# TEST STRATEGY FOR PLANETARY GEAR SYSTEMS OF MODERN AIRCRAFT ENGINES

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**Abstract**
For the production of the planetary gearbox for modern aircraft engines, it is of the highest priority that the high performance planetary gearbox is tested after assembly and meets the high quality requirements. In order to ensure this, a method will be developed with which a qualitative statement of the components and the overall system can be made. The focus is placed on vibration measurements due to various advantages like analysis of the complete system, detection of random faults and non-destructive testing method.

## 1. INTRODUCTION

Aircraft engines of the future must be both, more economical and more efficient. This can be achieved by using a wide variety of approaches. One of these is to change the architecture of the engine. Normally, all blades within an engine rotate on a single shaft. As a result, not all pressure ranges of the turbine run in the optimum range. In order to resolve this conflict, it is advisable to install a planetary gear between fan and intermediate pressure compressor. The planetary gear is coupled with the fan and the low-pressure turbine. As a result, the fan rotates slower and the low-pressure turbine faster. Ultimately, the bypass ratio and efficiency are increased by adjusting the speeds for the optimal operating points, and noise emissions are reduced by the slower rotating fan.

Planetary gearboxes have various advantages over fixed-shaft gearing systems: Compact structure and high power density. For the production of the planetary gearbox, it is of the highest priority that the high performance planetary gearbox is tested after assembly and meets the high quality requirements. In order to ensure this, a method will be developed with which a qualitative statement of the components and the overall system can be made. For high performance planetary gearboxes it is more important, because failures end in catastrophic accidents.
Several physical measurands like temperature, torque, acceleration can be measured. The focus is placed on vibration measurements due to various advantages like analysis of the complete system, detection of random faults and non-destructive testing method. With this method also come disadvantages: the optimal measuring point is not always accessible. Particularly disadvantageous with rotating parts. As well as a limited bandwidth of the recording sensors. [1] [2] Vibration response of sensors fixed to the gearbox housing can measure useful signals for diagnostic the entire gearbox system. [3] For this purpose, a planetary gearbox will be assembled on a test bench in later investigations and the test bench with planetary gearbox is mapped virtually in a simulation for preliminary investigations.
Goal and mission is to develop a robust detection of component and assembly errors. Especially the detection of known and unknown faults. This is where one of the advantages of structure-borne sound becomes particularly apparent, since structure-borne sound always considers the entire system. The concept is based on three blocks (FIG. 1): developing a simulation for preliminary investigations for better understanding, structure of a testing method on a test bench, last the analysis combined from simulation and tests. In this paper the focus is on the first step, developing a simulation preliminary investigation for better understanding the vibration measurements on planetary gearboxes.

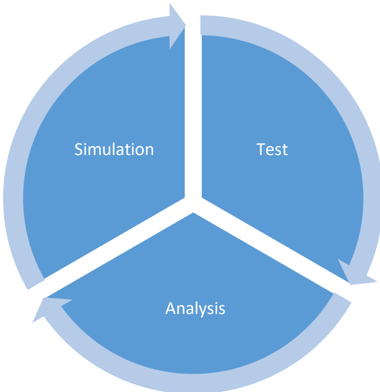


FIG. 1: Test Strategy

## 2. MODEL DEVELOPMENT

Planetary gear boxes usually consist of a sun gear, several planet gears and a ring gear. The planet gears roll between the sun and ring gear. The advantage of planetary gear units is that the load is distributed over several planetary gears. This makes it possible to transmit high torques with a compact design. Thereby several gear pairs meshing at the same time. This results in complex excitation mechanisms. Internal forces and external loads, together with the gear´s own behaviour, lead to dynamic effects. Since the wheels rotating on the revolving axles orbit a central wheel, planetary gears are also systems with time-varying excitations. Different vibration sources and transmission paths lead to a complex vibration diagnosis for fault identification of planetary gearboxes. To carry out the fault identification, the focus was placed in advance on vibration measurement. Since accelerometers for vibration measurements are usually mounted on the housing, a wide variety of phenomena must be taken into account. One is the time varying distance between the fixed transducer and the planet gears. This leads in different transmission paths. In order to identify a fault in a planetary gearbox, it is necessary to take a closer look at the origin of the vibrations. For this purpose, various analysis techniques have been developed. [2-5]

In the simulation-based representation of the vibration behaviour of gearboxes, the level of detail is of particular interest. Since the reduction of the model means a lower computational effort, but also a loss of information. The theoretical consideration of the dynamic behaviour requires the transfer of reality into mathematical substitute models. In transmission design, the force excitation is not directly on the housing structure, but on the gear teeth. Structure-borne noise calculations can be solved in combination in the multi-body simulation by supplementing the dynamic model with sufficiently accurate forms of representation of the structure subject to vibration, such as the wheel-shaft-bearing-housing system. Sufficiently accurate discretized finite element meshes are suitable for this purpose.

### 2.1. Analytical simulation of the vibrations

The primary goal of the test strategy is to find imperfections and small defects in the planetary gearbox. Since mechanical imperfections are always a source of vibrations, it is possible to make a statement about the condition of the planetary gear based on these. The following section describes how component-specific errors are expressed in these signals.

As an example for the calculation of the tooth meshing frequencies, a planetary gear is considered, where the input shaft is connected to the sun gear and the output shaft is connected to the planet carrier. The ring gear is fixed with the housing and does not move.

#### 2.1.1. Gear meshing and roll over frequencies

The sun gear $(z_1, n_1)$ and planet gear $(z_2, n_2)$ or planet gear and ring gear $(z_3, n_3)$ are in mesh with each other. This results in the following transmission ratios:

$$\frac{n_2}{n_1} = \frac{z_1}{(z_1 - z_2)} \tag{1}$$

Compared to bearing vibrations, strong vibrations occur in gear drives even in the fault-free theoretical state. Within the gearbox, the tooth meshing frequency has the greatest significance for the force excitation. Further harmonic orders are contained in the frequency spectrum, whereby their amplitudes decrease towards higher orders. In slow-running gears, the frequency lines of the higher harmonics often no longer occur because of the low energy content. [6]

Tooth meshing frequency

$$f_z = \frac{z_1 \cdot z_3}{(z_1 + z_3)} \cdot f_{n1} = z_3 \cdot f_{n4} \tag{2}$$

Sun-gear rollover frequency

$$f_{nS} = \frac{k \cdot z_3}{(z_1 + z_3)} \cdot f_{n1} = k \cdot \frac{z_3}{z_1} \cdot f_{n4} \tag{3}$$

Planet-gear rollover frequency

$$f_{nP} = \frac{2 \cdot z_1 \cdot z_3}{z_2 \cdot (z_1 + z_3)} \cdot f_{n1} = 2 \cdot \frac{z_3}{z_2} \cdot f_{n4} \tag{4}$$

Ring-gear rollover frequency

$$f_{nR} = \frac{k \cdot z_3}{(z_1 + z_3)} \cdot f_{n1} = f_{n4} \tag{5}$$

$S$: sun-gear $n_1, z_1$
$P$: planet-gear $n_2, z_2$
$R$: ring-gear $n_3, z_3$
$C$: carrier $n_4$
$k$: number of planet-gear

Tooth meshing vibrations usually appear dominantly. Their causes are diverse. In a fault-free gear, they are mainly caused by tooth deformation under load. Tooth deformation is load-dependent. Consequently, the vibrations are also load-dependent. [6]

#### 2.1.2. Frequencies of rolling bearing

Rolling bearings generally consist of two bearing rings with integrated tracks. Rolling elements are arranged between the rings and roll on the tracks. A cage usually guides the rolling elements, keeps them at a constant distance from each other and prevents the rolling elements from touching each other. This results in rolling friction between the rolling elements and the bearing rings.
The frequency components, which form the typical damage patterns in the vibration signals due to the repeated rolling over of a damaged area, correspond to the kinematic frequencies of the respective bearing resulting from its geometry. [7]

Outer-ring rollover frequency:

$$f_{or} = \frac{1}{2} \cdot f_n \cdot z \cdot \left(1 - \frac{D_w}{D_T} \cdot \cos\alpha\right) \tag{6}$$

Inner-ring rollover frequency

$$f_{ir} = \frac{1}{2} \cdot f_n \cdot z \cdot \left(1 + \frac{D_w}{D_T} \cdot \cos\alpha\right) \tag{7}$$

Rolling-elements rotation frequency

$$f_{bb} = \frac{1}{2} \cdot f_n \cdot z \cdot \left(1 - \left(\frac{D_w}{D_T} \cdot \cos\alpha\right)^2\right) \tag{8}$$

Cage rotation frequency with rotating inner-ring

$$f_{ci} = \frac{1}{2} \cdot f_n \cdot \left(1 - \frac{D_w}{D_T} \cdot \cos\alpha\right) \tag{9}$$

Cage rotation frequency with rotating outer-ring

$$f_{co} = \frac{1}{2} \cdot f_n \cdot \left(1 + \frac{D_w}{D_T} \cdot \cos\alpha\right) \quad (10)$$

$z$: number of rolling elements
$D_W$: rolling element diameter
$D_T$: reference circle diameter
$\alpha$: pressure angle

### 2.1.3. Frequencies of journal bearing

The difference between rolling bearings and journal bearings is in the form of transmission of the movement. Journal bearings transmit the occurring forces by sliding elements. One advantage of plain bearings over rolling bearings is their low-noise operation.
In case of too much bearing tolerance or loose bearing seat, the rotational frequency of the shaft with many harmonics can be seen in the spectra of velocity or acceleration.
Sometimes subharmonic components of the rotational frequency can also occur. There are large differences in the oscillations between the axial and radial measurement directions.
In contrast to rolling bearings, there is no overrolling of defective areas on the tracks in plain bearings, which leads to regular shocks. Therefore, there are no specific overrolling frequencies that can be monitored. However, problems do occur in dependence on the rotational frequency. Characteristic here are vibrations at around 42% to 48% of the rotational frequency, which occur when turbulence forms in the oil film. [7] The causes are mainly overload, abrasive faults or poor bearing adjustment. [8]
Oil film whip is a bearing instability that often occurs in conjunction with whirl. The whirl occurs at the bearing's natural frequency and persists at that frequency as the speed is further increased. Can lead to destruction. [9]
Loose bearing components or unacceptably high bearing clearance are reflected in the increased occurrence of radial vibrations with the rotational frequency.
The basic appearance of bearing instability (whip) occurs mainly in unloaded bearings.

### 2.1.4. Basics of structure-borne sound conduction

Every mechanical structure and therefore also every machine and plant is characterized by its mechanical system properties. In addition to damping and natural modes of vibration, the system properties also include the natural frequencies.
Not all natural frequencies are of interest. Only the natural frequencies that are also stimulated lead to resonant vibrations. Material fatigue, faulty repairs or changes to the structure can result in unwanted stimulation. [1]

In machines with rotating components, the excitation is often caused by a speed-dependent vibration component. Therefore, the speed is divided into the ranges non-critical, critical and supercritical.
Critical speed is defined as the speed at which a vibration component corresponds to a natural frequency of the system and which leads to resonance phenomena and as a result to vibrations with high amplitudes. The degree of this amplitude increase is significantly influenced by the damping. [7]

The measurement of structure-borne sound has the advantage that the vibrations are measured directly at the gearbox housing and influences due to external noise are not measured. The measurement of acceleration forces is a method for the summed assessment of vibration excitation and structure-borne noise path, since the measurement is usually performed at the gearbox housing and the vibrations originate elsewhere. The measurement data is affected by the entire vibration system. Usually, piezoelectric accelerometers are used to convert the mechanical vibrations into electrical signals. The choice of mounting location and the correct mounting technique have a significant influence on the measurement result. The measuring point should be selected in such a way that the transfer path from the sound source to the sensor is short and the material has a high degree of stiffness. Elastic components should not be on this transfer path. In the case of rotating elements, such as in gearboxes, the bearing points are usually well suited as mounting points. Thin-walled elements are not suitable due to their deformation. In gear construction, the force excitation is not directly at the housing structure, but at the gear teeth. Thus, the structure-borne sound behavior of the housing, i.e. the radiating structure, is triggered by the excitation forces acting on the bearings of the shafts. If bearings are not in the line of action of the housing wall, but in protruding structures, excitations of the housing walls due to bending moments occur in addition to the force excitations. [1]

### 2.1.5. Finite-Element-Method with Multi-Body-Simulation

Usually, a simplified model approach with analytical consideration is not sufficient for the assessment of the dynamic operating behavior of planetary gears. The highest level of complexity is provided by the fully three-dimensional simulation model, which can be a finite element model or a multi-body simulation model.
The analytical model is used for this purpose as a basis for further development of a more accurate simulation model.
The finite element method is suitable for describing the simulation model three-dimensionally with the help of multidimensional degrees of freedom.
Resonance frequencies can be determined via the structural mechanical properties stiffness, inertia and damping. Usually, finite element models are very finely discretized. In addition, the dynamics simulation results in a high number of computing steps. This results in a considerable calculation effort. Therefore, only the resonant frequencies of the structure are to be determined in this model with the FEM.
To limit the computational effort, the FEM simulation is combined with a multi-body simulation. In the multi-body simulation, the planetary transmission is represented by several undeformable bodies that are not in focus. In addition, the ability of the bodies to move with respect to each other is restricted by idealized kinematic joints.
Structure-borne noise calculations can be solved in combination in the multi-body simulation by completing the dynamics model with sufficiently accurate forms of representation of the vibrating structure, e.g. gear-shaft-bearing-housing system. For this purpose, discretized FE meshes with sufficient accuracy are suitable. The computational effort for such models may exceed the acceptable limit for the numerical calculation in the time domain, so that a modal representation of the structural dynamics in the form of reduced models has to be performed instead. [1]

## 3. FAULT DETECTION AND ANALYSIS

Depending on the application and objective, various display formats are available for the analysis of vibration data. The most commonly used diagrams are the time signal and the frequency spectrum. These form the basis for further evaluations.
From the time signal, a spectrum can be generated with the help of the Fast Fourier Transformation, in which vibration amplitudes are represented over a frequency axis instead of a time axis.
The waterfall diagram and the spectrogram are suitable for visualizing time- or speed-dependent changes in vibration behaviour or for comparing measurements over a long period of time. Here, the resonance ranges of the gearbox can be identified particularly well.
Most vibration signals are complex signals consisting of a large number of frequency components with different amplitudes. The frequency components are easier to recognize in the spectrum display. [7]
In vibration measurement, the state of motion of a mechanical system is described by the following measurands: vibration displacement, vibration velocity and vibration acceleration.
For all three vibration quantities, a sinusoidal curve is obtained, only out of phase. If a measurand is known, the other two measurands can be calculated from it. In practice, the measurand acceleration has prevailed over the measurand velocity for the detection of machine vibrations. Acceleration covers a wide frequency range with similar amplitude and enables the diagnosis of various problems.
About the signals. It makes most sense to use both formats. The time domain and frequency domain analysis. In the time domain analysis the vibration signals can be judged by optical characteristics. Time signals have regular periodic or random irregular events. The signal deflections can have different amplitude heights in positive and negative. [7]

### 3.1. Fault Detection of Transmission

Uniformly distributed wear of a gear is manifested by increasing amplitudes of the tooth meshing frequency and its harmonic. Errors in the gearing are often manifested by the appearance of sideband components on both sides of the tooth meshing frequency and its harmonic.
The vibration excitation at the tooth meshing is transmitted to the housing via the shaft and the shaft bearing. Due to the forces acting during tooth meshing, the intermeshing teeth undergo elastic deformation, even in the optimum condition. Since this is load-dependent, the resulting vibrations exhibit a strong load dependence. The process of tooth meshing in an intact planetary gear usually shows low amplitudes.
Starting from an intact planetary gear, the sideband components are represented only with low amplitudes in relation to the frequency component of the gear mesh. A lack of lubricant can increase the magnitude of the amplitude and the number of harmonics of the frequency components as the tooth mesh becomes "harder". Similarly, increased tooth backlash also leads to an increase in amplitude. With wear, the amplitudes of the harmonics increase proportionally more than the amplitude of the frequency component.
If a tooth is damaged, it generates a stronger vibration, which results in different amplitude values in the course of one revolution of the gear. This results in amplitude modulation and a significant increase in the sidebands.
For the detection of broken teeth, the evaluation in the time domain is recommended, since this shock excitation can be detected well there. [7]
Eccentrically running gears are detected by means of amplitude modulation, since the tolerance in tooth meshing changes with speed. In addition, the rotational frequency component of the gear is noticeable due to the shifted center of gravity axis.

### 3.2. Fault Detection of rolling bearing

Most damage to rolling bearings starts with small pittings on the surfaces of the tracks of the outer ring, inner ring or rolling element. [7] Discrete faults in ball or roller bearings lead to a sequence of impulsive shock excitations during rotary motion. Each time the fault rolls over, a shock pulse occurs. The pulse frequency can be calculated and assigned from bearing geometry and speed. Because of the impulsive character, the single event will be rich in high frequency components (dominated by the bearing resonances). The components with the repetition frequency itself will have only secondary importance in the spectrum. If the error occurs when rolling over different load zones, the pulse sequence can occur with amplitude modulation. [8]
A defective rolling element rotates at the rotational speed of the cage and generates a few shocks per one rotation of the shaft. Since a defective spot on the rolling element can alternately contact the inner ring and the outer ring, the spectrum shows not the single but the double rolling element rollover frequency as a shock frequency with sidebands of the cage rotation frequency. [7] It is important to note whether the inner ring or outer ring is rotating.
Initially, multiples of the damage frequency show up in the high frequency ranges, possibly with sidebands, typically especially in the range of natural frequencies. The further the damage progresses, the more clearly the corresponding components can be seen in the lower frequency ranges. From this course of damage development and the characteristics of the signals, it follows that bearing damage in the velocity spectrum can only be identified at a significantly later stage than in the acceleration spectrum. Externally braced bearing seats result in a spectrum that can resemble an outer ring damage pattern. Excessive bearing tolerance or loose bearing seating can be seen in the rotational frequency of the shaft with many harmonics. [7]

### 3.3. Fault Detection of journal bearing

In the field of frequency analysis, it is worth taking a look at the spectrum below the rotational frequency to identify a loose bearing seat or an oil film crack in plain bearings.
In contrast to rolling bearings, there is no rolling over of damaged areas in journal bearings, which leads to regular shocks. This also means that there are no specific over rolling frequencies that can be monitored.

### 3.4. Fault detection of unbalanced mass

If a rotating body has a mass distribution that is not rotationally symmetrical, unbalance occurs. In the vibration time signal, this is shown by a sinusoidal curve in the radial direction of measurement. The frequency components correspond to the rotational frequency of the rotating unbalance mass.

## 4. STRATEGY OF SIGNALANALYSIS

In this project, the vibration signals of a planetary gearbox are measured on a drive test bench in order to carry out fault identification. Assembly errors, damage to tooth flanks, defective bearings and also random errors from production are to be detected. In order to understand the vibration measurements, a simulation model is developed as in chapter 2. The signal analysis strategy (FIG. 2) takes advantage of the fact that the effects of different events occurring at different points merge in the housing, ideally only one sensor per gearbox is needed. [10] Information about the vibration behaviour can be measured by means of the vibration displacement, vibration velocities or vibration accelerations. [11]

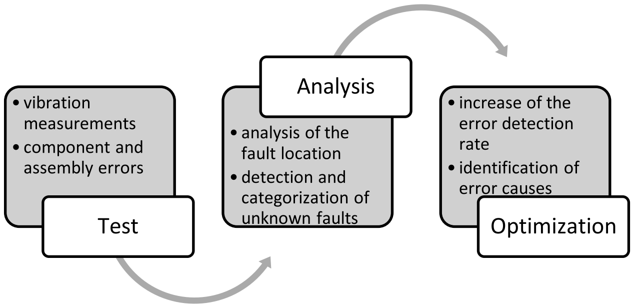


FIG. 2: Strategy of signalanalysis

Vibration displacement, vibration velocity and vibration acceleration are mathematically related. These are dependent on the frequency.

Thus, the amplitude is also frequency-dependent. This results in different dynamics of the signal over the frequency for the vibration displacement, vibration velocity and vibration acceleration as shown in FIG. 3. If a wide frequency spectrum is acquired, the vibration acceleration is often more suitable with the lower dynamics of the amplitude over the frequency. If an optical measurement method is used, it is recommended to use the vibration velocity or vibration displacement, since these are based on the determination of the vibration velocity. Within this project the focus is on the acquisition of the vibration acceleration for the high frequency part and the acquisition of the vibration displacement for the low frequency part.

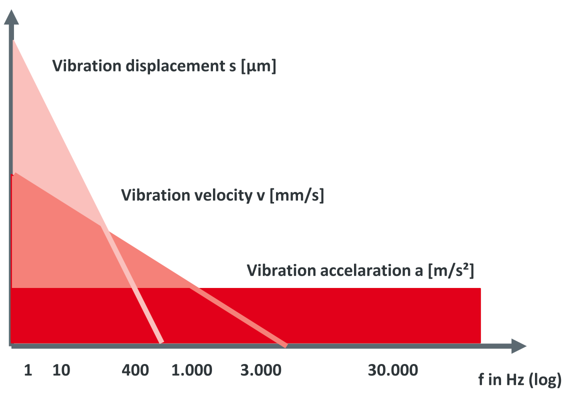


FIG. 3: Comparison amplitude over frequency

### 4.1. Envelope analysis

By means of envelope analysis, hidden pulse sequences of shocks can be detected in a vibration signal. This is particularly interesting for rolling bearing diagnostics, especially if the excitations originate from different bearings and further sources of structure-borne noise are present, as in a planetary gear. [11]
The measured time signal from the tests is demodulated by bandpass filtering before rectification of the relevant carrier frequencies. Low frequency components of the time signal are thereby suppressed. The signal now consists only of the modulation signal, the envelope. A subsequent fast fourier transformation of the envelope results in the envelope spectrum. The amplitude is a measure of the extent of the damage. The advantages of envelope anlysis are a direct display of the error frequency and insensitivity to speed fluctuations. [8]

### 4.2. Order analysis

Order analysis is a proven method for performing frequency-selective investigations on machines or drives with variable speed. Here, a time signal is provided proportional to the angle of rotation. The resulting spectrum is called order spectrum. The reference of the angle of rotation is mostly done on the faster shaft. The rotational frequency of this wave corresponds to the first order. The harmonics form the second, third order. The dependencies of the tooth meshing frequency and the rolling bearing peaks are no longer influenced by the rotational speed. [12]

Basically, two methods are available:
1. Data acquisition is triggered externally with an incremental encoder that provides a constant number of pulses per revolution of the shaft.
2. Data acquisition is time-controlled and the speed is recorded. Subsequently, the computational processing of the time signal takes place. [12]
Combined order analysis and resonance monitoring can detect various forms of damage and vibration. These include shaft cracks, resonance peaks, oil film instabilities and journal bearing defects. Resonance monitoring can detect bearing anomalies, loose shaft couplings or altered structural stiffness. [11]

### 4.3. Run-up and run-down analysis

In contrast to measurements in stationary operation, run-up and run-down measurements offer valuable additional information. However, these require advanced measurement and evaluation technology. Through the variable speed analysis, not only a single spectrum is examined, but a whole field of spectra. Of interest here are the critical speeds at which resonance overshoot occurs. These give an indication of the state of stiffness of the structure. For example by a shift of the resonance frequency. Here, the error mainly affects the resonance frequency and less the maximum amplitude. [8]
For the investigation of the spectra field different representations are suitable: Cascade diagram/waterfall display, spectrogram or also Campbell diagram.
The representation in the Campbell diagram offers the possibility to interpret forced oscillations or structural resonances.

## 5. CHALLENGES

Due to the more complex kinematics of planetary transmissions, the analysis of vibrations is complex. This is caused by a wide variety of vibration sources. On the one hand by the rotating axles and rotating planets. This results in a signal propagation time of the vibration from the excited tooth mesh planet-ring gear to the collecting sensor with amplitude modulation. [13] This makes the identification of faults in planetary gearboxes more complexity. [4]

## 6. OPORTUNITIES

The theoretical approaches presented in this paper will be investigated in the near future in measurements on a planetary gearboxes. By integrating component data from the digitalized production process, it will be possible to predict the operating performance of planetary gears in the future. By integrating the specific component data from the production process, no test samples are needed to generate databases with comparative measurements. Furthermore, it is possible to introduce a self-learning process to replace fixed limits by self-learning limits. This can be advantageous for the consideration of production tolerances. Here, there are interesting approaches for the consideration of production tolerances. Here, there are interesting approaches for the design of fully automatic diagnostic algorithms.

## 7. CONCLUSION

In this project, a test strategy was designed to test planetary gearboxes on a test rig directly after production. The focus is on structure-borne noise measurements. The different mechanical phenomena in planetary gears have different effects on the measurable vibration signal. Exact knowledge of the signal origin as well as knowledge of the signal analysis are the basis for a solid evaluation and interpretation of the measurement results.
Therefore, the developed test strategy for planetary transmissions is based on a model development in which the planetary transmission is modeled with analytical and finite element methods with the addition of multi-body systems. To get a better understanding of the measured vibrations. This is intended to identify faults in the planetary gearbox on the test bench without having to measure a large number of intact and defective planetary gearboxes on the test bench.
For variable speed drives, order analysis often provides the means to investigate and accurately identify vibrations. Speed-up and speed-down processes provide information on natural oscillations and resonance states. By applying envelope analysis at steady-state operating points, a variety of defects in planetary gears can be detected and a precise statement of condition can be made.
The theoretical approaches will be followed and validated in the further process of the project.

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