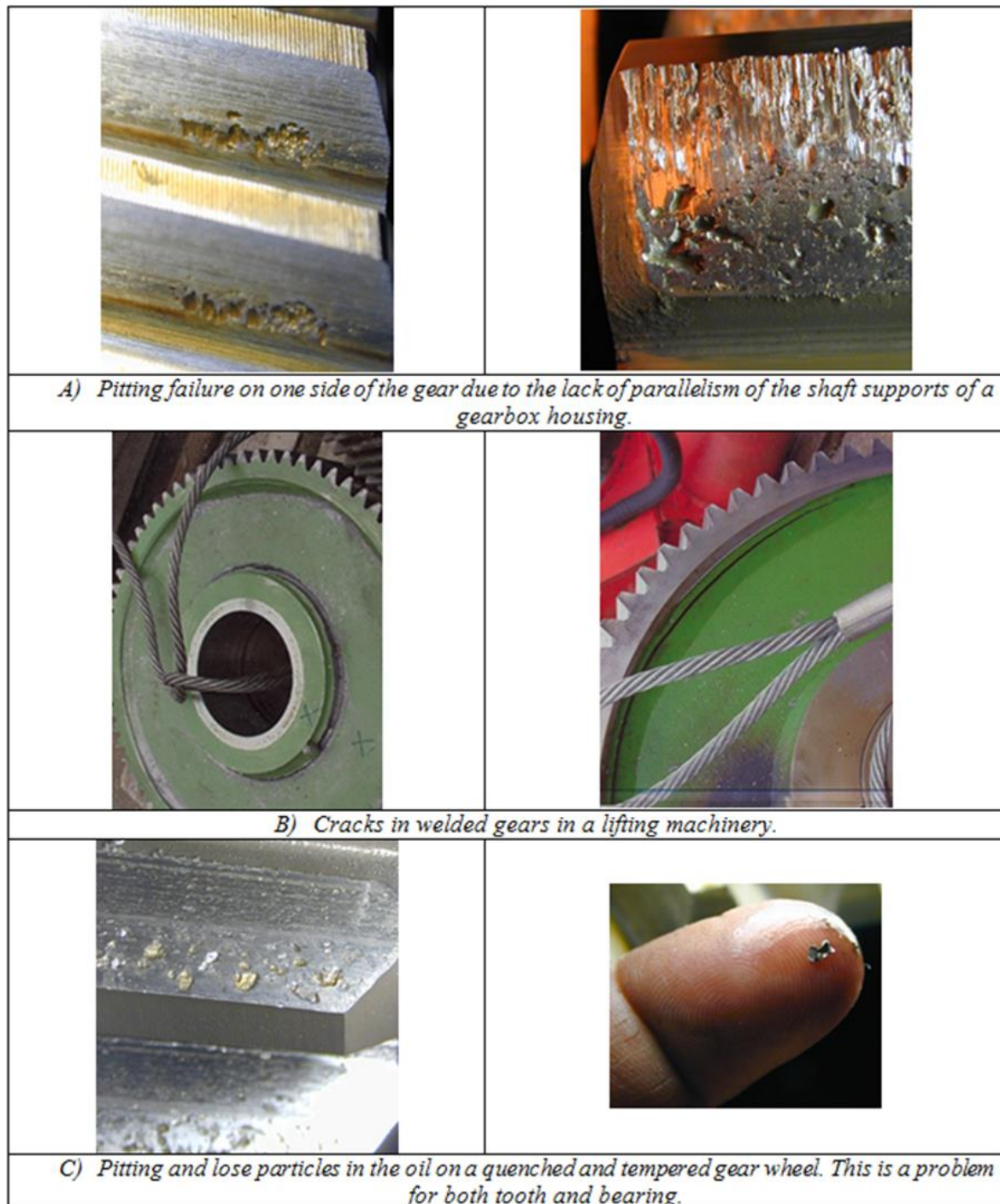


Figure 4.3. Results of investigations of failures in gearboxes of cranes and lifting machinery

Welding makes the parent material, close to the weld joints, brittle, hence sensitive for cracks. The heat during the welding process produces residual stresses and also changes in the shape of the structure, or wrapping because of weld heat. These problems are cause of problems in gears and in some cases in bearings. Some disadvantages of welded joints are that they produce residual stresses; they distort the shape of the piece, metallurgical changes occur, and disassembly is usually a hard problem [Hamrock, 1999].

Residual stress distortions can give tolerance failures in steel structures in general. Gearboxes with bad tolerances, due to welding problems like wrapping, when connected to motors and rope drums can result in dangerous failures and crashes. Figure 4.3.A) shows gearbox cases where the gear flank is

loaded only on one side due to an incorrect shaft parallelism, what eventually resulted in pitting defects. Other problems in welded material are in the gear wheel itself, figure 4.3.B) is showing cracks in a weld joint (or close to it) for the reason mentioned above in combination with fatigue stress.

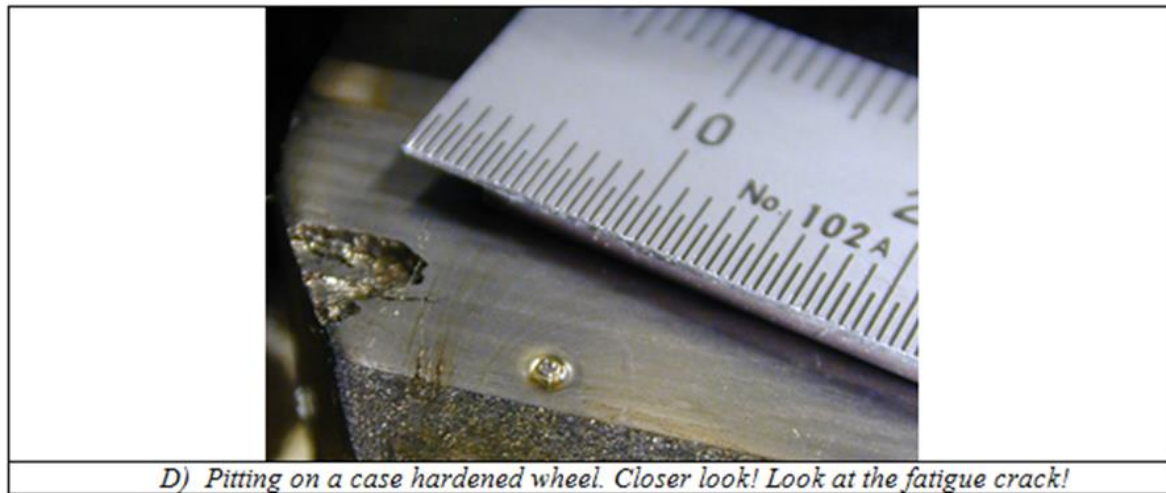


Figure 4.3. Results of investigations of failures in gearboxes of cranes and lifting machinery (continued)

One more example from inspection is showing that pitting and loose particles in the oil are making, like abrasive wear in the gears and bearings and eventually fatigue cracks (see Fig. 4.3.C&D).

4.2 Shaft failures

The most common cause of shaft failures is fatigue. Fatigue is the primary mode for more than 90% of mechanical failures. Fatigue starts at the most dynamic stressed place in a shaft, so called stress raisers, which may be metallurgical or mechanical in nature, or both [Geitner, 1999]. Shaft fractures originate at stress-concentration points either inherent in the design (keyways, fillet, holes, etc.) or introduced during fabrication (grinding damages, machine marks, quench cracks, etc.). As mentioned above, stress concentration factor at a keyway on shaft can be up to 5, which means that the stress in that particular place can be 5 times the nominal stress. When a crack has started at the stressed area, it will develop at any direction like shown in Fig. 4.4, depending on cyclic loads.

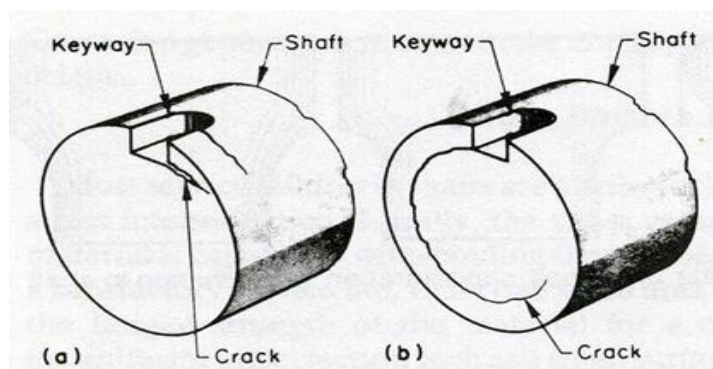


Figure 4.4. Torsional-fatigue cracks in shaft with keyways [Bloch, 2012]

There are metallurgical factors which are acting as stress concentrators under certain conditions and may originate fatigue fracture. These metallurgical factors are internal discontinuities such as porosity, large inclusions, laminations, forging bursts, etc. A shaft stressed by repeated high tensile loading must be free from serious imperfections.

For shafts subjected to one way bending, a crack starting at a stressed area will develop until the failure will appear. When the shaft is subjected to two way bending, may happen that two cracks will develop to fracture at the same time, like in the case of reversed bending. But when a shaft is loaded

by rotational bending a number of cracks can be initiated and developed until fracture. Another investigation result of gearboxes in cranes and lifting machinery in Southern Sweden shows a shaft which failed during service (see Fig. 4.5.). The first sight shows that the crack/cracks started at shoulder with fillet radius. Closer inspection showed that the fracture was caused by multiple cracks which had started at the surface.

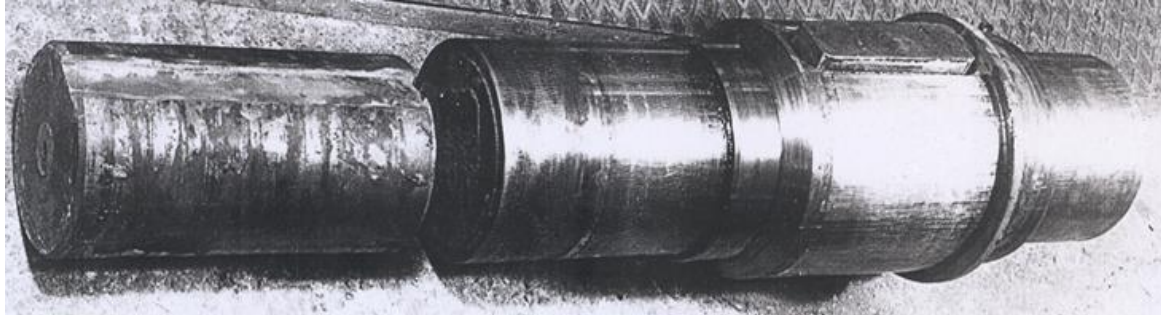


Figure 4.5. Shaft failed because of fatigue fracture originated from many places on the surface

The cranes and lifting machineries which are included in the inspection results were designed and manufactured in 70's and 80's of the last century. It has shown that some of the shafts were designed for static loads, but with high safety factors to compensate for fatigue loading. It has also shown that some of the stress raisers have been placed in high stressed areas of the shaft which has resulted in failures. It is not possible to know about the material, and the strength data that has been used in calculations.

Table 4.3. Standardized classification of bearing failures

1. Fatigue	1.1 Subsurface fatigue	
	1.2 Surface initiated fatigue	
2. Wear	2.1 Abrasive wear	
	2.2 Adhesive wear	
3. Corrosion	3.1 Moisture corrosion	
	3.2 Frictional corrosion	3.2.1 Fretting corrosion
		3.2.2 False brinelling
4. Electrical erosion	4.1 Excessive voltage	
	4.2 Current leakage	
5. Plastic deformation	5.1 Overload	
	5.2 Indentation from debris	
	5.3 Indentation from handling	
6. Fracture	6.1 Forced fracture	
	6.2 Fatigue fracture	
	6.3 Thermal cracking	

High strength materials are more sensitive for fatigue stresses than the low strength material. For example, if a shaft with a diameter of 70 mm made of a steel having yield strength of 1200 MPa and an ultimate strength of 1450 MPa with a fine-ground surface, seems to be strong shaft. But what will the fatigue limit be if that rotating shaft is subjected to a combination of bending moment and torque (assuming bending moment to be equal to torque) and 95% reliability? Taking into consideration the surface, load, gradient, and reliability factors in addition to the alternating bending stresses and static torsional stresses, the fatigue limit will be only approximately 275 MPa. If we will calculate with a safety factor of 1,5 and a stress concentration factor of 2,5, then the allowable stress in the highly stressed area should not be more than approximately 75 MPa. This example is showing that shafts subjected to fatigue stress should never be dimensioned for static stresses even if a high safety factor would be used.

4.3. Bearing failures

It says that only 0,5% of the bearings are replaced because of bearing failures. 34% of the failed bearings are failed because of fatigue, 36% by inadequate lubrication, 14% contaminations and 16% by other causes [Van Beek, 2009]. Rolling bearings operate under high contact (Hertzian) stresses, and are susceptible to the effects of small debris particles in the lubricant. These may well be the reasons why the bearings frequently suffer before the gears. Also causes of bearing damages are classified according to the ISO standard (ISO 15243), see table 4.3.

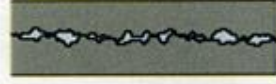
Gearboxes adjacent to hot machinery or hot processes may suffer misalignment from thermal growth. The hot side of the gearbox expands more, resulting in shaft center distances increasing on that side. Misalignment effects tend to be more severe on larger units. This is because the required accuracy of mesh contact (or bearing alignment) does not vary substantially with size, whereas the difficulty of achieving it is greater with large units. As mentioned above, misalignment and deflection in the system will cause problems and failures in both gears and most types of rolling bearings. In this case the bearings will be loaded in a way that they are not designed for.

Good lubrication is one of the most important parameters for a bearings life. For life bearing calculation a required viscosity v_1 will be calculated. This will be used in relation with the lubricants actual viscosity of the lubricant (v) at working temperature. The viscosity ratio (κ) will be calculated according:

$$\kappa = \frac{v}{v_1} \quad (4.1)$$

The viscosity ratio will affect the life of the bearing as shown in the table below:

Table 4.4. Relation between the viscosity ratio and the type of lubrications [Berggren, 2005]

	$\kappa = 4$	Hydrodynamic lubrication	If $\kappa > 4$ the viscos friction will increase
	$\kappa = 1$	Mixed film lubrication	If $\kappa > 1$ the life length will increase
	$\kappa = 0,3$	Boundary lubrication	If $\kappa < 1$ the life length will decrease

Unfortunately, there are no statistics or any documentation from bearing failures in lifting machinery gearboxes investigated in southern part of Sweden. The bearing supplier use to take care of such statistics and investigation. But as seen in other statistics, the material impact on rolling bearing failures is very limited. Most bearing failures can be attributed to one (or more) of the following causes: Defective bearing seats on shaft and in housing, misalignment, faulty mounting practice, incorrect shaft and housing fit, and/or inadequate lubrication.

4.4 Seal failures

A mechanical seal has failed when leakage becomes excessive. Some of the causes of seal failure can be Mishandling of components, incorrect seal assembly, incorrect seal selection, inadequate environmental controls, fluid contamination, worn-out seal, etc. Some types of seal failures due to mechanical action are: face distortion, seal face deflection, tracking, extrusion, and erosion.

One of the seal failures due to heat buildup is heat checking. Heat check which occur on hard faces in seals can cause seal failures, it is caused from excessive heat. The heat check will tell a lot about what is happening inside the seal. Heat check shows up as radial crack marks along the seal face (see Fig.

4.6.). When the seal material has cracked in many places, it will pull up slightly, causing the face acting like a grinding wheel. The later will wear down the opposite face (contact face) much faster. Grinding action, in its turn, will produce more heat, and will damage other parts of the seal.

The causes of heat check can come from one or more of the following scenarios: application temperature, over compression, improper lubrication, dry running, or inadequate cooling.

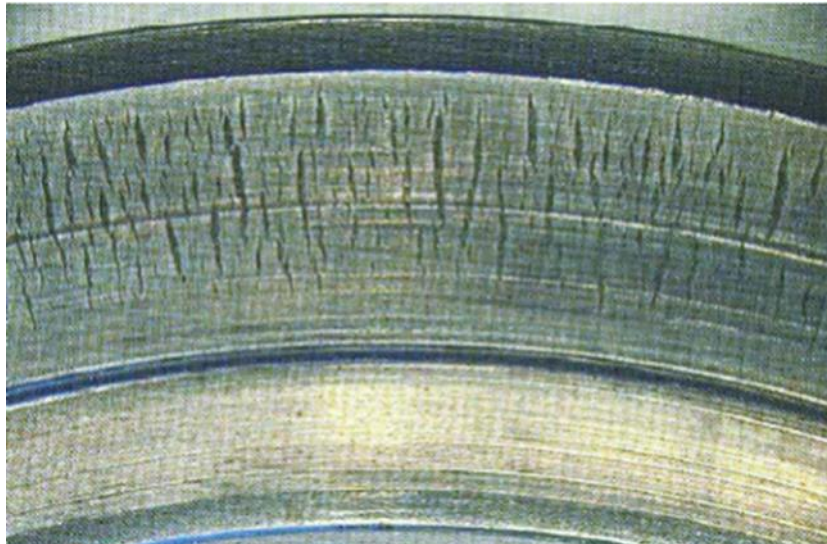


Figure 4.6. Failed seal due to heat check

There are no records about seal failure data either from inspections, but there are document showing gears worn out because lack of lubricant or no lubricant at all. In many cases leakage of lubricant has been due to seal damages.

5. CONCLUSION

This study is based on literature review, statistics taken from different references, and experiences of inspection personnel at an inspection company for cranes and lifting machinery in southern Sweden. In conclusion on lifting machineries high requirements for safety and reliability are set. But failures happen and the consequences are leading to big loses. The outcome of this work shows that it is not easy task to analyze the exact failure cause in such a complicated system. A careful study on the most sensitive parts and the possible failure that could appear in the material due to different reasons could lead to better understanding about the nature of the failure and the possible corrective action that could avoid them. The role of the materials in sensitive parts of gearboxes is related to other issues like loads and stresses, and lubrication for parts in contact with relative motions.

In lifting machineries a big variety of gearboxes are used. But when the sensitive parts, as roller bearings, gears and shafts are recognized and transmission of loads and forces are analyzed, it is easier to find the logic between the circumstances and the failures appeared. Some of the statistics are showing the material causes of failures together with design causes, for example surface damages can also be referred to the assembly of gearboxes.

As mentioned in this article, lubrication has huge impact on the life length of the gearboxes. Failures could appear due to lack of lubricant, failed lubricant caused by moisture or chemical reactions, wrong lubrication type for the current situation, etc. Analyzing failure in gearboxes or any machinery needs correct and wide statistic data, which is not always available for researcher. Sometimes only the manufacturing companies have access to this type of information.

Good theoretical knowledge about materials, tribological theories, and dimensioning and design of gearboxes are very needed to analyze failures in gearboxes. But of big importance is also to have

detailed records of inspections of gearboxes, as in this case, used in lifting machinery. The real life cases and the needed documentation is the source for deep study of failures. Failures are in very close relation to the selected materials. The physical properties of the material, and the geometrical accuracy, together with the right design, proper operation, and regular inspection are the tools to increase the reliability and life length of gearboxes.

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Figure 1.1. - <http://www.ghcranesusa.com/images/blowups/hoist-diagram-big.jpg>

Figure 2.3 - <http://media-2.web.britannica.com/eb-media/54/6354-004-CA4BDEA9.jpg>

Figure 2.4.A) - <http://nguyenxuong.com/upload/images/Seals/SKF%20NGUYEN%20XUONG%20-%20SEAL%204en.jpg>

Figure 2.4.B) - <http://1m28fbq9etc1rj5ga1fu0k61.wpengine.netdna-cdn.com/wp-content/uploads/2016/03/SKF-radial-bearing-shaft-seal-image.jpg>

Figure 2.5.A) - http://www.skf.com/binary/107-124515/0908%200207%20-%2010000_tcm_12-124515.png

Figure 2.7.B) - http://www.nrel.gov/wind/pdfs/day2_sessioniii_2_geartech_errichello.pdf

Figure 3.7. - http://www.itcsoftware.com/general_finite-eg-overview_files/image014.jpg

Figure 4.6. - <http://www.coloradosealsinc.com/wp-content/uploads/2014/09/heat-check.png>