European Union Aviation Safety Agency

SMART GREEN AND INTEGRATED TRANSPORT

Integrity improvement of rotorcraft main gear box

An Agency of the European Union

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DETERMINE DESIGN PARAMETERS

Review of the state-of-the-art design criteria for reliability and flaw tolerance in integrated bearing races and list of identified relevant design parameters (D2-1)

ZF Luftfahrttechnik GmbH

ZF Luftfahrttechnik GmbH is a worldwide company known for its helicopter transmission systems. With our EASA and FAA privileges we are active for design and development, manufacturing and maintenance of helicopter transmission systems and geared applications for fix wing aircrafts and engines. Our service activities are not only keeping transmission system components airworthy, also mission support for a broad range of customers are conducted. For nearly all helicopter manufacturers around the world, ZF Luftfahrttechnik GmbH designs and supplies worldwide turn-key solutions for gearbox and rotor test stands.

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SUMMARY

Problem area

Objectives: This task shall identify the most significant design parameters for Rotor and Rotor Drive System components with case-hardened steel bearing races that will influence the reliability and tolerance to flaws of these components when subject to rolling contact fatigue.

Description of work

Expected outcome: This task aims to provide a list of design parameters that are key to controlling the reliability and flaw tolerance of integrated bearing races when exposed to rolling contact fatigue. The role each of the identified design parameters plays with regard to reliability and flaw tolerance is described in general terms in order to permit accurate planning of the analysis and testing activities to be performed in subsequent tasks.

Task 2 — Define adequate design parameters for component reliability and flaw tolerance
Scope: In order to achieve the objectives stated above, this task shall include the following:
Review of the state-of-the-art understanding available in industry standards, research and other relevant literature with regard to the critical design parameters of gears with integrated bearing races that impact their behavior when exposed to rolling contact fatigue. The design parameters to be reviewed must take into consideration all elements that form part of the definition of the gear itself, plus all elements of the bearing including any integrated bearing races, and those that are defined at the design and manufacturing stages. These include, but are not limited to, operating contact pressure, lubrication film thickness, clearances, surface hardness, case-hardening depth, residual stresses and surface roughness.
Identification of the above design parameters that play a critical role in ensuring the reliability and flaw tolerance of integrated bearing races when exposed to rolling contact fatigue. This shall cover, at the minimum, the most relevant forms of flaws attributable to manufacturing, intrinsic defects, service and handling, including (but not limited to) corrosion, scratches, impact, material inclusions, residual stress variability, grinding burns, micro-pitting and spalling.

Reliability refers to the capability of a component to perform consistently without the development of unexpected modes of degradation or damages
Flaw tolerance refers to the resilience of a component in terms of operation for a safe period of time in the presence of intrinsic and extrinsic defects without developing any form of cracking that may ultimately lead to catastrophic consequences

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Results and Application

The key design parameters stated in Section 3 were selected to provide an overview of the mandatory design parameters for integrated bearing races. The resulting parameters are applicable to both standard bearings and bearings with integrated raceways. Additional parameters were only included for bearings with integrated raceways and planetary gears with integrated raceways. Most of the parameters are primarily influenced during the initial design stage, whereas others are contingent on the operational stage and surrounding components (e.g., lubrication). Some of the parameters are strictly defined by norms and standards, such as ISO or DIN standards. Others are determined by the respective application, customer preferences and the interplay between various interrelated parameters (e.g., operational clearance has a direct influence on the width of the load area and vice versa). For the most part, they all share common ground with their contribution to the reliability and flaw tolerance of the integrated bearing races. Based on the findings from this report, research conducted by SKF, the University of Hannover and ZFL and its relevance in current research projects, the design testing and selection of the correct parameters, isolated for each component, are well established processes, yet the interaction and impact of the parameters during operation remains largely uncontrolled and unresearched to date. In any case, it is possible to focus on a number of the key parameters that contribute to the damage hypothesis for integrated raceways (see Table 7) and their interconnections (see Figure 48). The defined parameters will subsequently be used to determine adequate tests and find solutions to reduce the criticality of catastrophic failure modes attributable to these parameters at later stages in the project.

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ABBREVIATIONS

ACRONYM	DESCRIPTION
ABEC	Annular Bearing Engineering Committee
AT	Automatic transmission
BEM	Boundary element method
CHD	Case hardening depth
CRB	Cylindrical roller bearing
DCT	Dual clutch transmission
EASA	European Union Avation Safety Agency
FE	Finite element
FEM	Finite element model
FVA	Forschungsvereinigung Antriebstechnik (engl.: Research Association Power Transmission Engineering)
HAP	Haupt-Arbeits-Paket (engl.: main work package)
ISO	International Organisation Standardization
LH	Lef hand
MBS	Multibody simulation
MRO	Maintenance, repair and overhaul
PV	Smearing risk factor (product of **P**ressure and **V**elocity)
RCF	Rolling contact fatigue
RGB	Rear gearbox
RH	Right hand
SKF	Svenska Kullagerfabriken
SRB	Spherical roller bearing
TBO	Time between overhaul
TRB	Tapered roller bearing
VAR	Vacuum arc melting
VIM	Vacuum induction melting
WEC	White etching crack
ZF	Zahnradfabrik Friedrichshafen
ZFL	ZF Luftfahrttechnik

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1. Introduction

1.1 Context of the project

This report is a deliverable for the EASA project "Smart Green and integrated transport - integrity improvement of rotorcraft main gearbox (MGB)". The work completed to produce this report was undertaken as part of stream 2, i.e., HAP200, to evaluate design criteria for reliability and flaw tolerance in integrated bearing races and provide a list of relevant design parameters. Previously, during stream 1 of this project [25], current gearbox architectures were described and their criticality was analyzed according to catastrophic failures by flow diagrams. In general, the risk of catastrophic failure for various helicopter architectures (see [25]) was identified not only in planetary gears with integrated raceways, but also in bearings with integrated raceways. In the subsequent project stages, the results of this deliverable, mainly the list of key design parameters, can be used in [26] to plan and define the tests required to evaluate the design limits for the specified parameters.

1.2 Design testing of a gear system

1.2.1 General design and verification process

An example of a general gear system design process can be found below in Figure 1. Load assumptions and load calculations are required as preliminary work. The gear design and verification then form key elements of the design process, with the verification of all individual components, structural analysis of the system, dynamic simulation and testing. As the final step in the design process, operating experience is tested and evaluated as an important milestone to verify the previously completed stages.

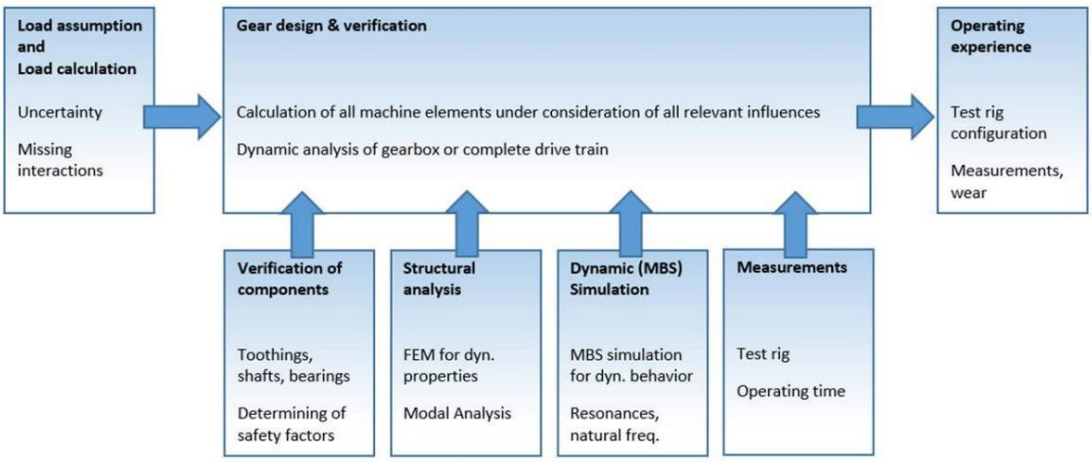

Figure 1: General gear system design [12]

The design process undertaken at the ZF group closely resembles the above approach. Figure 2 illustrates the main aspects taken into account for designing a gear system at the ZF group, including the various development stages, component testing and the verification process with the creation of guidelines for continuous improvement. Internal design guidelines are applied in addition to the pertinent standards and subsequently verified with virtual engineering and system verification with suitable tests.

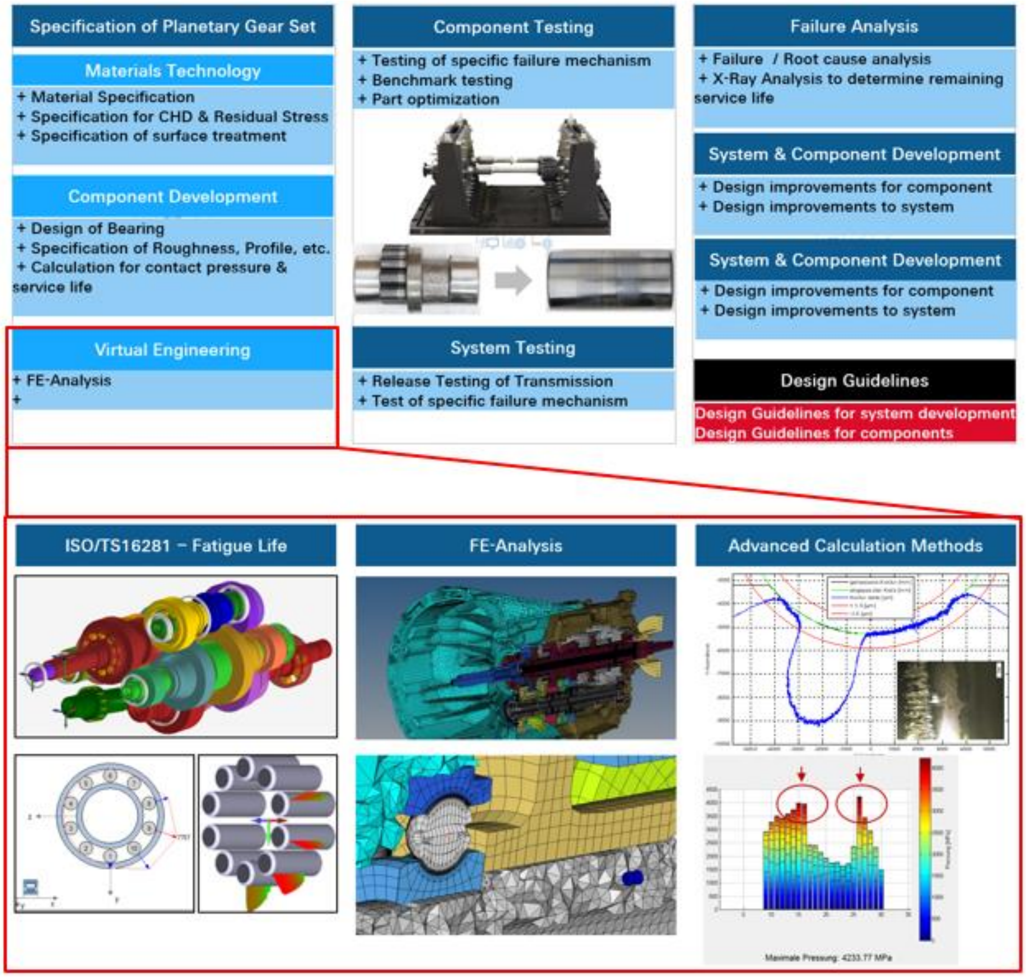

Figure 2: Gear system design at ZF group

An innovative approach for the development and design of a helicopter gearbox can be found in [24], which provides an insight to actual research topics.

1.2.2 Assessment of gear system with integrated raceways

In general, a gear system with conventional bearings is assessed in a similar way to a gear system with integrated raceways. However, in the case of bearings with integrated raceways, certain additional issues need to be accounted for (e.g., choice of material, rolling element profile, etc.). Generally, gear system and its bearings are tested independently on the basis of a number of available standards and the manufacturer's knowledge and experience.

The calculation of the estimated bearing life (e.g., according to ISO 281 [61]) is a key aspect to the bearing development and can be a first approach for the bearing design. Many customer-specific tools have been developed on the basis of the applicable standards with useful and proven additions to perform this test and support the design process. As a starting point for the design and selection of the right bearing for the respective application, the key aspects need to be defined in a specification. This specification also includes the selected material (e.g., M50), installation conditions, environmental conditions, the lubrication situation and loads with their respective limits according to pressure stress levels (see also ISO76:1988 [62]). These levels are defined for static and dynamic loads in the pertinent standards and may also be subject to restrictions with stricter limits or safety factors based on the experience of the gearbox/helicopter manufacturer. Typical values for helicopter applications are below 1600 MPa in the nominal condition and in a range of 2200-2400 MPa in the maximum condition based on the experience of SKF. These values can vary greatly depending on the final application and the spectrum of operating conditions it is subject to. The static limits according to the limit load consideration are normally higher by a factor of 2. The bearing loads can be determined using the load spectra of the helicopter. To facilitate this, ZF developed a tool to provide the bearing loads including the respective tilting moments for a bearing assessment. Moreover, this tool can also be used to review the change in the load situation for bearings while changing the distance between the bearings of a shaft or simulating a failure of one of the bearings (see Figure 3). These tools are often not considering the bearing's rings deformations. Nevertheless, in some applications the deformation of the bearings rings can lead to a non conventional load distribution on the bearing rolling elements.

This is for instance the case on planet gear bearings. For this type of applications, analytical approaches have been developed to determine the load distribution on the bearing rolling elements (e.g. [1]). However, in most cases, the deformation of bearing's rings cannot be determined analytically and require the use of FEM, as presented in Figure 2. Such models also allow the evaluation of rolling element load distribution and contact pressures as well as bearing rings stress levels considering the deformation of gears and supporting housings.

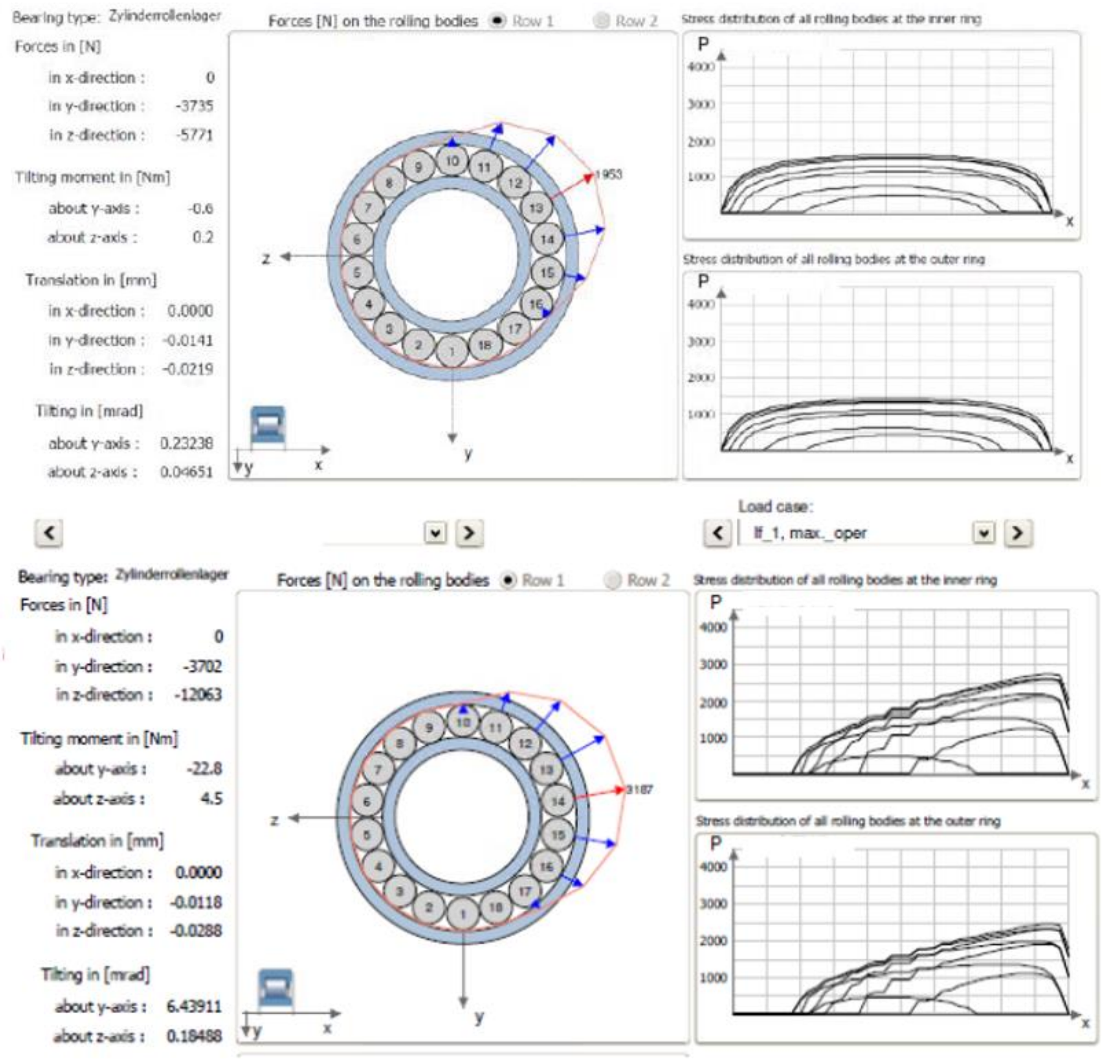

Figure 3: Example of the tool Genia, used by ZFL for bearing assessment

In addition to the above calculations/simulations, the following aspects are mandatory for the bearing design and will be explored in more detail in Section 3 ([3],[4],[20],[21]):

- Roller contact profile
- Lubrication
- Material and heat treatment
- Stress state and level

Current problems of gear design/assessment are represented in a good way by the research topics of the cluster project “planetary gears” led by the FVA (e.g. [2]).
Figure 4 provides an overview for the development of standards according to gear calculation and rating. Studies and reports are also available from the FVA (engl.: Research Association Power Transmission

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Engineering), indicating that the calculation and rating of gears remains a current research topic with scope for further research, in particular for systems with integrated raceways that combine bearing and gear testing.

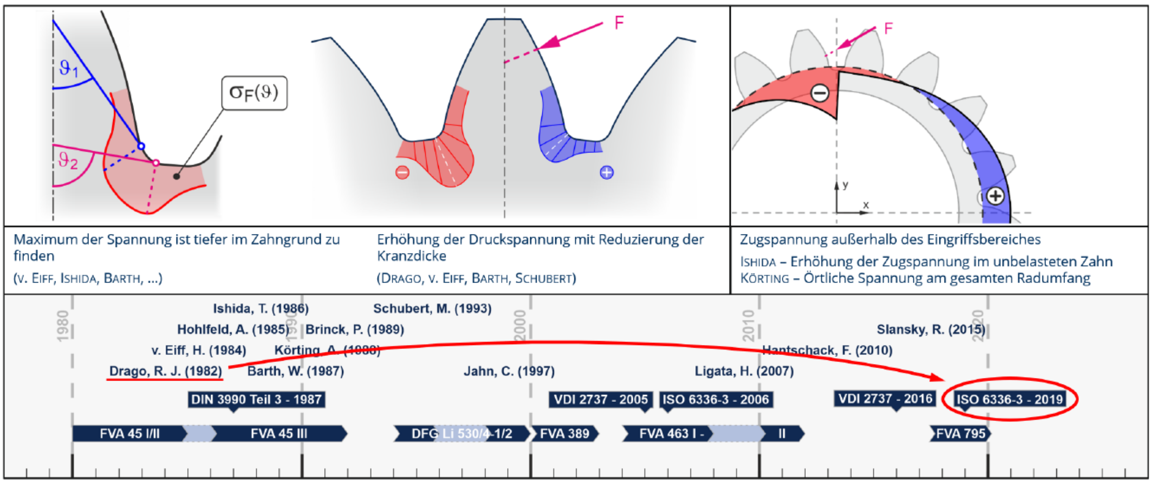

Figure 4: Extract of literature for elastic design of inner and outer gear meshing [2]

FVA research projects contribute significantly to the reliability of current gear system designs by helping to close the gap between the standards and experiences gained from use. Additional examples of ongoing and completed FVA projects on gears and bearings are listed in Annex A summarizing some of the research topics of the past.

FVA research project FVA795 – Tooth root and ring strength of thin rimmed planetary gears [2] was initiated by current changes in the design approach for planetary gears. The project looked at the tooth root and rim reliability of thin-rimmed planetary gears with integrated raceways on the basis of initial crack formation at the tooth root.

In addition to FVA projects, several other projects have been conducted in the past to improve the understanding of planetary gear systems and the rating of gears, not just planetary gears. Findings from research on these topics can be integrated into the development and design process for every gear system to increase reliability and flaw tolerance, and mitigate the number of catastrophic failures. However, the reliability and flaw tolerance of gears, especially planetary gears with integrated raceways, are not only contingent on the design process. The operational stage also plays a key role. Several aspects need to be taken into account during the design stage before entering the operational stage.

Figure 5 illustrates the possible contributing factors to crack initiation or propagation to demonstrate the wide range of required reliability aspects. The upper part (design and manufacturing stage) describes contributing factors during the design and manufacturing stage in terms of material and heat treatment, bearing design and gear design. As is standard in the aviation industry, these parameters are designed in a way that prevents conventional fatigue or crack initiation under the conditions defined in the specification for the gearbox. The underlined parameters will be explored in more detail in Section 3. Some other points (e.g. fracture toughness and fatigue limits) are often predetermined according to the application and use case and don't allow for a wide range of adjustments. The range of these values are typically predefined by the choice of material, whereas the material choice itself is based on the application and is limited to just a few different materials.

Nevertheless, the fracture toughness of a material can be influenced by the manufacturing and heat treatment processes (e.g. hardness or case hardening depth). The lower part of the diagram shows the operational stage with the factors that contribute to component reliability under the headings of operating conditions, lubrication and pre-damage. Due to the gear architecture principles and design principles, the factors that have an effect during the operational stage cannot be eliminated entirely. However, they can be minimized (e.g. by the adequate use of oil filters, the choice of oil type and smart design solutions). Nevertheless, the factors occurring during the operational stage may impact reliability while being relatively hard to avoid, as they are not generally influenced by the individual design of the component.

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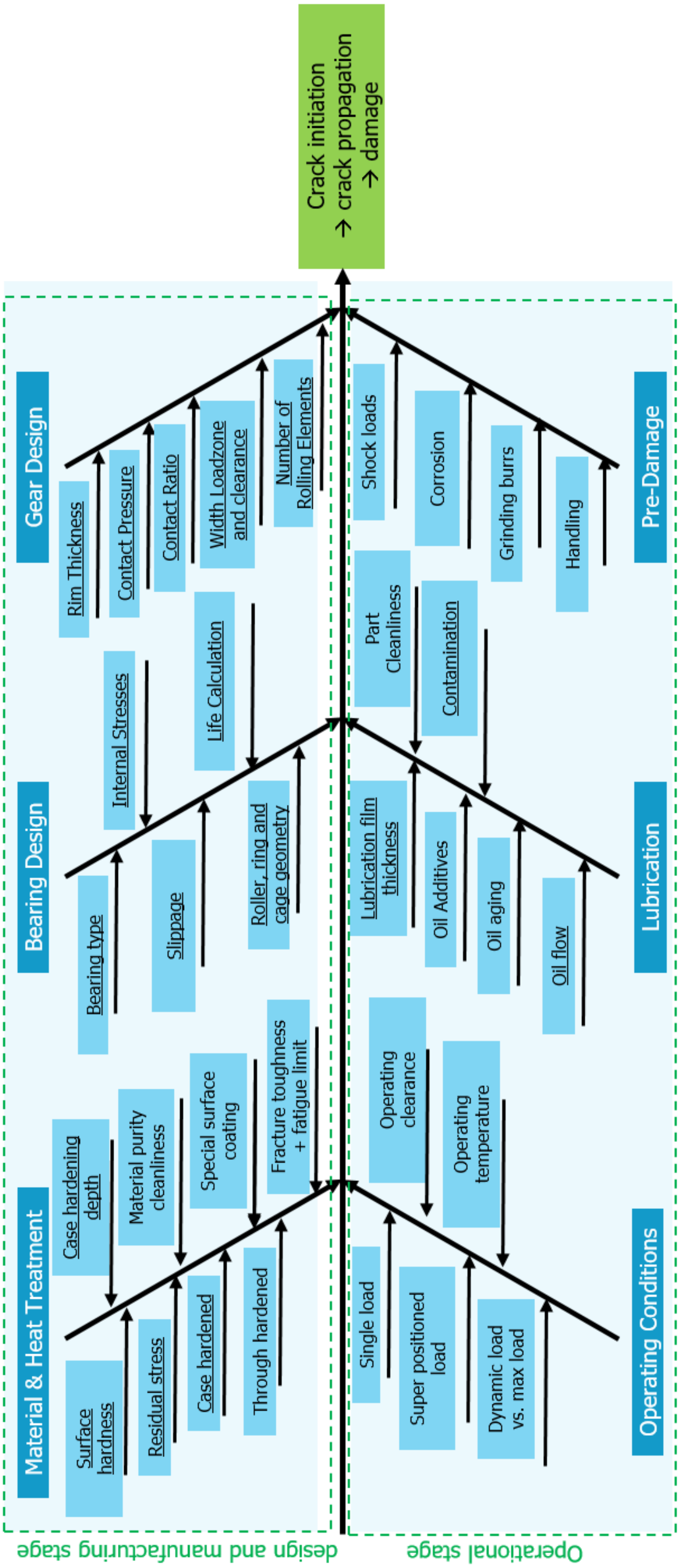

Figure 5: Fisbhone diagram of conrtributing factors for reliability assessment

Some design solutions, especially planetary gears with integrated raceways and thin rims, are challenging for designers to realize, as some of the basic guidelines are not fully applicable or archievable. The conventional guidelines typically do not consider the full interaction of the gear and the (integrated) bearing and thus do not consider the deformation of the planetary gear system in detail (see Figure 6).
Research on bearings with a deformable outer race has been conducted in the past (e.g. by Harris and Jones [59]). In particular for aircraft applications, bearing rings and their supports can not be considered rigid. The elastic behavior of the outer race and the impact of this on the internal load distribution lead to a significant reduction in the bearing's fatigue life and must be taken into account during the design process [59]. Even standard material and heat treatment processes need to be reconsidered in the case of bearings with integrated races, as they could lead to insufficient reliability of the components [60].

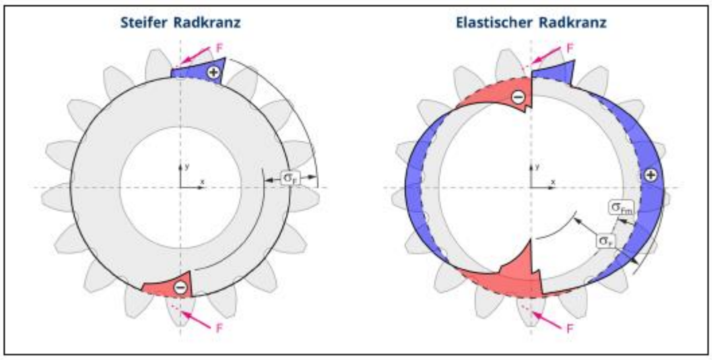

Figure 6: Comparison of conventional gear with stiff rim (left) and elastic rim as for planetary gears (right) [2]

In general, standard design procedures are consequently replaced with approaches tailored to the individual cases, predominantly supported by virtual engineering solutions, the manufacturer's experience or results of research projects. Conventional stress optimization of the tooth root stress, as performed for conventional gears, may cause issues in terms of crack formation when combined with a thin rimmed gear and will not ensure sufficient reliability of the component. This subsequently requires an in-depth evaluation, completed separately to the individual gear rating of conventional gears. Figure 6 provides an example of a detailed FE simulation for a planetary gear from ZF Wind taking into consideration the planetary gear deformation.

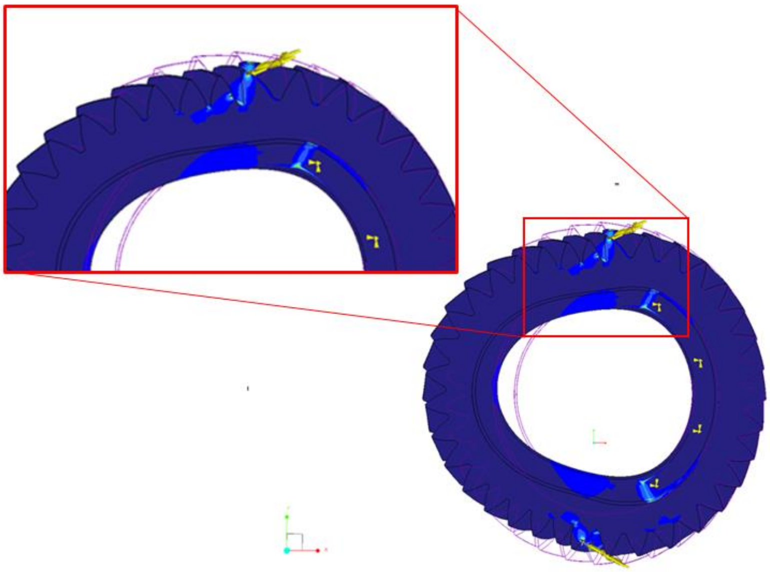

Figure 7: Example of FE evaluation of planetary gear with integrated bearing races at ZF Wind

At ZF, there are internal design rules to help the designer choosing the right design for a reliable gear. This guidelines provide some basic rules for the design of a gear, for example the minimum rim thickness in correlation to the tooth height. In addition to this parameter, some main points, which are typically considered within the design rules at ZF are the following:

- Gear rating specification (e.g. lifetime, loads)
- Oil data
- Calculation rules (e.g. load or safety factors)
- Gear quality specification (e.g. roughness, case hardening depth)
- Material specification (e.g. minimum elongation at break)
- Surface imperfections (e.g. limit surface tolerances)

However, overall research into the interaction between resulting stress levels at gears and shafts with integrated raceways between bearing race and gear meshing during the design process is currently not state of the art and requires further study. This lack of research becomes increasingly important in relation to integrated raceways. When coupled with an unfavorable choice of design parameters, this can lead to degradation of the bearing and gear assembly in terms fatigue life despite the fact that the components are designed according to the current standards. Accordingly, ZF, and all other gear manufacturers, are required to regularly update the ratings of gear systems in line with new findings and if necessary, to adjust the gear design. Finally, further research into the surface, lubricant and material properties is required to reach a higher level of accuracy and reliability [18].
Summarizing the mentioned aspects above, the following main design parameters for the rating of gears, based on available research projects, [1], [2], [8], [12], [13] and [22], can be pointed out and will be explored in more detail in Section 3:

- Load situation and deformation
- Stress level
- Material and heat treatment
- Geometrical parameters
- Operational influences

1.3 Use-cases and experience with integrated raceways at ZF

Section 1.3 explores the various applications for integrated bearing races and typical properties, differences and also similarities of use cases. In particular, similarities can be used to gain a better understanding of key design parameters for integrated bearing races and expertise obtained from a diverse range of engineering topics can be applied. Findings from the evaluation of similar issues in different applications help to push project progress, reach the right conclusions and generate beneficial results, not only for the aviation industry. As the ZF group is involved in multiple fields of applications, data is available for the aviation, wind power and automotive industry. A general overview on integrated bearing races can be found in Table 1.

Bearings with integrated raceways at ZF	
Industry	· Aviation · Automotive (car, truck, commercial vehicle) · Wind Power
Integration location	· Shafts · Planetary gears
Manufacturer	· SKF · FAG · Timken (Wind Industry)
Bearing types	· CRB · TRB · SRB · Ball bearings
Materials	· Bearing races are made of non-structural through hardened steel (100C6 & M50) or Case-hardened/structural steels (32CDV13; M50NiL; 16NCD13 but also AISI9310, VascoX2, Pyrowear 53) · Rolling elements are made of M50 or 100C6 or eventually Si3N4 · For aerospace high speed bearings, only steels produced by VIM-VAR process are used · Silver plated 40NCD7 or brass alloys cages are mainly used

Table 1: Overview of integrated bearings at ZF

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Integrated bearing races are used in a wide range of different applications, but the reasons to use integrated bearing races instead of conventional bearings are often quite similar. Table 2 features a general comparison between conventional bearing systems and integrated bearing systems, for the aviation industry, for instance.

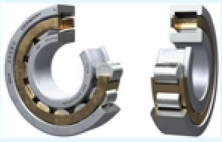
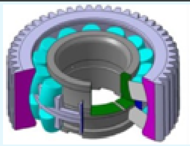
	Conventional rolling bearings	**Integrated rolling bearings**
		
PROS	+ Less engineering effort + Limited influence of the shaft material + Larger range of potential steels for selection + Crack propagations stops at the inner ring (safety barrier)	+ less components → weight & space savings, tolerance stack-ups + no ring creep → no fretting, debris, ... + increased load capacity + high power density
CONS	- more material and components used - speed limitation - space for inner and outer ring needed - Ring creep possible - Require means to maintain rings in position (interference and/or tightening nuts) - Risk of fretting at the interface between parts	- high demands on gear/shaft material (material of inner/outer race must be a bearing material) - added engineering effort - complex manufacturing process - failure of outer ring → failure of gear

Table 2: Comparison standard bearings and integrated bearings ([12],[14], [19],[21],[22],[27])

In principle, integrated bearing races are employed to save space, use less components, increase the load capacity and generate a high-power density with improved load sharing. With these properties, integrated raceways can contribute to the key contemporary engineering demands for cost reduction, smart system solutions and sustainability [14]. As illustrated in Table 2, there are several pros and cons that need to be weighed up prior to selecting the right bearing system for the application. Weight and space savings in combination with the high-power density are often behind the reason to use integrated raceways. However, higher engineering and manufacturing costs and the risk of damage/failure to the raceway components (e.g., shafts) also need to be accounted for during the design process. Integrated raceways can be found on shafts, gears or even housings of gearbox systems. Integrated raceways can often be found in planetary gears, which are typically used in applications with less space and high-power densities.

1.3.1 Aviation

ZFL is experience in using integrated bearing raceways and regularly works with them throughout the design process and as part of MRO. Figure 8 provides an overview on ZFL experience with products containing integrated bearing raceways.

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Figure 8: Integrated bearing raceways within ZFL field of experience

Shaft bearings and planetary gear bearings are the main components used in combination with integrated raceways. Figure 9 features an example of the integration of integrated bearing raceways on shafts, as is the case for current MGBs at ZFL.

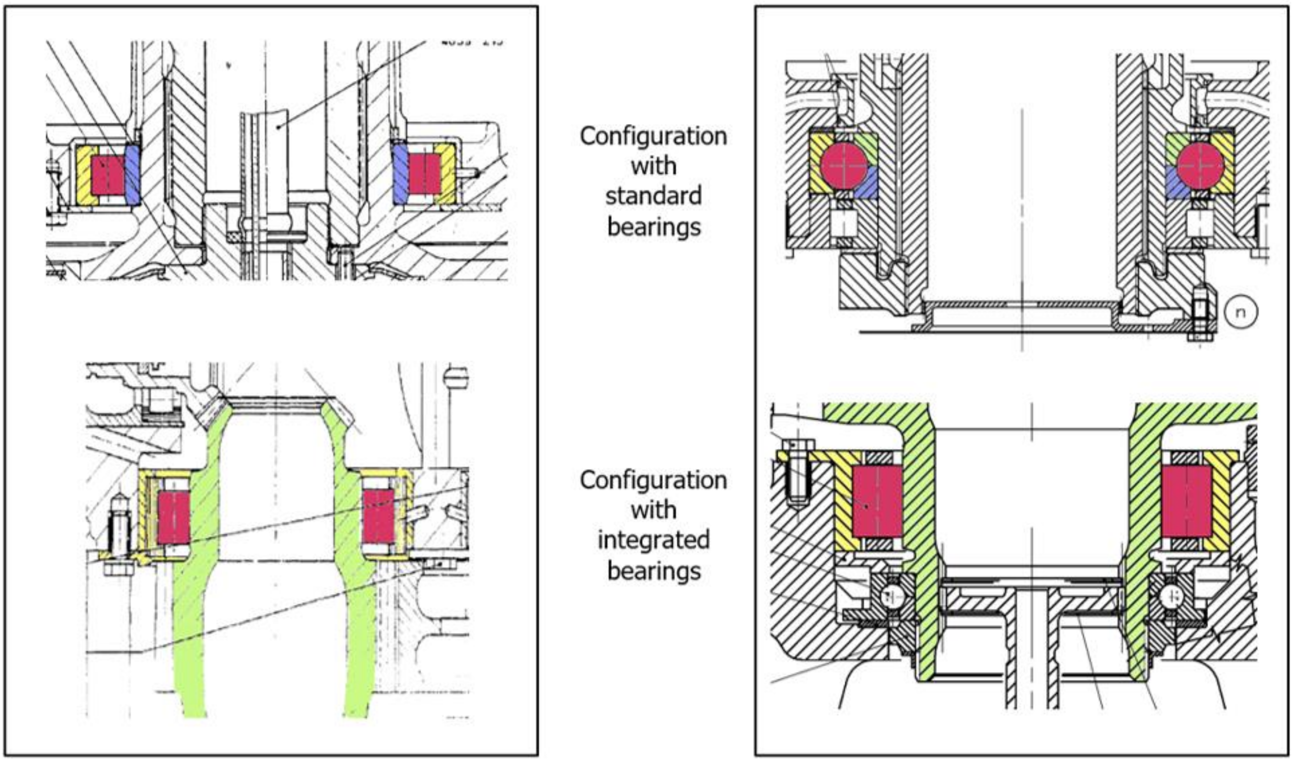

Figure 9: MGB application of integrated raceways for CS27

For one of these solutions, an investigation was started to analyse the MRO investigations (2005-today). It can be seen as a good example demonstrating the performance and reliability of integrated raceways on shafts, as no MRO event was detected with a significant bearing issue that led to a failure of the gearbox. The tables

(Table 3 and Table 4) represent the MRO values for a current MGB gearbox from 2005-2015 with regards to conventional bearings and bearings with integrated bearings. The tables give on overview of the findings, number of parts and the action, which was done. Moreover, the ratio of replaced components to components with no replacement is provided.
The tables illustrate that bearings with integrated raceways tend to be subject to higher wear in comparison to conventional bearings, but critical failures can not be determined from this table as these failures are not linked to consequences at system level. Related to this, there is also no experience on failure consequences based on service.

Input Pinion LH	Finding	Number of Parts	Action	Ratio
Bearing position 1 (Integrated raceway)	Wear	46	replaced	7.24%
	Rough-running	2	replaced	0.31%
	Damaged	11	replaced	1.73%
	Cage cracked	17	replaced	2.68%
Bearing position 2 (Conventional raceway)	Wear	8	replaced	1.26%
	Rough-running	1	replaced	0.16%
	Pitting / Spalling	30	Replaced	4.72%
Input Pinion RH	**Finding**	**Number of Parts**	**Action**	**Ratio**
Bearing position 1 (Integrated raceway)	Wear	55	replaced	8.66%
	Rough-running	1	replaced	0.16%
	Damaged	13	replaced	2.05%
	Cage cracked	8	replaced	1.26%
Bearing position 2 (Conventional raceway)	Wear	5	replaced	0.79%
	Rough-running	1	replaced	0.16%
	Pitting / Spalling	11	replaced	1.73%
	Cage wear	1	replaced	0.16%

Table 3: TBO Finding (Part I)

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Intermediate Shaft LH	Finding	Number of Parts	Action	Ratio
Bearing position 1 (Integrated raceway)	Wear	77	replaced	12.13%
	Brocken Lug	1	replaced	0.16%
	Cage Damaged	7	replaced	1.10%
	Spalling on Outer Ring	2	replaced	0.31%
	Roll Elements Damaged	18	replaced	2.83%
Bearing position 2 (Integrated raceway)	Wear	90	replaced	14.80%
	Roll Elements Damaged	17	replaced	2.68%
	Spalling on Outer Ring	14	replaced	2.20%
Bearing position 3 (Conventional raceway)	Wear	23	replaced	3.62%
	Rough-running	2	replaced	0.31%
	Spalling on Inner Ring	22	replaced	3.46%
	Roll Elements Damaged	2	replaced	0.31%
	Inner Ring Damaged	8	replaced	1.26%
Intermediate Shaft RH	Finding	Number of Parts	Action	Ratio
Bearing position 1 (Integrated raceway)	Wear	80	replaced	12.60%
	Cage Damaged	1	replaced	0.16%
	Spalling on Outer Ring	2	replaced	0.31%
Bearing position 2 (Integrated raceway)	Wear	80	replaced	12.60%
	Cage Damaged	1	replaced	0.16%
	Roll Elements Damaged	2	replaced	0.31%
	Spalling on Outer Ring	2	replaced	0.31%
Bearing position 3 (Conventional raceway)	Wear	23	replaced	3.62%
	Rough-running	3	replaced	0.48%
	Inner Ring Damaged	35	replaced	5.51%

Table 4: TBO finding (Part II)

Planetary gears with integrated raceways are a key design feature in current helicopter gearbox systems and have been used for several years. Figure 10 provides detailed illustrations of principle design solutions of integrated bearing raceways in planetary gears for an MGB.

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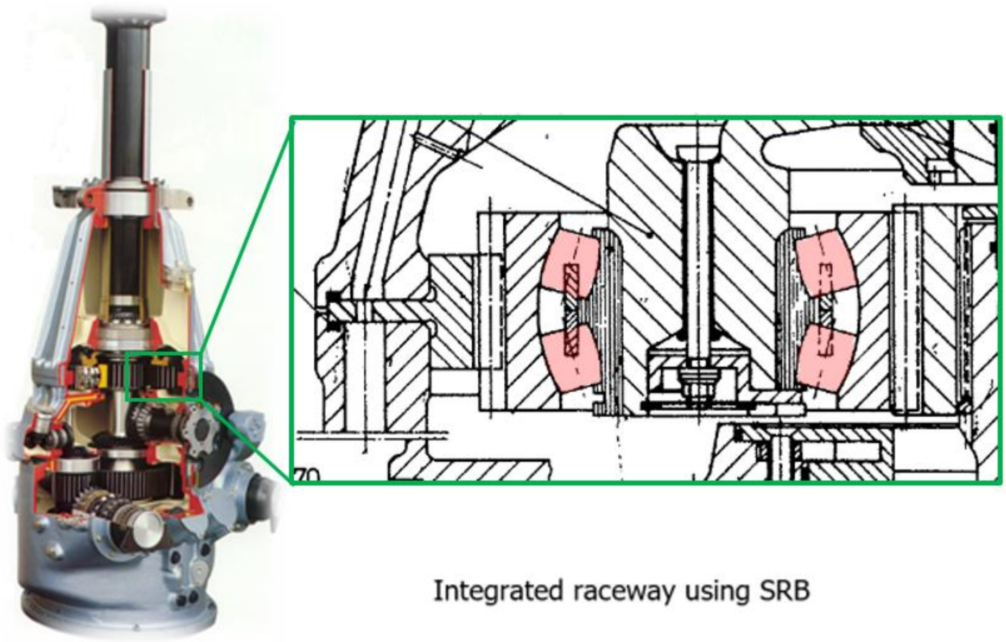

Figure 10: Integrated Raceways for a MGB planetary stage

For this MGB, a bearing failure in the integrated raceway of the planetary gear was investigated due to spalling. Pictures of the failed bearing are shown in Figure 11. The results of the investigations confirmed that no cracks or anomalies were present in the spalled area. The measured parameters, material and hardness was in accordance with the specification. The failure can be regarded as a typical spalling failure mode, without leading to crack initiation in the integrated raceway.

S/N 3066 of Roller Bearing 649550A   Outer ring (Planet gear)   Inner ring of Roller Bearing S/N 3066

Figure 11: MGB planetary gear bearing failure

Many units of this MGB have been manufactured with no reports of catastrophic damage related to the design of planetary gear stages resulting in deadly injuries. In total, three cases attributable to service faults during operation were analyzed. Findings from services included damaged roller cages and planetary axes, worn race tracks and damaged planetary gear teeth (see Figure 12). All of the associated components were replaced as a precaution during overhaul work.

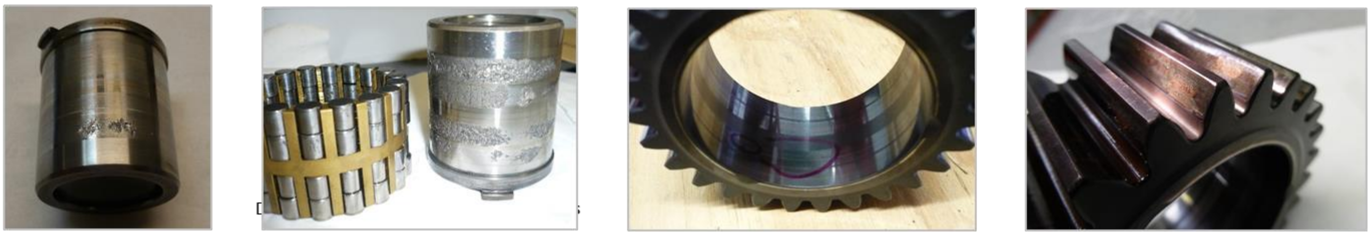

Figure 12: Preventive replacement of components during overhaul of a MGB unit

Another investigation was performed on a BO105 following and accident including an autorotation event (see Figure 13). After disassembly, an extensive amount of chips, debris and failed planetary bearings and gears were found inside, while the main planetary carrier was more or less intact (see Figure 14).

Figure 13: BO105 helicopter S-781 [51]

Figure 14: Expanded planetary gear stage [51]

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This failure can be seen as a representive example of a failed planetary gear stage, which did not subsequently led to a total destruction of the gearbox, by still maintaining the integrity of the gearbox and rotor mast of the helicopter.

1.3.2 Automotive

The aviation industry is not the only product group to use integrated bearing raceways as a design feature in products. ZF offers a wide range of products for passenger cars and commercial vehicles that feature integrated bearing raceways. The reasons behind their use are very similar to those of the aviation industry. Figure 15 provides an overview of ZF Group products that use integrated raceways.

Figure 15: ZFL experience with integrated raceways in passenger cars and commercial vehicles

ZF has extensive experience in integrated bearing raceways for the automotive sector as the manufacturer of over 2,500,000 integrated bearing assemblies per year. Planetary gears with integrated races are primarily used in automatic gearboxes due to their compact design and high-power density. The size of planetary gears and bearings is similar to the ones used in the aviation industry, but the design is typically flatter and more than one planetary gear stage is used in an individual gearbox system. In general, the roller elements are wider with a smaller diameter and the gears are helical spur gears. Two examples of an automotive gearbox with integrated bearing raceways can be found in Figure 16.

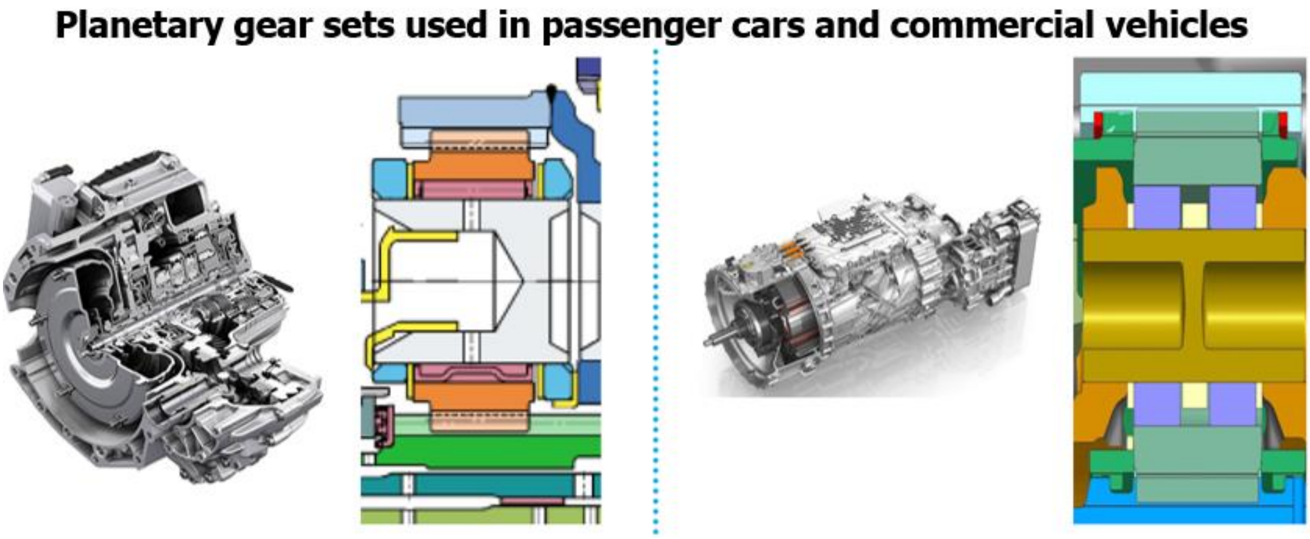

Figure 16: Integrated bearing raceways for automotive gearboxes

Several studies have been conducted in recent years (2000-2019), like component testing on test rigs as shown in Figure 17, to create a database of fatigue failures attributable to integrated raceways. Testing and available field data were used and, as a result, over 250 reports were analyzed, containing material checks, failure mode evaluations, crack/spalling evaluations and root cause analyses.

Illustrations of the damages found and an example of a test bench used in the studies can be found in Figure 17 - Figure 21.

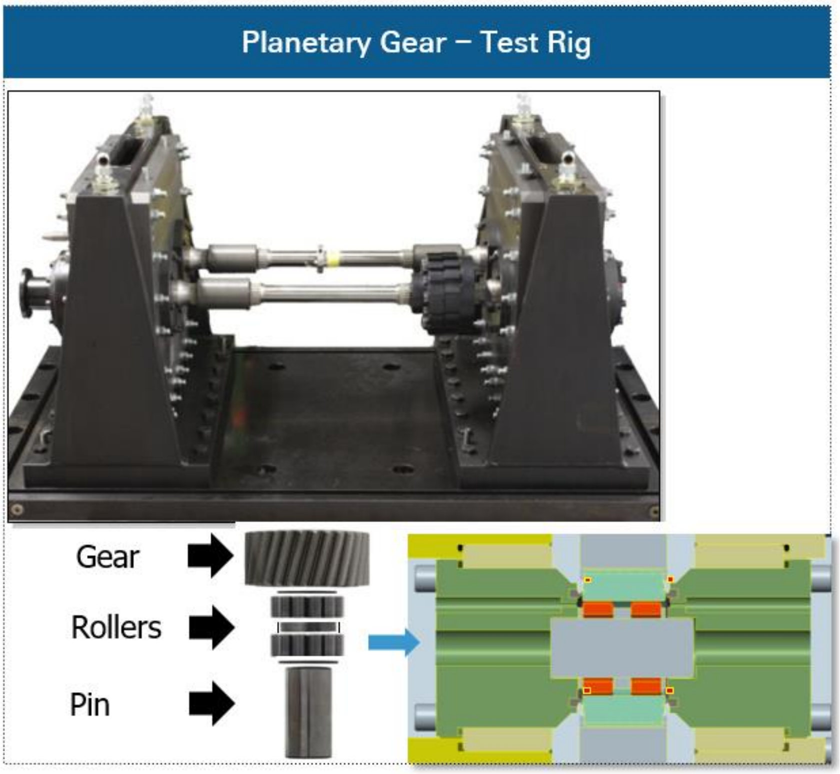

Figure 17: Planetary test rig for integrated bearings

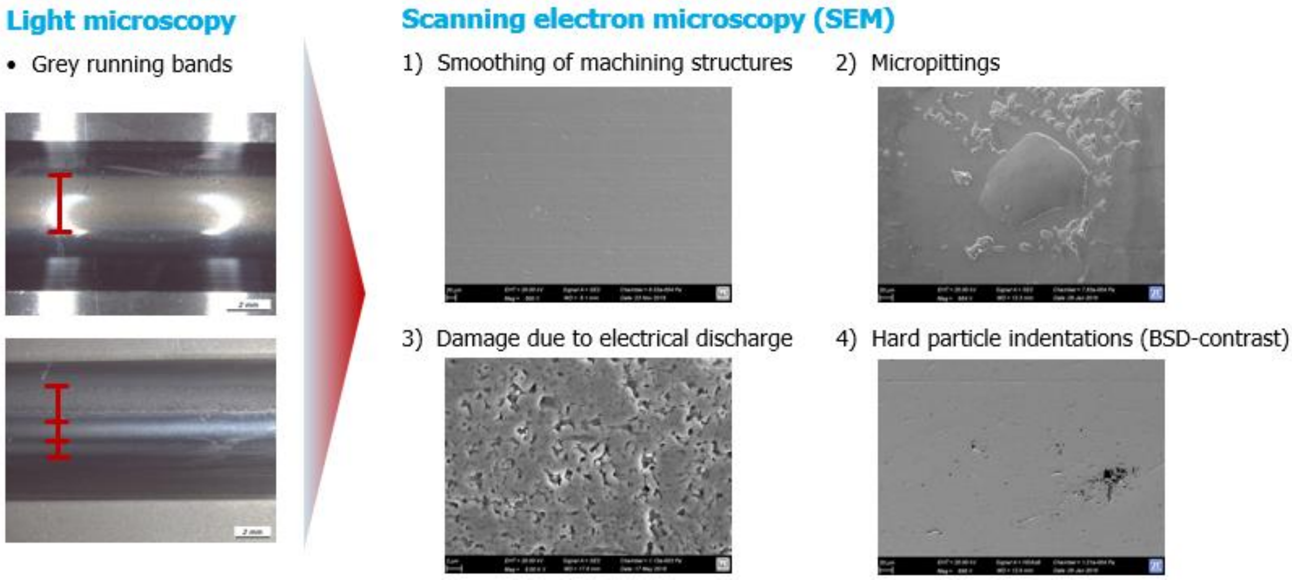

Figure 18: Investigation results for damage of integrated bearing races – part I

Figure 19: Investigation results for damage of integrated bearing races – part II

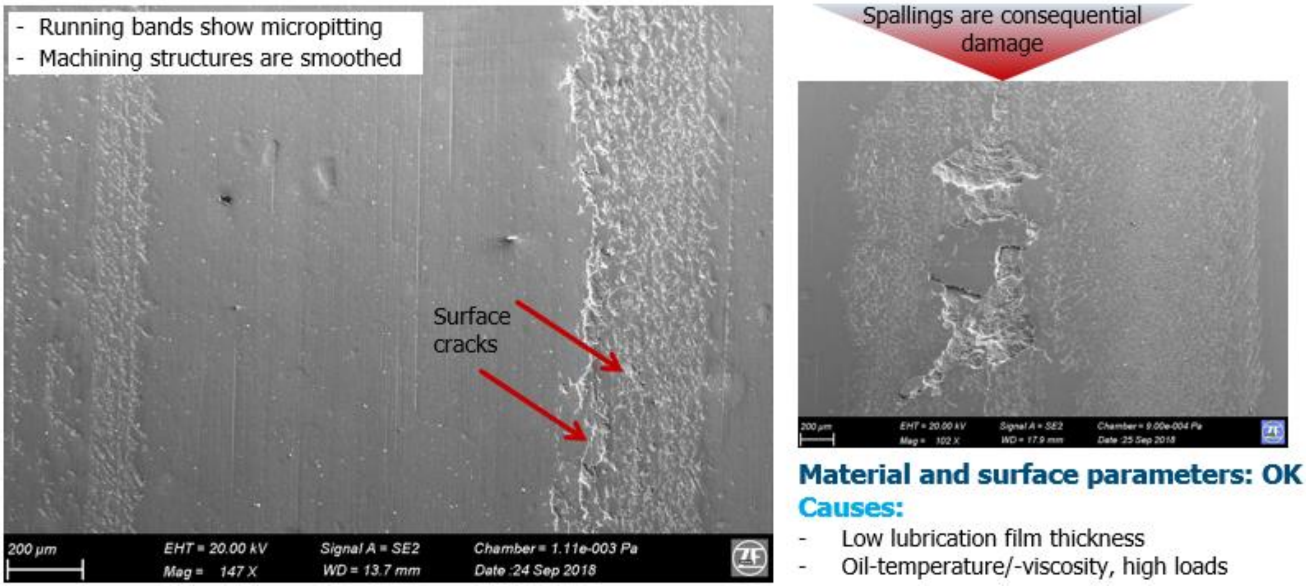

Figure 20: Investigation results for damage of integrated bearing races – part III

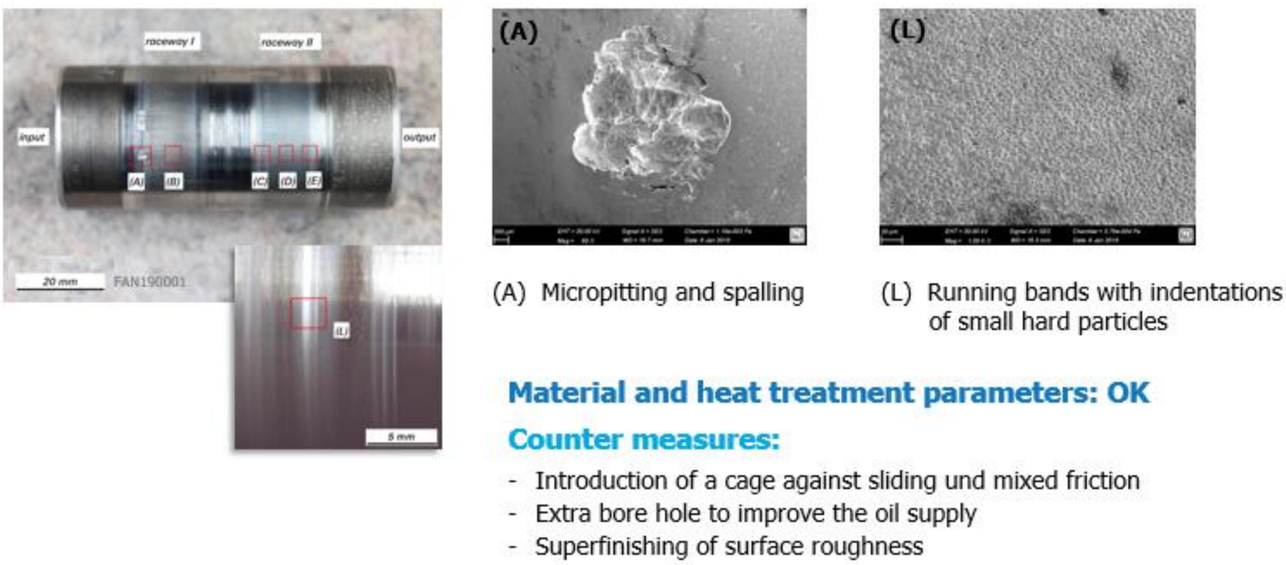

Figure 21: Investigation results for damage of integrated bearing races – part IV

Damage to shafts with integrated races was primarily detected in trucks and off-road applications as the result of spalling or pitting during use. Similar findings were made for planetary gear stages, mainly in passenger cars, trucks and off-road applications. Over 99% of the damages were attributable to tooth failures (e.g., pitting, breakage). During the investigation, only one case with breakage of a planetary gear rim was detected. Further inspection demonstrated that the breakage occurred due to an overload of the component.

1.3.3 Wind Industry

In addition to the automotive and aviation industry, the wind industry uses integrated bearing raceways, mainly for planetary gears. Figure 22 provide two examples of current wind gearbox systems with integrated bearing raceways manufactured by ZF.

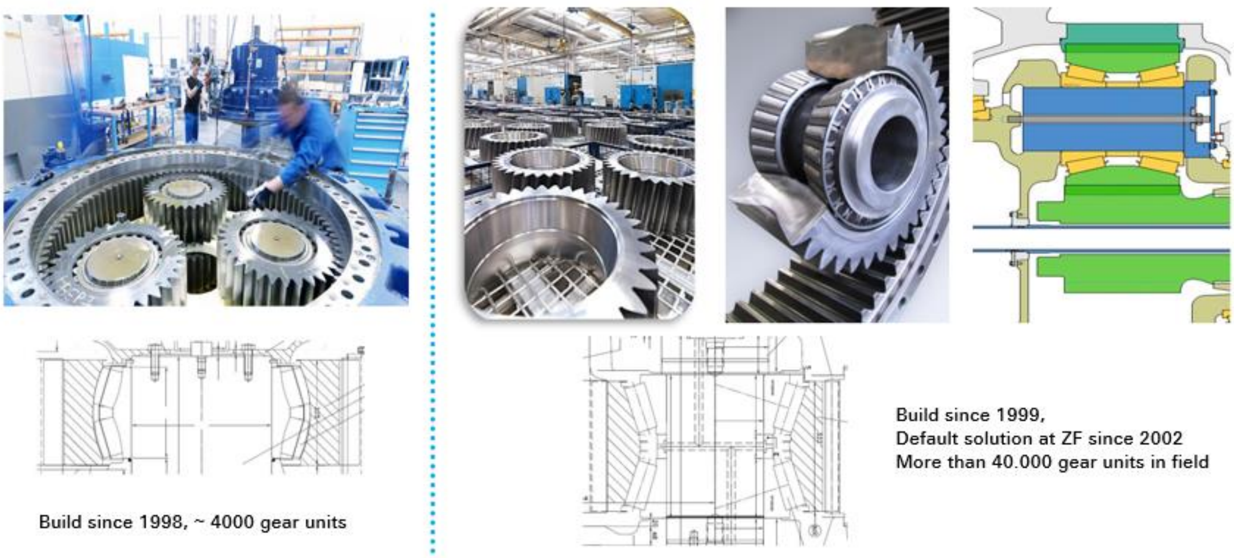

Figure 22: Integrated bearing raceways for wind industry gearboxes

ZF has gained over 40 years of experience in the wind industry in terms of design and service. Various design solutions are available for integrated bearing raceways for SRB, CRB and TRB bearings and dedicated design guidelines have been developed to ensure a reliable design, including defined rim thickness, bearing design (e.g., permissible pressure), special design features (e.g., lubrication holes) and the material selection and heat treatment. Similar to the other fields of application, basic research has been conducted to evaluate common bearing failures. As a result of this research, no rim breakage has been reported in recent years. Instead, white etching cracks (WEC) proved to be the most common failure mode, next to spalling/pittings and small cracks (see Figure 23 and Figure 24). According to current knowledge, WEC occurs due to low cycle fatigue at moderate pressures, leading to the formation of a network of cracks towards the surface (visible), which cannot be predicted by simulation (see also Figure 25). Moreover, this is not directly linked to the use of integrated raceways and is independent of the material used (through-hardened, case-hardened, bainitic, martensitic). Potential contributors to WECs are different based on the application and usage. Some main contributing factors can be discussed based on investigations done by ZF:

- Oil type, properties and contamination
- Slippage or friction
- Overload
- Electrical current

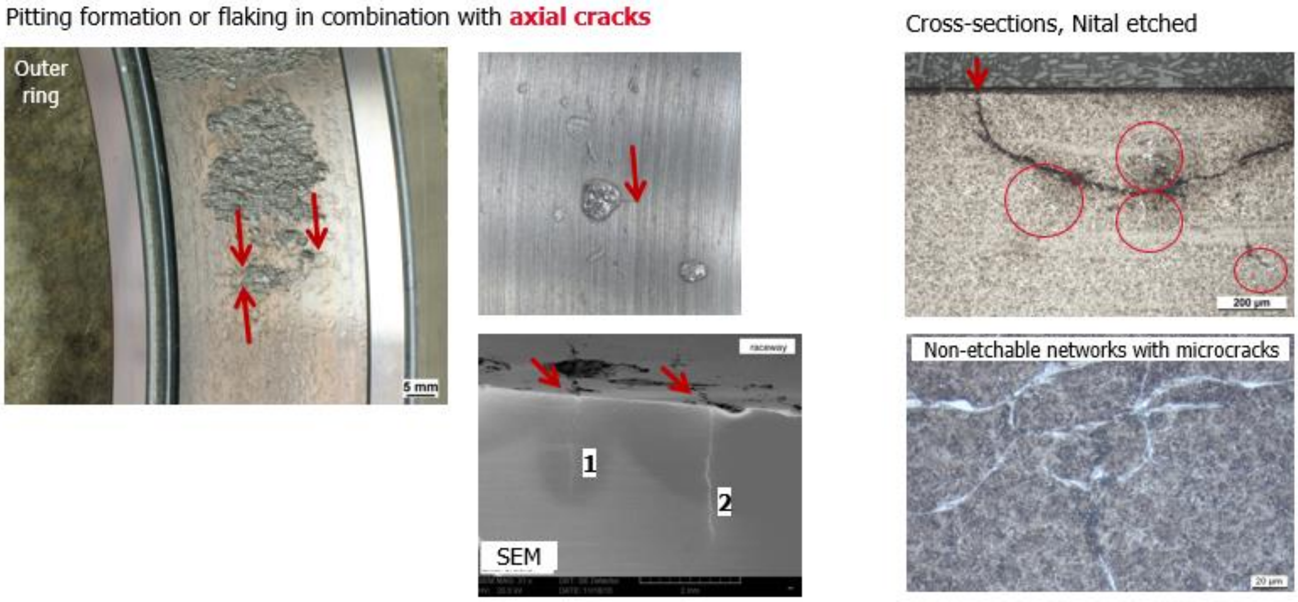

Figure 23: Detected damages of integrated bearings races – part I

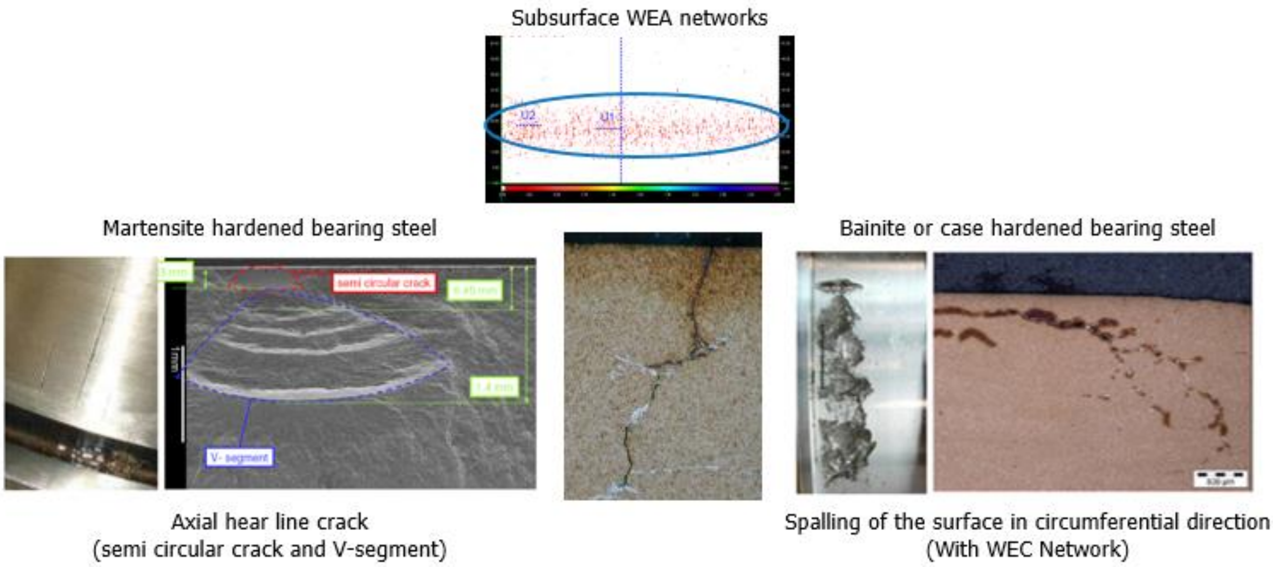

Figure 24: Detected damages of integrated bearings races – part II

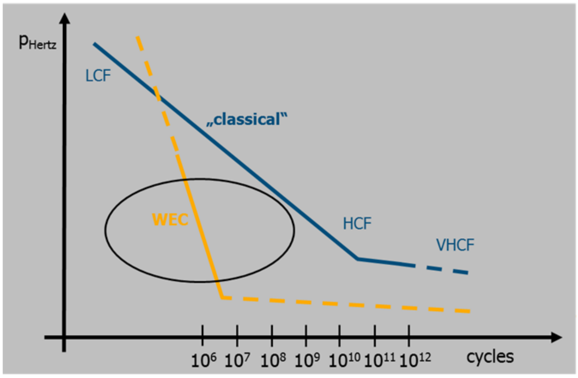

Figure 25: WEC vs. classical fatigue

In principle, the main difference between the planetary gear systems used in wind power and the ones used in the aviation and automotive industry is the vast size of the wind turbine planetary stage. The roller elements in wind turbines are typically designed to be longer than they are wide, and the number of planets in the planetary gear stage usually does not exceed three.

1.3.4 Summary

The previous sections looked at the breadth of experience gained by ZF in the aviation, wind power and industries with a particular focus on integrated bearing raceways. Whilst there are many differences between these industries, similarities can be found in their respective use of integrated bearing raceways. All of the above industries use integrated bearings in planetary gears stages, and the automotive and aviation industries also use them for integration on shafts. The bearing types and materials are largely similar with small deviations with regard to the specific application. Particularly in the aviation industry, manufacturing is heavily regulated and restricted to a few permitted processes in order to ensure high material cleanliness. While aerospace bearings typically feature rolling elements with similar length and radius, the wind power and automotive industries often use rolling elements with a length > radius. As such, there is a visual difference in the size of the components. Automotive bearings are similar to aerospace bearings, yet the bearings used in the wind power industry have larger dimensions and gears can be rigid instead of elastic.
ZF primarily partners with SKF, FAG and Timken for wind power bearing solutions. Failure modes that often arose in the past, were mainly attributable to RCF, conventional wear and WEC (not linked to integrated raceways), particularly for the wind industry. Over the entire course of investigations, the only critical failure detected during in-house investigations was the BO105 accident (see Figure 13).

2. Critical failure scenario / thesis

2.1 Rolling contact fatigue (RCF) of bearings with integrated races

The failure modes for bearings with integrated races are generally comparable to those exhibited by non-integrated bearing races, with the exception of crack initiation/propagation initiated by rolling contact fatigue. This failure mode is of more importance to integrated bearing races in comparison to non-integrated bearing races due to the missing inner/outer ring and direct contact with the surrounding components (e.g., planetary gear). Rolling contact fatigue (RCF) occurs when a surface is exposed to repeated rolling and sliding under high loads [7]. According to Ahmed [8], RCF can be defined as the mechanism of crack propagation caused by the near-surface alternating stress field within the rolling-contact bodies, which eventually leads to material removal. The formation of small craters in the surface (spalls) is referred to as spalling [7]. Additional types of damage related to RCF include subcase fatigue in surface hardened components (case crushing), surface originated pitting, peeling/micropitting and section fracture [9][10]. RCF can be found, for example, in ball and roller bearings, gears, cams, tappets and rails [8][11]. Table 5 gives an overview of some of the main differences (among others) according to classical and rolling contact fatigue:

	Classical fatigue	RCF in hard steels (e.g. rolling bearings)	RCF in soft steels (e.g. railway track)
Stress regime	May be uniaxial, proportional, usually tensile	Always multiaxial, nonproportional, random history, compressive, 'elastic', low friction	Always multiaxial, nonproportional, random history, compressive, 'plastic', high µ
Origin	Surface, associated with PSBs, notches, subsurface inclusions and pores	Surface asperities, subsurface inclusions	Surface, subsurface cementite – ferrite boundaries
Stages	Cyclic strain localization, crack formation, short crack growth, long crack growth, fracture	Local plasticity, inclined crack formation, inclined (short) crack growth, pit formation, fracture	Local plasticity, inclined crack formation, inclined crack growth, long crack growth, fracture
Factors affecting fatigue life	Stress–strain history, ratchetting, environment (corrosion, hydrogen, water)	Slide– roll ratio, hoop stress, roughness, environment (lubricant, hydrogen, debris)	Ratchetting, wear, axial stress, environment (corrosion, debris water)

PSB = persistent slipbands

Table 5: *Factors related to classical and rolling contact fatigue* ***[9]***

In comparison to classical fatigue, the origin of the crack is very similar for RCF, yet the stress regime, the stages of fatigue and the factors affecting the fatigue life vary. As a result, an RCF evaluation cannot be performed in the same manner and based on the knowledge obtained from classical fatigue. Sliding and rolling of the involved components, in addition to lubrication and debris management, which affects the fatigue life, lead to a more complex assessment of crack propagation, which has not been researched at length to date.

2.2 Crack through planetary gear with integrated raceways

The critical failure scenario for integrated bearing raceways was presented in chapter 2.1 and is defined as rolling contact fatigue. As described, planetary gears with integrated raceways have a high potential to fail under rolling contact fatigue. The risk of catastrophic failures leading to death or injury is present in both the aviation and automotive industries. Accidents that occur in the wind power industry are rarely reported and usually take place without human involvement. The following sections will look at some examples of reported accidents.

2.2.1 Aviation

The following failure scenario is based on a helicopter accident that occurred in 2016 near Turøy in Norway, with an Airbus EC-225-LP helicopter, registration number LN-OJF, operated by CHC Helikopter Serivce AS. The accident was triggered by a fatigue crack in a planetary gear with integrated raceways, as determined by final investigations into the accident (see Figure 26). The structural degradation of the planet gear, which is a critical part of the gearbox, was attributed to undetected subsurface cracks. The fatigue crack was initiated by an indentation from particle overrolling in the upper outer race of the bearing, before growing in the direction of the tooth root, resulting in a gear ring failure (see Figure 27). This failure mode was not previously assessed, despite the fact that a combination of several contributing factors was present (material properties, surface treatment, design, operational loading environment and debris).
This scenario can be regarded as a lesson learned, not only for the EC-225-LP helicopter, but also for other helicopter types. Following the accident, the aspects of design, safety assessment, fatigue evaluation, condition monitoring, certification and continued airworthiness were redefined [15].

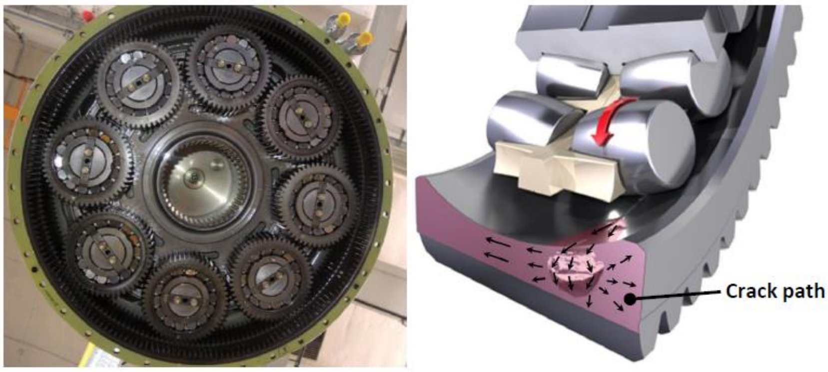

Figure 26: Failure Szenario – Crack integrated raceway of planetary gear [6]

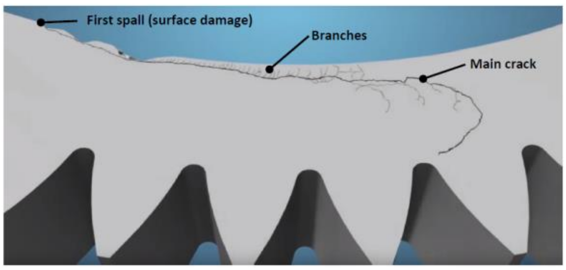

Figure 27: Crack stages on planetary gear [6]

In 2009, a similar accident was reported in an EC-225 helicopter, registration number G-REDL, whereby RCF at the planetary gear stage initiated a fatal outer gear ring failure and loss of the helicopter with several casualties. A photograph of the fractured outer ring and gearbox housing that led to the catastrophic failure is shown in Figure 28.

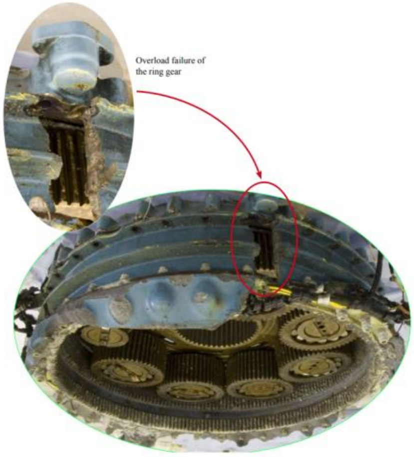

Figure 28: Reassembly of the fractured gearbox part of helicopter G-REDL[19]

2.2.2 Automotive industry

A planetary gear in an off-road truck broke during transportation and resulted in an unexpected blockage of the rear axle. Investigations have found that a crack through the rim of the planetary gear, initiated by non-metallic inclusions, was the reason behind this failure. Luckily, no one was hurt in this accident. The fractured gear ring and a schematic drawing of the planetary gear stage can be found in Figure 29.

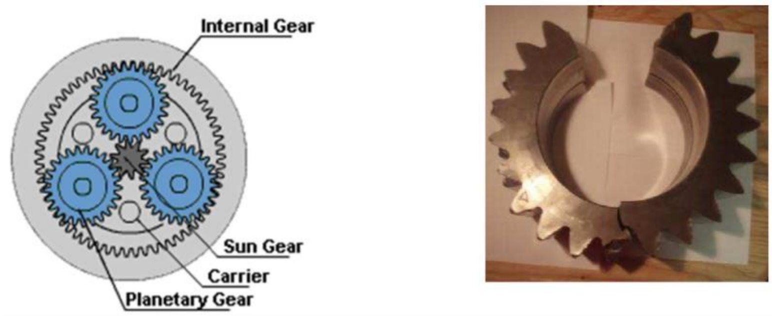

Figure 29: Fracture of planetary gear of an off highway truck [23]

2.2.3 Wind industry

No accidents in the wind power industry have been reported or disclosed to ZF attributable to failure caused by a cracked rim in a planetary gear stage.

3. Key design parameter

Chapter 3 gives an overview of the key design parameters for bearings taking into account parameters for bearings with non-integrated and integrated races. The fishbone diagram (Figure 5) in chapter 1 was used as a baseline for the selection of potential parameters. In an additional step of this project, these parameters can be used to define possible measures for developing design solutions that may help to prevent such accidents in the future.

3.1 General bearing parameters

This chapter gives a general overview about the main design parameters for bearings, by addressing first those parameters that are common between conventional bearings and bearings with integrated raceways. Specific parameters and additional considerations on common parameters for bearings with integrated raceways are listed in chapter 3.2.

3.1.1 Bearing type

In general, similar design parameters apply to non-integrated bearings and bearings with integrated raceways for shafts and gears. The process for selecting the right bearing parameters and the right design can be applied to SRB (spherical roller bearings), CRB (cylindrical roller bearings), TRB (tapered roller bearings) and also ball bearings, in terms of integrated raceways. In other words, regardless of the type of integration or non-integration in the bearing, it always features an inner and outer ring. Therefore, the main key parameters are considered applicable to integrated or non-integrated versions. The following paragraphs address the details for the 2 configurations.
The choice of bearing type is normally restricted, depending on the intended function of the bearing. Therefore, it is not within the scope of this report focus on and assess the impact of the bearing type further.

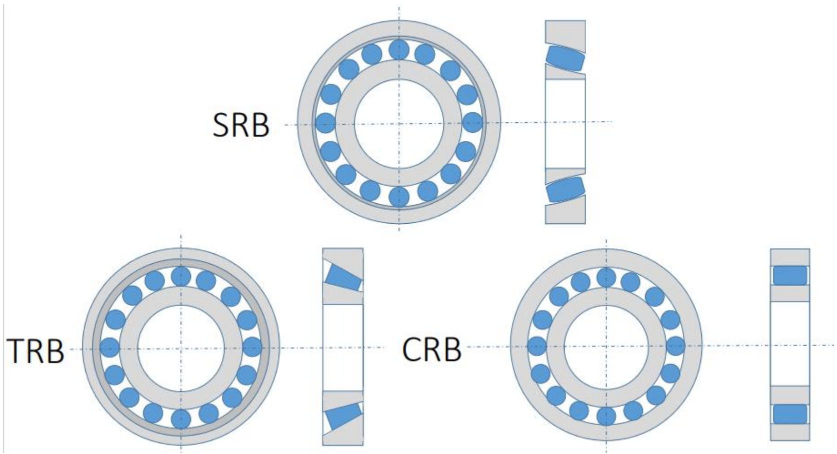

Figure 30: Example of SRB, CRB and TRB bearings [53]

3.1.2 Internal Stresses

3.1.2.1 Tightening – Hoop stress

In order to prevent the bearing inner/outer ring from rotating around the shaft/housing (as a function of the rotating ring), and hence prevent ring creep of the bearing interface surfaces, the bearing inner/outer ring is usually press-fitted. The amount of diametral interference, and therefore the required pressure between the ring and the shaft/housing diameter, primarily depend on the load and secondarily on the rotational speed. The outer ring will be subjected to compressive hoop stress whereas the inner ring will be exposed to detrimental tensile stress. Each of the stresses attributable to press-fitting or ring rotation are superimposed on the subsurface stress field caused by contact surface stresses. It is key that the correct tightening level is defined to avoid the ring–shaft/housing motion and prevent ring weakening. Moreover, the calculated theoretical interference level should be feasible in practice: mounting/dismounting (loads and heating/cooling temperatures). Excessive hoop stress on the inner ring can lead to structural cracks. Furthermore, small motions between rings and the shaft/housing can cause fretting wear as potential fatigue crack initiation. These weak areas can only be detected following bearing dismounting.

As is perhaps clear, the ring deformations induced by the tightening need to be included in the bearing clearance calculations. The geometry, material and potential deformation of the shaft/housing must be included in the tightening calculations. If the shaft or gears feature integrated raceways, there will be no issues with ring tightening. Nevertheless, ring internal stresses may exist in relation to other phenomena (shaft bending, loading on gears).

3.1.2.2 Roller raceway full contact & truncation

The contact area between the roller and the raceway (contact ellipse) is a key element to the bearing design. Particular attention is paid to the risk of truncation/roller-raceway full contact (see Figure 31). As such, the roller shape is designed for contact with the raceway and predefined according to length and area. An increased load and/or misalignment can lead to a 'full/truncated' contact situation followed by contact between the side edge of the roller (chamfer/corner) and the raceway. This edge contact is highly detrimental to both the roller and the raceway since it will induce high contact pressure peaks, which can lead to spalling and cracking. In order to avoid edge contacts and truncation, the roller profile drops can be increased, or very low osculation can be implemented (the raceway radius is much larger than the roller radius). Nevertheless, such changes will directly increase the contact pressure, potentially reducing the bearing life and increasing the risk of subsurface spalling.

Accordingly, a critical design choice needs to be made: should edge contact/truncation situations be avoided for every load case (including extreme/limit cases) to the potential detriment of nominal operating conditions with an increase in the contact pressure? Or is edge contact/truncation acceptable in certain/low ratio load cases? This design compromise will impact the roller/raceway profile & radius.

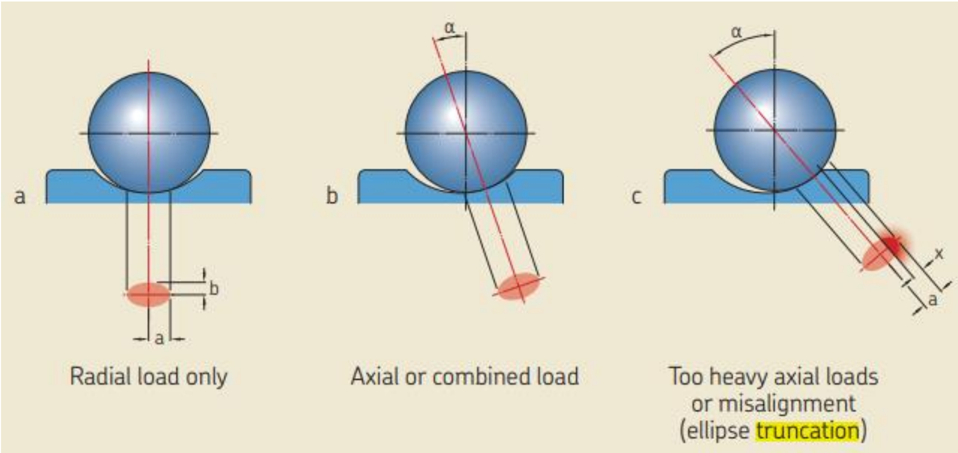

Figure 31: Example of truncation at ball bearing [4]

3.1.2.3 Contact stress

Contact stress is systematically calculated for the various operating conditions (load cases, misalignment, rotational speed, temperature, etc.). It depends on several parameters (geometrical, material, operating conditions). The calculation is made using Hertzian elastic contact formula. The main parameter is the normal contact pressure value (maximum shear stress and depth of maximum shear stress are also extracted). The normal contact pressure is calculated for both rings and the calculated values are compared with defined criteria to assess the reliability of the bearing design. The acceptance criteria will be based on the bearing type (CRB, SRB, TRB), the materials used, the duration of the load cases and the target bearing life. Moreover, even if the direct risk of excessive contact stress is subsurface failure (deep spalling), other phenomena (such as surface spalling from dents, surface wear etc.) may be influenced by the contact stress level. As a result, the lubrication properties (film thickness, oil cleanliness) are also taken in account in the contact stress permissible limit. It is important to note that, even if the roller/raceway design can be modified to adjust the contact pressure level, it is not systematically possible to reach a risk-free value. Indeed, a design compromise with the risk of edge contact/truncation, friction and power loss, misalignment and bearing size has to be found.
Generally, the bearing raceway is mainly loaded by three different components:

- Ovalization
- Hertzian Pressure
- Residual stresses

It is well established that cyclic Hertzian stress due to the contact pressure (enhanced by oil pressurization) is a driver of subsurface crack propagation. If particles are not released, they still transmit Hertzian stresses to the crack front, favoring its propagation. If they are released, e.g. by spalling close to the main crack front, it becomes unloaded and can be arrested. Thus, one can see a competition between the spalling rate and the subsurface crack growth rate [5]. Therfore, a minimization of the hertzian stresses could lead into a favourable condition for the reliability of a bearing.

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3.1.2.4 Misalignment

Misalignment (see Figure 32) between the inner and outer ring is a key parameter that directly impacts the contact between the roller and the raceways with additional effects on the contact pressure and risk of truncation. The level of misalignment can be:

- calculated based on the system geometry (shaft/housing/gear), temperatures and loads (flexible modelling)
- provided as a direct input for the bearing design

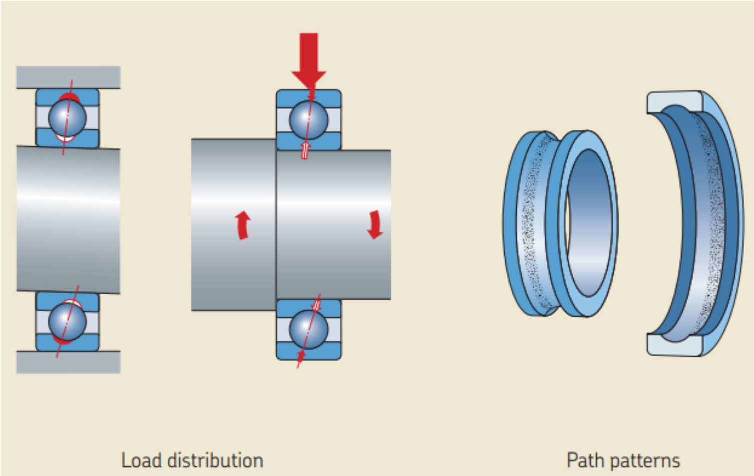

Figure 32: Example of inner ring misalignment [4]

3.1.3 Slippage & PV (Smearing Risk)

In the calculation phase, the risk linked to the slippage of the rolling elements can be estimated in combination with local contact stress. A significant sliding ratio can lead to early surface wear (see Figure 33). In addition, when combined with contact pressure, the product of the contact pressure with the sliding velocity (PV factor) is identified as a main contributor to the smearing risk (micro-welding of surface). It can be difficult to assess the sliding of the roller in a CRB since there are no direct geometrical sliding contributors. The sliding rate is consequently determined by dynamic phenomena such as low loads or parasitic motions (skewing) of the roller when leaving the load area. Only advanced dynamic modelling can be used to evaluate such phenomena. Nevertheless, general design rules can be applied to control the minimum bearing load. For SRB, the roller spin can be geometrically defined and used to calculate a PV factor along the contact ellipse to determine the risk of smearing and surface degradation.

If there is a high risk of smearing, several strategies can be used to bring the risk level down, such as modifications to the geometry (radius, roller profile), decreased contact pressure (if possible) or surface reinforcement (heat-treatment, coating).

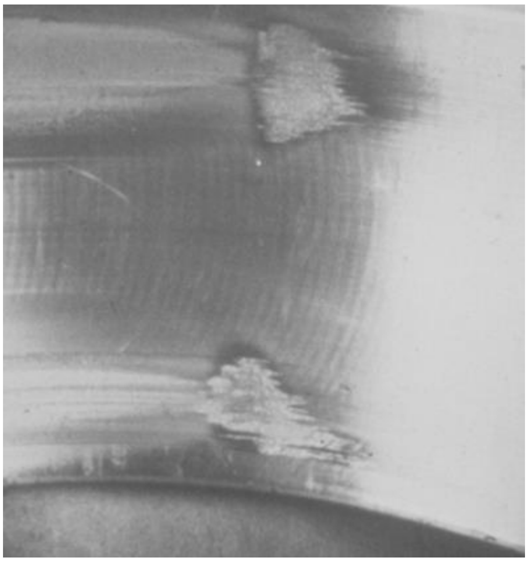

Figure 33: example of damage related to smearing at spherical roller bearing outer ring at entrance of load zone [4]

3.1.4 Lubrication

Lubrication is a key factor in influencing bearing behavior. Aircraft gearboxes bearings are generally oil lubricated. It is important to note that the choice of oil is not specific to the bearing but rather entirely dependent on the system, as the oil will be also used to lubricate other transmission mechanisms (such as gears). Moreover, the cleanliness of the oil is also determined by the system (filtration) and the application (environment, maintenance, etc.). Based on these considerations, three key design parameters can be defined.

3.1.4.1 Lambda ratio lubrication

The lambda ratio is calculated as the ratio between the minimum film thickness and the composite roughness of rolling elements and raceways. The lambda ratio gives an indication of the quality of the lubrication conditions relative to surface separation (probability of contact between surface micro-asperities).
Acceptable Lambda ratio values $\lambda \geq 1.2$ guarantee full film lubrication, whereas $\lambda < 1$ illustrate boundary layer lubrication with a high risk of surface wear. Moreover, under low lambda conditions, surface degradation (as surface dents) may evolve more easily into more severe ring failure (deep spalling). Several complex options have been proposed to improve the lubrication conditions:

- increase the oil viscosity, but the choice of oil is often based on the system.
- increase the rotational speed or decrease the temperature by adjusting the oil cooling performance or oil flow
- decrease the ring surface roughness, but the process limits are often reached (in consideration of commercially valid solutions)

3.1.4.2 Oil flow

The oil flow is calculated to determine the release of heat generated by the bearing. The temperature difference between the bearing temperature and the oil temperature is a key input value. The thermal oil properties are also included in the calculation. This calculation does not directly impact the bearing design but is considered an important input to the system.

3.1.4.3 Oil cleanliness / pollution

The cleanliness of the oil is critical to the reliability of the bearing raceway, both for integrated and non-integrated bearing raceways. Indentations from particle overrolling are the result of contaminants inside the oil reaching the raceway of the bearing. These indentations could further lead to a concentration of stress and crack initiation.

3.1.5 Bearing Life

It is often interesting to calculate an estimation of the bearing life for a new design. Indeed, the calculated life time can be directly compared with the target bearing life for the application but also be compared with previous similar bearing design life results. The results can vary depending on the formulation and the boundary conditions used. Therefore, the bearing life can not be seen as a typical key design parameter but can still be used as a tool to support the design process. Nevertheless, for the sake of completeness, some main points regarding the bearing life calculation will be adressed in the following in order, to emphasize that bearing life should not be viewed as a key design parameter.
Based on ISO 281:2007, the Basic Rating life is the simpliest bearing life estimation. It gives a relatively good approximation based on the bearing capacity and the loads applied. However, this result is unable to consider any deviation from the optimum bearing operating conditions (e.g. poor lubrication, contamination, etc.).
The ISO Rating Life uses the $aISO$ factor to calculate a better life estimation than the Basic Rating Life by integrated in a basic way the effects of contamination, lubrication and fatigue (not for Aeroengine bearings).
The Adjusted Fatigue Life (AFL) is the current reference life result of the aeroengine bearing. The AFL will calculate independently the life of each ring and the ring fatigue life calculation uses the actual load on each rolling element / raceway contact. It will also integrate material and lubrication factor.
The SKF Advanced Fatigue Calculation (AFC) life method takes the full integration of rolling element contact stress across the roller length into consideration and evaluates the total number of stress cycles until life in the entire loaded volume is consumed. The SKF AFC life also incorporates the condition of the lubricant, taking into account operating film thickness and contamination for each contact individually.
Recently in 2015, SKF has presented a state-of-the art bearing life calculation method called SKF Generalized Bearing Life Model (GBLM, see also Figure 34), an innovative new bearing rating life model that is designed to help engineers calculate bearing rating life in a more realistic manner. Indeed, the SKF GBLM will retain the standardized probabilistic approach used up to now in rolling bearing life ratings based on a two parameter Weibull distribution but it considers now two regions: one for the subsurface and another for the surface. This integration surface life (and not only lubrication or contamination parameters) is key to fine-tune the bearing life estimation when most of bearings fail due to surface initiated fatigue.
So far, this model is not used to calculate the life of aerospace transmission bearings and mainly applied to hybrid bearings. The calculation tool and the model calibration for Aerospace materials are still under development.

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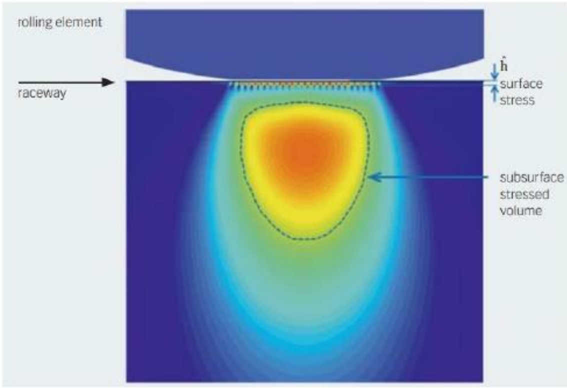

Figure 34: Separation of surface and subsurface as proposed by GBLM [3]

At the end, every bearing life calculation results should be considered carefully since it is based on several assumptions from the model calibration to the final bearing mounting and running conditions. Usually, experiences from fatigue tests (life testing) using an approximation of the conventional (operational) conditions can be used as a baseline for the bearing life model. Correction factors are also used to scale the results of the fatigue tests to the conventional conditions (e.g. the influence of corrosion or temperature is sometimes not tested in parallel during fatigue). This is because there are still uncertainties left in the approximation of the bearing life model, that could make a difference to the real application condition range (see Figure 35).

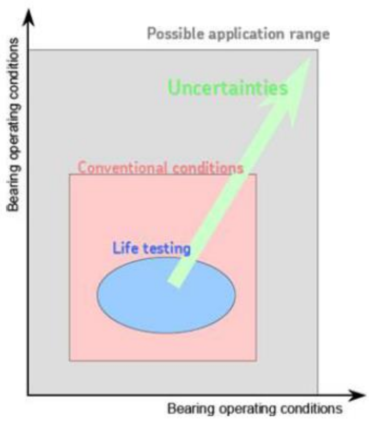

Figure 35: Illustration of bearing life model uncertainties [3]

3.1.6 Geometry

SKF Aerospace bearings are manufactured in compliance with ISO and ABEC tolerance standards, with a minimum tolerance of ABEC 5. The ABEC scale is an industry accepted standard for the tolerances of ball and roller bearings. The ABEC scale is designed to provide bearing manufacturers dimensional specifications that meet the standards of precision bearings in a specified class. These tolerance classes apply to all geometrical tolerances, dimensions and shapes (circular, concentric, square, parallel, etc.). Furthermore, specific design parameters can be defined in relation to the geometry of bearings.

3.1.6.1 Internal radial clearance

The radial internal clearance (see Figure 36) refers to the potential motion between inner and outer rings measured in the radial direction. This clearance may emerge as a result of the various operating conditions depending on the temperatures and ring fitting/deformation. The final bearing geometry (drawing geometry) is calculated to meet a target radial clearance range for the various load/speed/temperature conditions (including the impact of rings tightening).

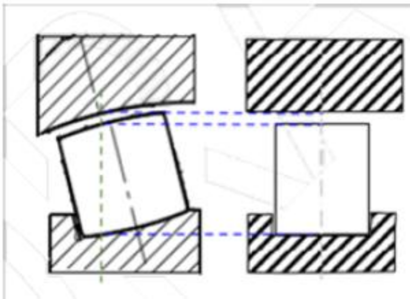

Figure 36: Measurement of internal radial clearance [3]

If the radial clearance is too small, the main risk is that clearance will be lost (negative clearance) for certain operating conditions, leading to additional loads on the contact roller/raceways and an increase in contact pressure. This may lead to rapid failures (spalling, seizure). The radial clearance has a direct impact on the load area angle/width. Moreover, in the case of transmission bearings, in particular planetary/satellite bearings, the load level is typically quite high, which can impact the critical load area size and lead to overloaded rolling elements. Nevertheless, it is important to note that in the case of an integrated gear on the outer ring, the radial clearance is similar to the potential outer ring deformation values. As such, the radial clearance must be kept higher than the ring deformation (under gear loads, for example) in order to avoid displacement of the contact zone position and loading the ring perpendicular to the load application direction.
If the clearance is too loose, the rollers will have a high degree of movement and guidance of the inner vs outer rings cannot be ensured.

3.1.6.2 Axial clearance

The axial clearance (see Figure 37) can be decomposed in two different parameters:

- the gap between the roller length and the ring shoulders spacing shoulders (when the rollers are guided by two shoulders)
- the potential motion of inner vs outer ring measured in the axial direction

In terms of the clearance to be maintained between the roller and the ring shoulders, the value is based on geometrical analysis that aims to avoid the contact area between the roller and the shoulders (in the event of a contact, the target is to achieve a full face-to-face contact instead of a smaller line contact). This clearance is controlled by the ring after the assembly of the rollers.
With regard to radial clearance, a target value is defined to limit the motion (minimum value) while avoiding any risk of seizure. The final bearing geometry (drawing geometry) is calculated to meet a target axial clearance range for the different load/speed/temperature conditions.

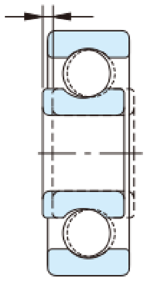

Figure 37: Example axial internal clearance [54]

3.1.6.3 Cage pocket clearance

The cage pocket clearance (see Figure 38) is the gap between the roller dimension and the cage pocket (length and diameter). This clearance will directly impact the potential motion of the rollers and their guidance.
As a result, it is controlled to ensure a minimum value is maintained to avoid any structural effects on the cage (fatigue load).

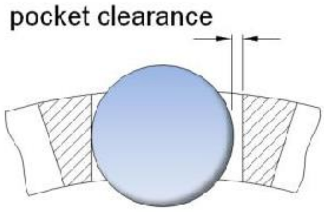

Figure 38: Example of cage pocket clearance [55]

3.1.6.4 Osculation

The osculation (see Figure 39) is the ratio between the raceway and roller curvature (for SRB, not used for CRB). The osculation is therefore dependent on the roller and raceway profiles. The determination of the osculation ratio has a significant impact on the risk of full-contact/edge contact on the roller (refer to the paragraph 3.1.2.2 – Roller-Raceway Full Contact & Truncation). If the roller's curvature is similar to the raceway's, then minor variations in load or alignment can lead to edge contact, resulting in a risk of over-pressure and spalling. If the osculation is too loose, the rollers can easily be exposed to parasitic motions (skewing) and the load capacity will be significantly reduced (higher contact pressure decreases the bearing life and increases the risk of subsurface and surface failures).

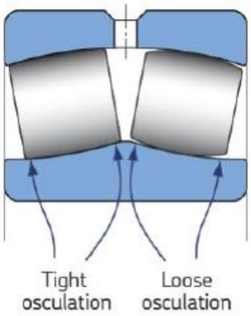

Figure 39: Osculation scheme explanation [3]

3.1.6.5 Inner or outer ring diameter

The inner and outer ring diameters are key dimensions to ensure the level of ring tightening with the shaft and the housing (refer to paragraph 3.1.2.1 - Tightening – Hoop stress).
As a result, the geometrical tolerances applied to these diameters are critical and must be calculated to ensure target tightening is maintained for all operating conditions and the entire range of potential shaft/housing dimensions. As discussed in the 'Tightening – Hoop stress' paragraph, the calculated theoretical interference should be feasible in practice: mounting/dismounting (loads and heating/cooling temperatures).

3.1.6.6 Contact angle

The contact angle (see Figure 31, α) is a central design parameter that will impact the contact pressure, the sliding rate and the axial/radial load repartition. The global design approach is to target the minimum contact angle value leading to acceptable contact pressure values. Indeed, an increase of the contact angle will result into longer rollers and a decreasing of the contact pressure (increase of the contact area). Nevertheless, it will also increase the sliding and the axial loading that should be minimized. The contact angle range is limited by the maximum roller length in combination with the available space and the minimum roller length to support the load will with a permissible contact pressure.

3.1.6.7 Roller diameter

The roller diameter has a direct influence on the radial internal clearance. The variation in roller diameter will increase or decrease the radial internal clearance. Additional information is provided in the section on radial internal clearance.
Note: As this parameter is directly linked to parameter radial clearance, it will not be retained for the conclusions provided in 3.5 as considered already covered by the other mentioned parameter.

3.1.6.8 Roller profile

The roller profile is a critical element as it defines the contact stress profile (see Figure 40). It may take several shapes and dimensions. For cylindrical rollers, profiles can be defined as logarithmic or crown in terms of radius. The roller profile is selected on the basis of its ability to optimize the contact stress profile while avoiding (if possible) full contact/edge contact for all load/misalignment conditions.
Additional information is provided in the sections on Osculation and Roller-Raceway Full Contact & Truncation.

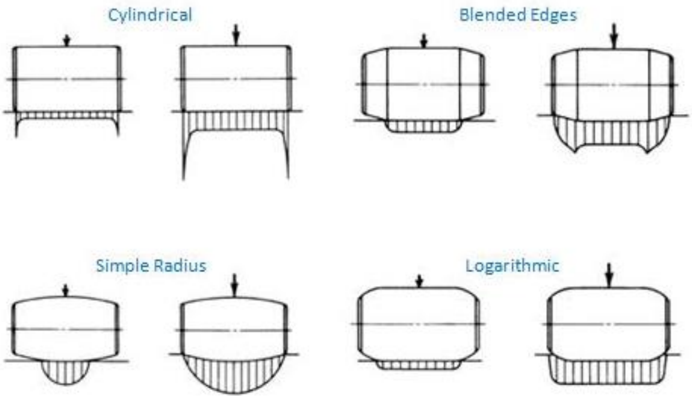

Figure 40: Example of resulting contact stress due to roller profile [56]

Note: As this parameter is directly linked to parameter contact pressure, it will not be retained for the conclusions provided in 3.5 as considered already covered by the other mentioned parameter.

3.1.6.9 Roller length

The roller length has influence on the axial clearance and also the contact stress. The variation of the roller length will increase or decrease the axial clearance or contact stress. Additional information is provided in the section on axial clearance.

Note: As this parameter is directly linked to parameter axial clearance, it will not be retained for the conclusions provided in 3.5 as considered already covered by the other mentioned parameter.

3.1.6.10 Roller geometrical tolerances (chamfers, circularity, perpendicularity)

The roller geometric tolerances are a key factor for ensuring optimal bearing dynamic behavior. The aim is to achieve the best roller geometry while managing the manufacturing capabilities. Tolerance limits are adjusted in accordance with the pertinent ABEC and ISO standards.

It is important to note that a smaller chamfer dimension is always beneficial for avoiding detrimental edge contact between the roller and ring shoulders. The minimum chamfer size is also contingent on manufacturing capabilities (as a function of the roller size and geometry).

3.1.6.11 Roller diameter roughness

As previously mentioned (section on Lubrication), the roughness of the contact surface is a direct contributor to the efficiency of the lubrication in terms of separating surfaces. Even if the roller roughness is an order of magnitude below the raceway values, it is important that this parameter is defined and controlled.

3.1.6.12 Roller faces roughness

The roller faces are exposed to pure sliding contact, resulting in a significant risk of wear. The surface roughness needs to be managed and controlled to limit this phenomenon.

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3.1.6.13 Cage pocket geometry

The cage pocket geometry tolerances are important to ensure the good performance of the roller/cage contact. As explained for the cage pocket clearance, the aim is to ensure a minimum clearance value around the rollers but nevertheless define the roller alignment and circumferential distribution to guarantee the load repartition and dynamic behavior. The calculation of the optimal geometry is governed by manufacturing capabilities.

3.1.6.14 Cage guiding diameter and cage landing clearance

The clearance between the cage and the ring guiding diameters (see Figure 41) has an effect on the motion of the cage. The aim of the design is to minimize this gap in order to ensure optimal guidance and avoid any dynamic parasitic effects (unbalanced loads) while also avoiding any risk of tightening (after thermal expansion of the ring or due to high centrifugal forces, for example). This parameter also affects the oil flow inside the bearing and the ability of particles to exit the bearing.

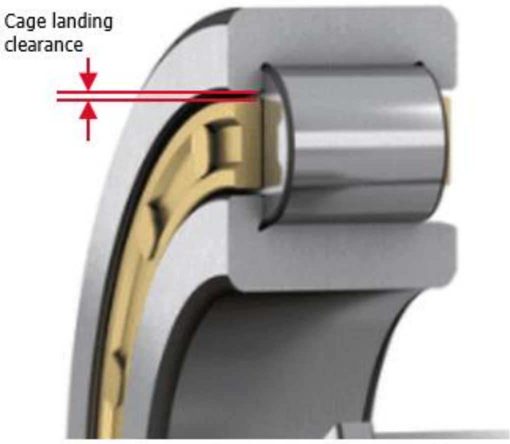

Figure 41: Example cage landing clearance

3.1.6.15 Rings/shaft/gear raceway roundness and location

These tolerances are based on ISO/ABEC standards. The ISO/ABEC class is defined based on customers inputs, process capabilities and operating condition severity. This paragraph is linked to the roller geometrical tolerances.

3.1.6.16 Rings/shaft/gear raceway profile

The raceway profile works with the roller profile to optimize the contact stress profile avoiding (if possible) full contact/edge contact for every load/misalignment condition.
Additional information is provided in the Osculation and in Roller-Raceway Full Contact & Truncation paragraphs.

3.1.6.17 Rings/shaft/gear raceway roughness

The lubrication regime (full/mixed or boundary) is directly influenced by the roughness of the raceway as illustrated by the calculation of the lambda ratio. Decreasing the roughness helps to guarantee the full separation of surfaces and avoid early surface wear. Certain process limitations make it impossible to permanently reduce the surface roughness, in particular for SRB raceways due to their complex shapes. The drawing roughness value represents the maximum permitted value based on the values calculated for film thickness (for the different running conditions) in alignment with the process capabilities. Additional information is provided in Lubrication paragraph.

3.1.6.18 Roughness of cage piloting surface on ring/shaft/gear

With regard to the roller faces, the contact area between the rings/shafts/gears and the cage faces pure sliding contact at high speed, leading to a significant risk of wear. The surface roughness needs to be managed and controlled to limit these phenomena.

3.1.7 Material

The material choice for ring and rolling elements is dependent on several factors, as listed below, and can be regarded as an individual design parameter:

- If the ring has structural features (flanges, squirrel cage or integrated gears) then case-hardened (carburized or nitrided) steel should be used. If not, through hardened steels may be used.
- The application temperature is a key parameter, as steels will be subjected to significant hardness decreases and potential deformations if the usage temperature is close to the steel tempering temperature.
- In certain applications, the use of stainless or corrosion-resistant steels may be required.

The ring steel can be implemented directly by the system designer, typically as a function of other ring features (e.g.: integrated gears). Generally, the following steels are used for transmission bearings:

- Non-structural through hardened steel: 100C6 & M50
- Case-hardened/structural steels: 32CDV13; M50NiL; 16NCD13 but also AISI9310, VascoX2, Pyrowear 53

Rolling elements are made from M50 or 100C6. However, several development projects are currently in progress for both helicopter and engine gearboxes with the aim of introducing ceramic (Si_3N_4) rollers in planetary gearboxes. It is important to highlight that the material characteristic such as the Young modulus, thermal expansion coefficient, fatigue limit and fracture toughness are necessary inputs for the calculation. For high-speed aerospace bearings, only steels produced by VIM-VAR processes are used (for material cleanliness). All of them are defined in accordance with an AMS standard. Silver-plated 40NCD7 or brass alloys cages are primarily used. Polymeric cages can also be used but the polymeric materials may weaken overtime due to an ageing process that accelerates at high temperatures.

3.1.7.1 Hardness

The hardness guarantees the mechanical resistance of the part due to its direct link to the steel's yield stress. The target hardness level will depend on the steel choice and on the tempering temperature. In order to permit the use of steel at high temperatures, a higher tempering temperature may need to be applied to decrease the target level of hardness and consequently the mechanical resistance of the rings.

The hardness level may vary due to a hardness problem resulting from manufacturing issues (heat-treatment, grinding burn) or from the mechanical characteristics of the steel. Both may potentially lead to early failure (spalling, cracks).

3.1.7.2 Case-hardening depth (IR/OR)

For rings made out of structural steels (case-hardened), the depth of the hardened layer needs to be calculated and the process feasibility to reach this depth target determined. The determination of the target case-hardening depth is contingent on the calculated maximum shear stress depth. The objective is to include the entire subsurface shear stress area (= critical area as the site of initiation for subsurface rolling contact fatigue cracks) in the effective hardened layer of the material. It is important to note that the hardness profile in terms of depth presents an important gradient from the surface to the core hardness. The minimum hardness value is calculated below the maximum shear stress depth to ensure the required level of steel mechanical resistance. Subsequently, it is important that the process feasibility to reach the target hardness depth is determined. In any case, certain process limitations (duration of the heat-treatment, microstructure evolution, etc.) may restrict the achievable depth. The maximum hardened depth is generally higher for carburized steels compared to nitrided steels. Moreover, the limitation may also be geometrical if the target case-hardening depth is not adapted to the ring thickness.

3.1.7.3 Residual Stress

Residual stresses result from the manufacturing process (finishing and heat-treatment). Residual stresses are often characterized by the surface values and depth gradient in both directions (tangential and axial). The residual stresses are in the same order of magnitude as the contact or structural stresses. As a result, they can have a major impact on fatigue resistance and crack propagation.
Residual stresses are rarely included in the bearing design calculations or in bearing specifications. This means that there is potentially an opportunity to optimize them. Nevertheless, their magnitude is generally controlled in order to maintain a defined compression level (as tensile stresses are known to be detrimental to fatigue strength).

3.2 Particular parameters for bearings with integrated races

In addition to the parameters mentioned in Section 3.1, other parameters need to be taken into account for the introduction of integrated raceways.

3.2.1 Material and surface treatment

The right choice of the material for the application is vital to ensure the reliability and flaw tolerance of the component, not only for conventional bearings, but also for bearings with integrated raceways. For bearings with integrated raceways, innovative, high strength materials are required, which are suitable for bearings and the corresponding parts of the rolling contact [60]. In addition to the information given in chapter 3.1.7, some particularities of bearings with integrated raceways can be discussed according to the choice of material. Based on field experience and test results, the surface hardness should be in the range of 700 – 800 HV and the case depth should average 1 mm at 560 HV for power gearboxes. For high-load bearings, a surface hardness of at least 700 HV and a case depth averaging 1mm at 650 HV is required [60]. For the heat treatment, it is important to ensure that the chosen material can achieve the required hardness, case hardening depth, microstructure and residual stress. With a well-suited heat treatment process (e.g. duplex hardening), the material's RCF behavior and performance under contaminated conditions (operation with contaminated oil) can be improved and designed for better reliability of the components [60].

Carburizing is widely used as an effective technique to increase the surface hardness of steel. Available experimental stress data indicate a compressive state of stress located in the carburized layer, whereas the theoretical stress field is known at each depth and provides information beyond the treated layer. It shows that the core treated component is exposed to a tensile state of stress to balance the compression of the carburized layers. As an example, Figure 42 presents a 3D finite element simulation of the residual stresses in an outer gear ring with an integrated bearing raceway on the inner side. This view highlights the tension and compression areas, which roughly represent the carburized layer (in blue), achieved by the diffusion of carbon during carburizing, and the core material (in red), not enriched by carbon [5].

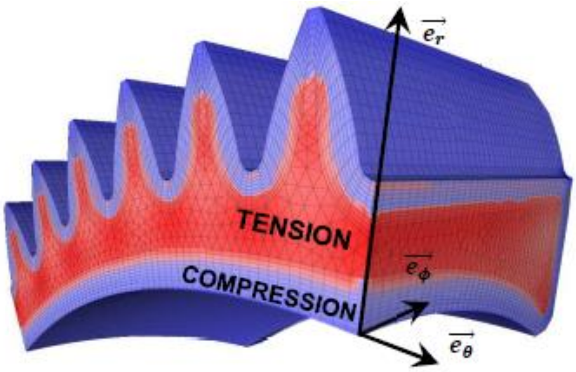

Figure 42: Signed von Mises stress field of simulated residual stresses due to the carburizing [5]

The impact of these residual stresses on the crack propagation can be detected, as shown in Figure 43. The type of crack propagation changes when the main crack reaches the end of the carburized layer. It corresponds to the bifurcation of the main crack into the core, which is characterized by a double curvature (ellipsoidal shape) crack path.

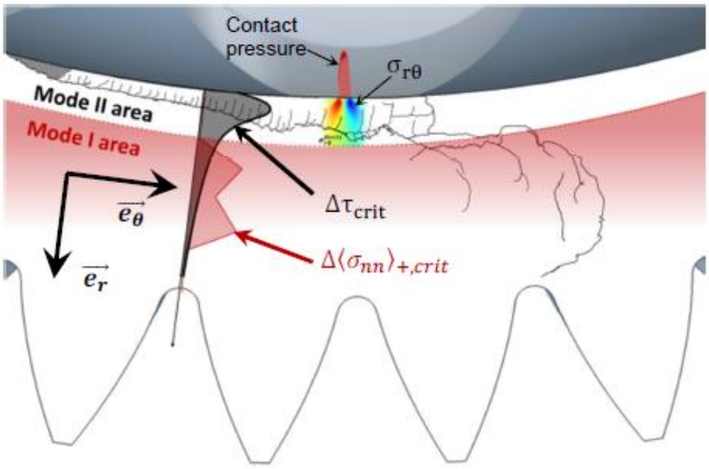

Figure 43:Stages of crack propagation [5]

The start of the crack bifurcation is located at the interface between mode I and mode II, where the residual stress becomes positive. The crack behavior can vary widely, depending on the dominant propagation mode. The almost 'coplanar' propagation of the main crack with numerous small branches in the carburized layer becomes a network of 'curved' cracks in a 'C' shape in the core of the rim [5]. Experimental data and experience have proven that the carburized layer has an influence on the crack propagation path and is a driver of component reliability.

The responsibility of designing the rolling raceway does not, at this point, solely lie with the manufacturer, but also with the manufacturer of the shaft, gear or housing. The technical discussion and exchange of ideas between these parties must be included in an early stage of the design process and the required values for this parameters must be aligned. Based on this, particular limitations may be necessary according to the materials of the shafts, gears or housings in comparison to the use of conventional bearings with non-integrated bearing raceways.

3.2.2 Body stress

The raceways of the outer/inner rings in non-integrated bearings are typically exposed to a different load situation than integrated raceways. While there is a simple stress state in non-integrated raceways, integrated raceways are loaded with a superpositioned load in many cases (e.g., torsion stress of shaft + rolling contact stress). With regard to the fatigue behavior of the raceway, this may lead to a higher risk of rolling contact fatigue and also degradation in terms of the component's flaw tolerance.

3.3 Parameters for planetary gears with integrated raceways

3.3.1 Rim thickness

The impacts of rim thickness and gear stiffness are currently being explored as part of an ongoing research project (FVA795 [2]), which has verified a direct link to component reliability and crack propagation behavior. It has been proven that the qualitative loading of a gear is influenced significantly by the stiffness thereof (see Figure 44).

Looking at the stress distribution of a stiff gear, the stress is focused on a small area around the loaded teeth of the gear and mainly driven by the bending load. The loading is more or less local and the maximum stress is also smaller in comparison to an “elastic” gear loaded with the same force. An elastic gear shows a different loading situation. Due to the loaded teeth, the “elastic” gear is deformed and a combined loading of normal, shear and bending load occurs. The stresses can not longer be considered as local and are distributed over the full gear. Moreover, the maximum stress level is higher in comparison to the stiff gear rim. Due to the ovalization of the gear itself, there is also a high potential of an increase of the contact pressure between the rolling elements and the bearing raceway, which could lead into a reduction of the reliability of the components.

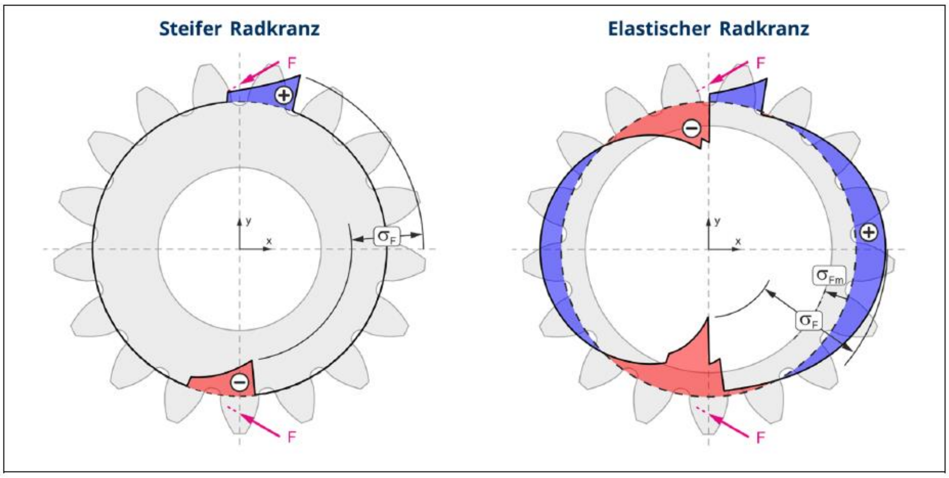

Figure 44: Comparison of the qualitative loading of a stiff (left) and elastic (right) planetary gear [2]

Lewicki [1] researched the link between rim thickness and crack propagation direction in several experimental and simulative studies. The findings illustrated comparable results for testing and simulation, with the outcome that a ratio of tooth to rim thickness below 0.5 leads to crack propagation through the rim. Ratios higher than 0.5 will lead to a crack through the tooth (see Figure 45). This research was only conducted for cracks that start at the tooth base and further study is required [1].

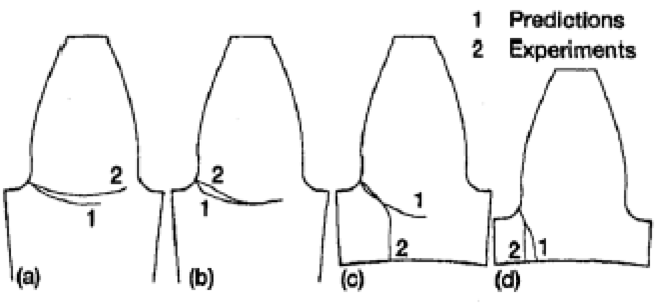

Figure 45: Comparison between predicted and experimental crack propagation paths for a)m_B=3.3,b)m_B=1,c)m_B=0.5 and d)m_B=0.3 [1]

ZF group follows the rules of a minimum rim thickness that has been proven to be safe in operation in the range of 1,2 x tooth height. However, this value might be subject to more detailed analysis based on the specific requirements of the application (e.g. level of loads).

3.3.2 Bearing race contact pressure

As mentioned in chapter 3.1.2.3, the bearing race contact pressure (Hertzian Pressure) may be significantly increased for a planetary gear, as the rim thickness is small enough to generate an ovalization of the gear ring. The ovalization of planetary gears and therefore also of the bearing raceway is often increased due to the limited design space and the requirements of thin-rimmed structures in comparison to conventional bearing installation locations (see also Figure 44). The smaller the rim thickness, the higher the influence on the raceway contact pressure of the planetary gear, driven by the ovalization. Finally, the peak values of the contact pressure could lead into a degradation of the reliability and flaw tolerance of the component. In comparison to conventional bearing applications with non-integrated bearing races, the limit for the maximum contact pressure must also be considered for the raceway material of the planetary gear.

Note: This parameter is providing additional information to the paramter “contact pressure” presented in 3.1.2.3. As design parameter, “contact pressure” is already addressed as a design parameter for bearings in general, it will not be addressed specifically for planet gears in the conclusions provided in 3.5.

3.3.3 Contact ratio

The contact ratio is mentioned in this report for the sake of completeness. Generally, this parameter is not really a design parameter for the bearing, but more for the gear. Nevertheless, it as a high influence on the stress situation for the bearing and must be considered during the design process. Failure in pitting of gear teeth is higher for lower contact ratio gearing. In other words, the higher the contact ratio of the planet gear, the lower the teeth contact pressure, which help prevent teeth surface damage and degradation. An underlying effect of the decreased resistance is the change in the load zone by varying the value of the contact ratio. Higher contact ratio gearing is achieved by modifying the addendum of the gear teeth [16]. For standard gears, a significant reduction in stresses on the gear is achieved by using a contact ratio of ≥ 2 [17]. The contact ratio also has an influence on the body stress of the gear, as the load introduction is distributed on more than one tooth. Generally, this is valid for standard gears. Going into the direction of high contact ratio planet gears, whereas the rim thickness is reduced due to an increased dedendum height, the reduction of the rim thickness has a higher impact on the stress state due to the ovalization compared to the load distribution in case of high contact ratios ([21],[22],[45]). Nevertheless, the contact ratio is considered not to be the main driver for the stress state of a planetary gear.

3.3.4 Width of load zone (loaded sector) and number of rolling elements

The width of the loadzone and also the bearing clearance can be seen as a design parameter also for conventional bearings in general. But as planetary gears are used with integrated raceways, this parameter gets more important, as it is a direct contributor to the ovalization of the planetary gear and could increase this phenomena additionally. The planetary gear is loaded with the dual meshing of sun and ring gears. This outer load is counterbalanced by the inner bearing load, which is distributed over the individual roller elements. The width of the load angle, together with the bearing clearance, plays a significant role in the load distribution and deformation of the planetary gear. With a bearing clearance close to zero, the load is distributed largely evenly over half of the raceway diameter. In comparison, with a higher bearing clearance, the area of load distribution decreases and the deformation of the planetary gear increases. As a result, this leads to a higher stress level in the planetary gear and a reduction in lifetime and safety factors [13]. The width of the load zone, stands in a direct relation to the bearing clearance (see Figure 46). The optimum width of the load zone can be understood as shown in the upper left corner of Figure 46. Typically, recommended values for the load zone are < 180° in order to reduce the peak stresses due to the ovalization, initiated by the meshing. A large load zone width will introduce an additional pushing force against the ovalization of the gear rim and will lead to an increase in local stress at the ends of the load zone. A too small width of the load zone has a negative influence as well. It contributes to a higher ovalization and increases the stress level in the same manner. The number of rolling elements has a significant influence on the load distribution for the integrated bearing raceway. Moreover, the stiffness of the planetary ring is affected by the support capabilities of the rolling elements. With fewer rolling elements, the deformation of the planetary gear increases, leading to a higher stress level. In general, the number of rolling elements is an influential factor for the planetary gear reliability and flaw tolerance.

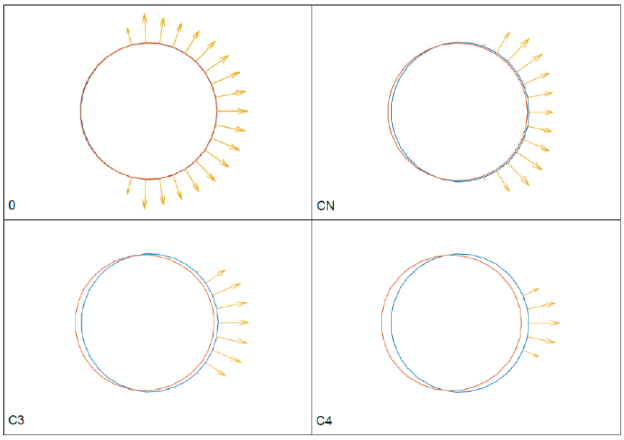

Figure 46: Bearing load distribution for different bearing clearances for equal rim stiffness [2]

3.3.5 Tooth root stress

Tooth root stress is a key aspect for non-integrated gear optimization processes, which significantly impacts the overall stress amplitude and therefore the fatigue life. Tooth root stress is mainly driven by the stiffness of the gear/gear ring (see Figure 47). If material limits are exceeded, it could lead to crack initiation. For planetary gears with a thin rim, tooth root stress cannot be analyzed in isolation at the loaded tooth while the course of stress grows in complexity. An integrated raceway on the inner side may also lead to an increased stress level, which may impact reliability and flaw tolerance. The criticality of the tooth root stress without considering the influence of the other stress contributors (e.g. body stress, residual stress) is low.

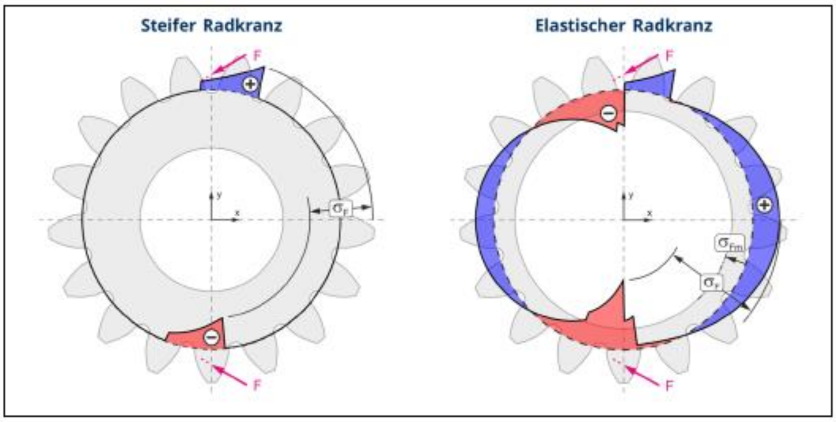

Figure 47: Comparison of conventional gear with stiff rim (left) and elastic rim as for planetary gears (right) [2]

3.4 Example of typical parameter values for SRB and CRB bearings

For some of the design parameters mentioned, it is possible to provide typical limits based on the findings of past projects (see Table 6). These values are not necessarily directly applicable to rotorcraft gearbox design, but can be seen as an orientation for projects and a starting point for each development and design process.

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SRB/CRB		Comments
Tightening Hoop Stress	Case hardened steels ~around 250 MPa (function of steel thoughness) Through hardened steels : max 200MPa	
Contact Stress	Below 1600 MPa in nominal condition Below 2200 - 2400 MPa in max conditions	Highly depends on final application and duty cycle partition
Misalignment	Misalignement until full contact / edge contact	
Oil flow	NA	Calculated on the application from the bearing estimated power losses and the oil in-oil out temperature variation
Internal radial clearance	(0.015 to 0.22 mm) ; Internal radial clearance value should guarantee that, with the max ring deformation, the loading zone angle is below 160-180°	
Axial clearance	CRB : linked to roller geometry and skewing risk SRB : depends on radial clearance, contact angle, skewing risk	
Cage pocket clearance	(0.13-0.45 mm) : to avoid fatigue due to roller skewing	
Osculation	(0.50-0.59) : to avoid full contact	
Contact angle	(8 to 18°) : to optimize pressure	
Roller length (typical length to radius ratio)	Length/Diameter superior or equal to 1 Length/Diameter max ~ 1.25	
Roller diameter roughness	0.05 to 0.1 µm	Values could be limited by the manufacturing process
Roller face roughness	0.15 to 0.4 µm (standard 0.2 µm)	
Cage landing clearance	0.05 to 1 mm	
Ring raceway roundness	0.00075 to 0.001 mm	
Ring raceway roughness	0.08 to 0.2 µm	Values could be limited by the manufacturing process
Roughness of cage piloting surface	0.4 µm	
Hardness	Surface hardness : (630) 650HV to 850HV (up to 1100HV for M50NiL nitrided)	
Case-hardening depth	Nitrided steels : from 0.5 to 0.9 mm (HV_{core}+100) Carburized steels : from 0.3 to 1.6 mm at 550HV	
Residual stress	Surface : -400 to -1000 MPa (-1200MPa for M50NiL nitrided) Case-hardened layer : -200 to -400 Mpa	

Table 6: Typical design parameter values [3]

3.5 Summary and conclusion

The key design parameters in chapter 3 were chosen to give an overview of the mandatory design parameters. The parameters are applicable for both, standard bearings and bearings with integrated raceways. Additional parameters were added for bearings with integrated raceways and for planetary gears with integrated raceways and their particularities were described.

Moreover, all the parameters can be ranked by their contribution to the reliability and damage tolerance of the planetary gear system. The following table gives an overview of the parameters of chapter 3 by assessing their influence on reliability and flaw tolerance:

Parameter	criticality for reliability and flaw tolerance of bearing races (low/medium/high)	Rationale
Parameters suitable for all bearings		
Bearing type	low	Scope of this task is not to evaluate differences between bearing types
Tightening – Hoop Stress	low	Not present for integrated raceways
Roller raceway full contact & truncation	high	Stress peaks leading to higher risk of RCF
Contact Stress	high	High stress amplitudes leading to high risk of RCF. A main parameter contributing to the contact stress is the roller profile.
Misalignment	high	Misalignment leads to high local stress peaks and risk of RCF
Slippage and P.V.	high	Slippage leads to increased wear. In the event of crack initiation, it could lead to a load situation that initiates crack propagation
Lambda ratio lubrication	high	Ratio is directly linked to the risk of spalling and, therefore, reliability of the raceway
Oil flow	low	Oil flow important for temperature management and only indirectly influences reliability
Oil cleanliness / pollution	high	Overrolling of particles is a main contributor behind damage to the raceways and could lead to a reduction in the reliability of the raceway
Bearing life	low	Calculations are mainly based on idealistic boundary conditions and inaccurate in terms of reliability
Internal radial clearance and roller diameter	high	Direct influence on loading situation and contact stress. (see also Contact stress)

Parameter	criticality for reliability and flaw tolerance of bearing races (low/medium/high)	Rationale
Axial clearance and roller length	high	Direct influence on loading situation and contact stress (see also Contact stress).
Cage pocket clearance	high	Direct influence on loading situation and contact stress. (see also Contact stress)
Osculation	high	Impact on full contact / edge contact (see also Roller raceway full contact & truncation)
Inner or outer ring diameter	low	Negligable for integrated raceways
Contact angle	medium	Direct influence on sliding rate and contact pressure with potential risk of RCF. As it is only relevant while exceeding the design contact angle, it is ranked with medium.
Roller geometrical tolerance	medium	Influencing lubrication efficiency, minor influence in comparison to other geometrical parameters and therefore medium
Roller diameter roughness	medium	Direct influence on lubrication efficiency (see also lambda lubrication ratio)
Roller face roughness	medium	Direct influence on sliding of the rolling element and consequently the loading situation. Usually the economically possible limits of manufacturing are already reached, so that no huge impact of additional optimization is expected. Therefore it is ranked as medium
Cage pocket geometry	medium	Influence on loading situation but smaller than axial clearance
Cage guiding diameter and cage landing clearance	low	No direct influence on the raceway, but on the reliability of the cage
Rings/shaft/gear raceway roundness and location	low	Complementary to roller geometrical tolerances
Rings/shaft/gear raceway profile	medium	Complementary to roller profile
Rings/shaft/gear raceway roughness	medium	Complementary to roller diameter roughness

Parameter	criticality for reliability and flaw tolerance of bearing races (low/medium/high)	Rationale
Roughness of cage piloting surface on ring/shaft/gear	low	Complementary to guiding diameter and cage landing clearance
Material and material cleanliness and composition	high	Material has great effect on fatigue limit and fracture toughness but is generally not freely selectable. It is not within the scope of this project to fully characterize the impact of all different characteristics that may be impacted by the material selected with regards to bearing reliability and flaw tolerance. The material cleanliness (melt quality) defines the amount of potential crack initiation locations. The material composition has an influence on the microstructure and potential crack initiation locations.
Hardness	high	Hardness has direct influence on the mechanical properties of the steel and can contribute to cracks or spalling
Case hardening depth	high	Mechanical properties of the steel change at end of hardening zone and can influence the flaw tolerance
Residual Stress	high	Change in stress level could lead to decreased flaw tolerance
Particular parameters for bearings with integrated raceways		
Body stress	high	Generally higher stress level due to superposition of loads at the raceway compared to conventional bearings with non integrated raceways. The higher stress level increases risk of spalling and crack initiation.
Material and surface treatment	high	The selection of the material and the corresponding heat treatment process influences the stress state and the resistance against damages and flaws.
Parameters for planetary gears with integrated raceways		

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Parameter	criticality for reliability and flaw tolerance of bearing races (low/medium/high)	Rationale
Rim thickness	high	As demonstrated in previous research studies, rim thickness directly influences the loading and stress situation of the gear. A small rim thickness leads to an ovalization of the gear with a higher stress level and a combination of bending, shear and normal load.
Contact ratio and tooth root stress (linked to body stress)	high	The body stress for planetary gears has a high criticality as for integrated gears in general. The contact ratio influences the stress state and level in the gear and directly affects the body stress. For thin rimmed planetary gears, the body stress is mainly driven by the ovalization and high contact ratios can even lead to to a stress increase (reduced rim thickness due to increased dedendum height). The tooth root stress may also affect the general body stress when associated to thin rimmed planetary gears.
Width of load zone (load sector) and number of rolling elements	high	Width of loadzone (load sector) and number of rolling elements has a direct influence on stress state and level and also the amount of ovalization of the gear. A similar effect was described for the parameter rim thickness and the axial clearance. The ovalization is mandatory for reliability and flaw tolerance

Table 7: Summary and classification of selected design parameters

Based on the findings from this report, research conducted by SKF and ZF and its relevance in current research projects, the design testing and selection of the correct parameters, isolated for each component, are well established processes, yet the interaction and impact of the parameters during operation remains largely uncontrolled and unresearched to date. However, it is possible to focus on some main parameters (highlighted with the criticality "high") contributing to the reliability and flaw tolerance in bearings for integrated bearing races (see Table 7).

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One can say, that many of the mentioned parameters are influencing each other and can not be optimized without having an influence on other parameters. For a better overview, the parameters with the classification 'high' are shown in Figure 48 together with their dependencies and interconnections. For the most part, they all contribute similarly to the overall stress state of the integrated bearing races and therefore the potential of crack initiation. Oil cleanliness is a unique factor in this regard, as it does not directly contribute to the stress state, but does increases the risk of crack initiation due to particle overrolling and indentations on the raceway surface (pre-damage).

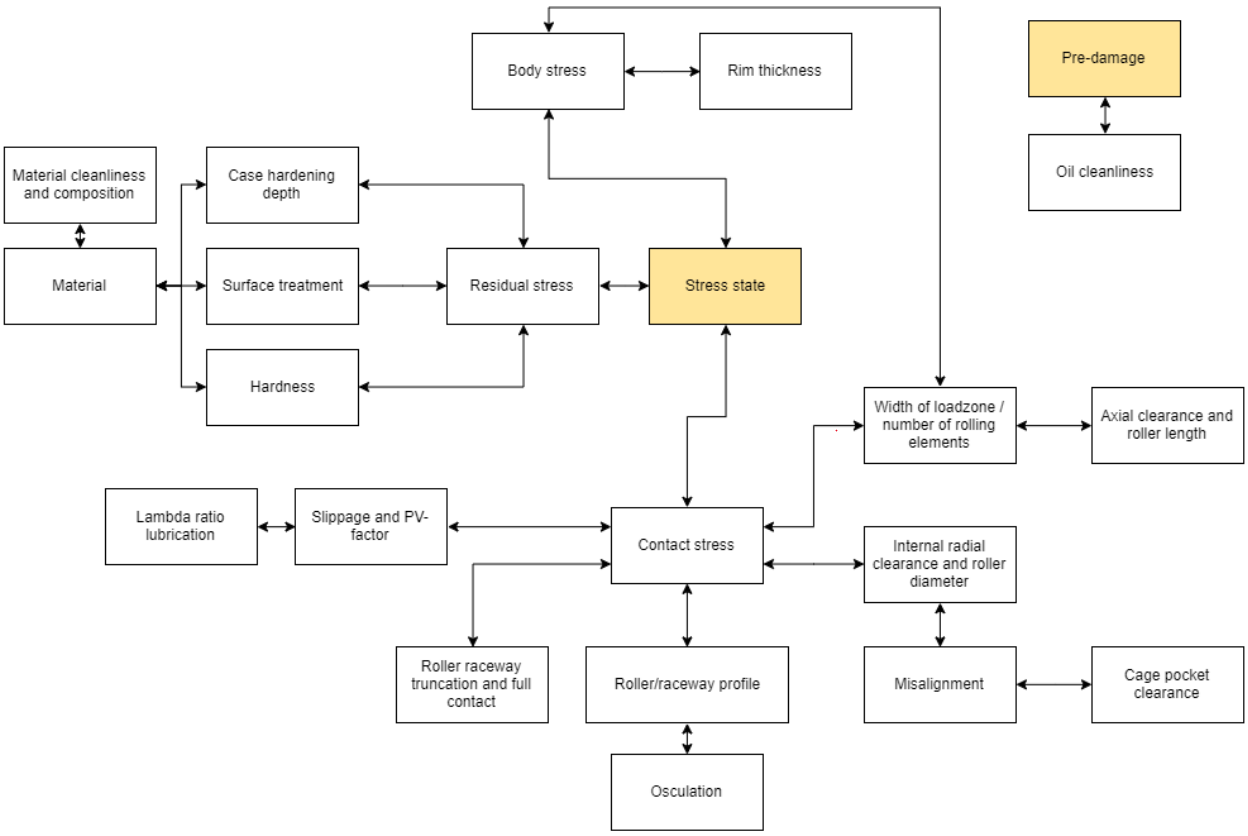

Figure 48: Interaction of critical parameters

Further on in the project, the defined and presented parameters will be used to define adequate tests and find solutions for reducing the possible number of catastrophic failures of a rotorcraft main gearbox by considering them during the design process.

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- FVA 30 - Calculation and simulation
- FVA34 - Influence of the tooth root rounding and the helix angle on the stress and strength of helical gears
- FVA51 - Simulation of the operating behavior of a planetary gear stage
- FVA127 - FE-spur gear simulation
- FVA179 - Particle damage – influence on foreign particles in bearings and actions for avoidance
- FVA336 - Increase in the service life of components subject to rolling under contaminated lubricant
- FVA374 - Considering of special events for the tooth root strength
- FVA425 - Tooth root core strength
- FVA 463 – Service life of straight and helical gears with different rim thicknesses and connection geometries
- FVA485 - Digital guidline for the fatigue strength calcuation
- FVA493 I - Influence of the operational conditions on the amount of particles in gearbox lubricants
- FVA493 II -Influence oft he amouunt of particles of gearbox lubricants on the secondary wear
- FVA498 - Extended fatigue strength of gears
- FVA504 - Mixed friction rolling bearing fatigue
- FVA625 - Dynamic simulation and operation analysis of cylindrical roller bearings under consideration of the surrounding construction
- FVA662 - Development of improved and less sensitive to flow bearings for high rotational speeds
- FVA663 - Harmful roller bearing slip
- FVA707 - Cracks on bearing rings
- FVA709 - FE-tooth root strength optimization
- FVA718 - Local tooth root strength of spur gears for alternating bending load
- FVA732 - Creation and implementation of a tool for fast numerical stress analysis for any tooth cross sections in RIKOR
- FVA743 - Notch stress with FEM
- FVA761 - Influence of grinded ground tooth root roundings to the tooth root load capacity of case-hardened spur gears
- FVA774 - Planetary gear deformation
- FVA795 – Tooth root and ring strength of thin rimmed planetary gears
- FVA796 – Calculation and measurement of load sharing in planetary rolling bearings
- FVA798 – Fatigue life for surface damages
- FVA866 – Influence of short overloads on the life time of rolling bearings

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