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Methodical Evaluation of Sensor Positions for Condition Monitoring of Gears

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Abstract (300-500 words)

With the increasing digitalization of mechanical engineering and the development of mechanical systems into cyber-physical systems, predictive maintenance and condition monitoring have become more widespread in technical applications. However, the fault diagnosis and especially the prognosis of remaining useful lifetime is still not accurate in many applications.

This paper presents a methodical approach for evaluating and choosing measurands and sensor positions in order to improve condition monitoring systems by enhancing the quality of the measurement data. The underlying concept of sensor-based condition monitoring systems is that faults affect measurable properties of the machine, enabling the deduction of the fault status by analyzing the measured data. Most current systems use quantities that are easy to measure, like vibration at the machine housing, and successively apply advanced data processing methods for diagnosis and prognosis. For many failure mechanisms, these commonly used measurands are too far away from the actual fault location, e.g. a tooth root crack in a spur gear transmission. This leads to a loss of information between the fault location and the sensor: The property of interest (depth of the tooth root crack) is transformed into a variation of the contact force and mechanical vibration. Thus, the vibration can be seen as a mechanical signal, which is masked by disturbances, attenuated and modified along the transmission path from its source to the sensor. Therefore, the information about the fault, which is conveyed by the vibration signal, is more accurate when the sensor is positioned close to the fault location. The consequential problem for the developer of condition monitoring systems is to choose measurands and sensor positions in order to maximize the quality of the gathered data.

A methodical approach for evaluating measurands and sensor positions is presented and explained in this paper using the example of gear condition monitoring. First, possible measurands and sensor positions are explained. Then, sensor positions for vibration-based condition monitoring are evaluated by estimating and measuring the vibration transfer function between the fault location (gear tooth) to various sensor positions, both on the shaft and on the

housing. The results show that the bearings, due to their compliance, isolate vibration and thus decrease the information content of vibration measurements at the housing, compared to measurement on the shaft. Therefore, it is proposed to develop new sensor concepts that allow cyber-physical systems to reach deep into machines and gather relevant information closer to the actual point of interest, in this case the fault location.

Keywords: ***Big Data/ Digital Design, Cyber-Physical Systems, Condition Monitoring, Predictive Maintenance, Sensor Integration, Tooth Root Crack, Bearings***

1 Introduction

Condition monitoring and predictive maintenance are one of the most important use cases of Industry 4.0 (Anderl 2014), and one of the most-discussed topics in the context of Industry 4.0, the Industrial Internet of Things and the digitalization of mechanical engineering in general.
More specifically, condition monitoring of gears has been an object of research for a long time and vibration-based techniques are common industrial practice (Randall, 2010). A lot of research is focused on signal processing and the application of machine learning techniques, i.e. *extracting* information about the current fault status from sensor data. The amount of research conducted in this field is demonstrated by a still often-cited review (Jardine, Lin, & Banjevic, 2006), covering 271 papers. In comparison, the aspect of *acquiring* meaningful sensor data has received less attention. This question – what sensor data would be best suited for condition monitoring – has three aspects. Firstly, *what* to measure: Choosing the measurand (vibration, oil analysis, inverter current etc.); secondly, *where* to measure: Choosing the sensor position that yields the best data quality; thirdly, choosing the sensor principle (and the actual sensor model), e.g. choosing between piezoelectric and MEMS accelerometers. While this third aspect is interesting with respect, among others, to problems of packaging, this paper only deals with the first two aspects. As suggested by (Martin, Schork, Vogel, & Kirchner, 2018), the integration of sensors close to the location of interest offers potential for reducing uncertainties and disturbances, and thereby to acquire meaningful data as a basis for data analysis.
We use the example of monitoring tooth root cracks in spur gear transmissions, such as the one shown in Figure 1. These transmissions are used e.g. in production lines, where an early detection of incipient cracks and an accurate prediction of remaining useful lifetime is necessary in order to plan the necessary maintenance actions.

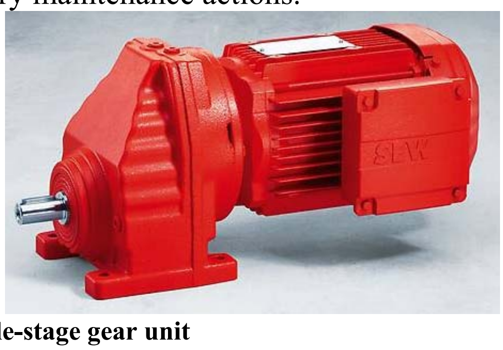

Figure 1. Gearmotor with single-stage gear unit

Section 2 of this paper deals with a general discussion of the information transmission from a fault to a sensor and its processing in a condition monitoring system. Some general requirements for evaluating and choosing measurands and sensor positions are developed.
In section 3, these general requirements are applied to the problem of monitoring tooth root cracks in spur gear transmissions, and more specific evaluation criteria for the evaluation of vibration sensor positions are derived. Finally, sensor positions on the shaft and the housing are evaluated in section 4 by modeling and measuring the vibration transfer functions.

2 General requirements for evaluating measurands and sensor positions for condition monitoring

The terms *condition monitoring* and *predictive maintenance* are used in literature with somewhat overlapping meaning. Therefore, a clear definition is difficult to give. The understanding of the authors of this paper is that a *condition monitoring system* considers the current situation, while a *predictive maintenance system* predicts the remaining useful lifetime in order to guide maintenance scheduling. This understanding is described briefly below:

The function of a *condition monitoring system* is to estimate the current fault status (Figure 2, top). One aspect of this is classifying a system into the categories faulty and non-faulty. In the example of a tooth root crack, "faulty" means that there is an incipient crack at the tooth root. In addition to this binary classification, it is also desirable to assess fault severity, for example the depth of a tooth root crack and especially to track the growth of the crack.

The function of a *predictive maintenance system* is to estimate the remaining useful life (RUL) of a system or a component (Figure 2, bottom). One approach to predictive maintenance is to monitor the load on a component and then estimate the remaining useful life using a load-dependent model, e.g. Miner's rule, as in (Foulard, 2015). One disadvantage of these approaches is that the statistical variation of the strength between individual parts cannot be taken into account. Other approaches consider the period after the incipient fault occurs and estimate the fault propagation until breakdown of the machine (Jardine et al., 2006). Just like in condition monitoring, these methods use sensor data related to the fault to estimate the current fault status and then predict remaining useful life using a fault propagation model and – at least implicit – assumptions about the future load profile.

Summing up, the basis of both condition monitoring and predictive maintenance systems is an estimation of the current fault status, based on sensor data.

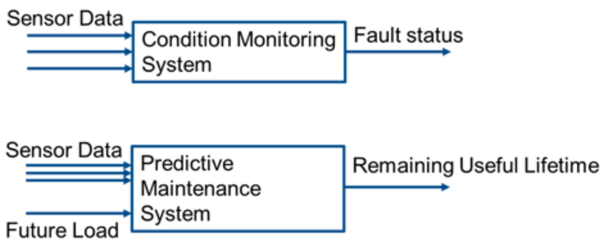

Figure 2. Function of a condition monitoring system and a predictive maintenance system

The underlying concept, illustrated in Figure 3, is that a fault is related to measurable properties of a machine through physical effects. The physical effects can be influenced by disturbances, as shown in Figure 3 for physical effect 1. The measurable properties are the measurands which are acquired by the sensors. In general, a physical effect will influence more than one measurand, and a measurand will depend on more than one physical effect, symbolized in Figure 3 by the connection between physical effect 2 and sensor 3. The actual measurement of the measurands by the sensors is subject to a measurement error, illustrated in Figure 3 for sensor 1. The sensor data is fed into the condition monitoring system, which incorporates a model relating the measured sensor data to the fault status. The task of the product developer is to choose the inputs to the system (measurands, sensor positions and sensors) and develop the model that estimates the fault status. In current practice, developing the model and choosing the inputs is a rather intuitive and experience-based process. In this section, some general requirements that can be used for choosing measurands and sensor positions methodically, and thereby to support the developer in the process of choosing the inputs, are presented.

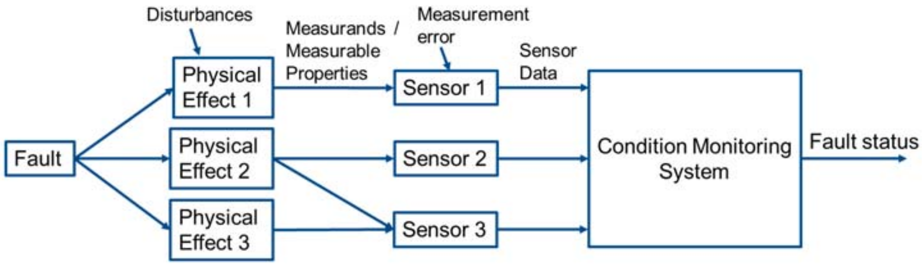

Figure 3. Model for the relationship between a fault and the fault status estimated by a condition monitoring system

Developing the model that relates the sensor data to the fault status can be seen as modeling the physical effects that relate the fault to the measurands. Judging by the literature cited in a comprehensive review (Jardine et al., 2006), the majority of approaches does not explicitly model the physical effects, but uses advanced data processing and analysis tools. In this sense, the model is an algorithm.

The measurement error and the model uncertainty can cause errors in the estimation of the fault status, and their reduction can therefore be an approach to increase the performance of condition monitoring systems. The model uncertainty can be reduced by including more effects in the model, using more advanced data processing tools, adapting the model to the specific situation or using more input variables (i.e. sensors). However, these approaches are costly and limited, because some effects, for example in tribology, show a behavior that is difficult to predict and depends strongly on disturbances, for example on small variations of the geometry caused by tolerances. Such effects are very difficult to incorporate in a model. Therefore, it seems promising to use measurands whose relation to the fault can be determined and is robust towards disturbances.

Summing up, two general requirements for choosing measurands and sensor positions can be formulated: a) the signal-to-noise-ratio (SNR) of the fault-related measurand should be high, and b) the physical effects relating the fault status to the measurands should be determinable and robust towards disturbances in order to reduce model uncertainty.

3 Example: Evaluation criteria for measurands and sensor positions for condition monitoring of spur gears

In this section, the general ideas presented in section 2 are applied to the example of monitoring tooth root cracks in gear transmissions. Firstly, measurands found in literature are integrated into the presented model. Then, for the specific case of vibration measurement, specific criteria are derived from the general requirements formulated in section 2.

3.1 Measurands for condition monitoring of tooth root cracks

In the extensive literature about tooth root crack monitoring, many different types of measurands are used, e.g. Acoustic Emission (Singh, Houser, & Vijayakar, 1999). In this paper, we focus on three measurands that are associated with the tooth stiffness reduction caused by the tooth root crack: Vibration, bearing load and shaft rotation angle. First, the relation of these measurands to the tooth root crack is explained. Then, their respective advantages and disadvantages with respect to the general requirements formulated in section 2 are discussed

Let us assume that the fault status of a gear with respect to a tooth root crack can be characterized by one parameter, the depth of the crack *d* (Figure 3, left). Under this assumption, the function of the condition monitoring system is to estimate this target quantity, *d*. Figure 3 shows the effects relating the crack to the measurands vibration, bearing impedance and shaft rotation angle. Below, the effects are explained in more detail. The explanations about parametric excitation, vibration and transmission error are based on (Randall, 2010) .

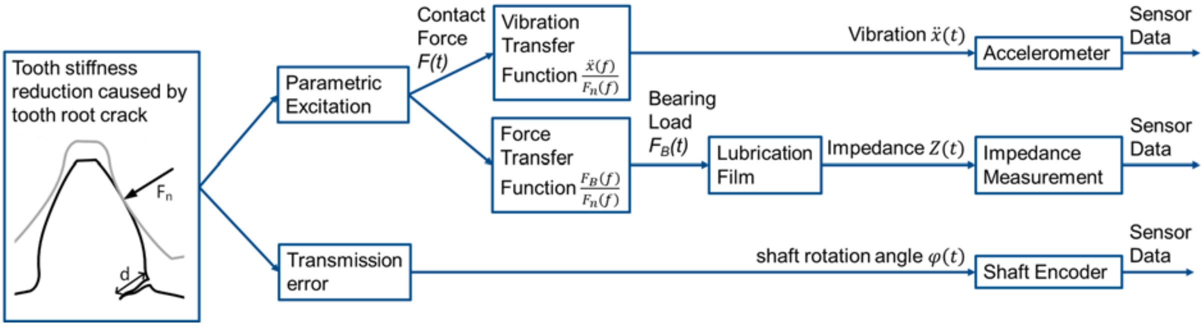

Figure 4. Relationships between a tooth root crack and measurands used in literature for condition monitoring

An inherent characteristic of a gear transmission's working principle is the periodical fluctuation of the force transmitted between the meshing gears. It fluctuates because the stiffness of the meshing gears depends on the number of teeth that are in contact at a given time. This effect is called "parametric excitation" and the corresponding excitation frequencies are the gearmesh frequency $f_Z = f_n \cdot z$ and its harmonics ($2 \cdot f_Z$, $3 \cdot f_Z$ etc.), with f_n being the rotational speed and z being the number of teeth of the gear. A schematic representation of this force fluctuation in the time and frequency domain is shown in Figure 5 (top).
If a tooth root crack is present, the bending stiffness of that damaged tooth is reduced, due to the reduced cross section. Once per revolution of the gear, when the tooth with reduced stiffness meshes with the opposite gear, the force fluctuation is larger than in a healthy tooth (Figure 5, bottom left). In the frequency domain (Figure 5, bottom right), this causes addition excitation frequency components called "sidebands", whose spacing is equal to the rotational frequency f_n. The presence of sidebands in the force spectrum or the vibration spectrum can be used to diagnose tooth root cracks. It must be noted that the sidebands can also be caused by other faults, e.g. by pittings in the gear flanks (Randall, 2010). The point here is that the sidebands contain information about the fault.

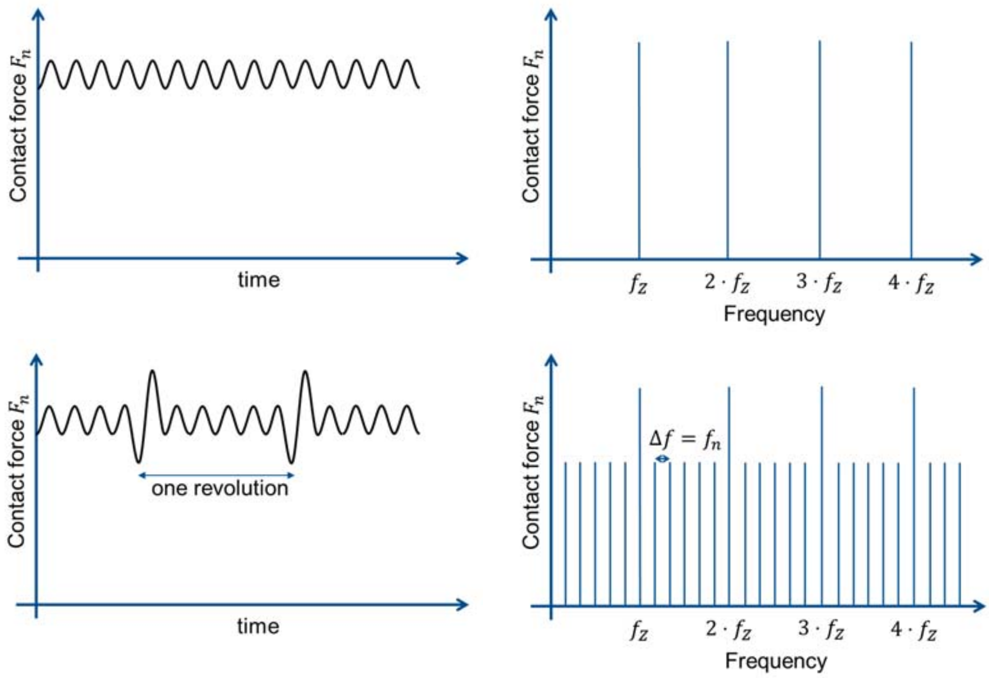

Figure 5. Schematic representation of the contact force of a healthy gear (top) and a gear with a tooth root crack (bottom), in time and frequency domain

The first measurand described in Figure 4 is vibration. The contact force excites vibration in the entire structure of the machine (e.g. on the shaft and housing of the gearbox), which can be measured by using displacement sensors or accelerometers. The relationship between the exciting force and the resulting vibration (e.g. acceleration $\ddot{x}$) is described in the frequency domain by the vibration transfer function $\ddot{x}(f)/F(f)$. An example of a transfer function and the vibration spectrum resulting from the excitation with sidebands can be seen in Figure 6.

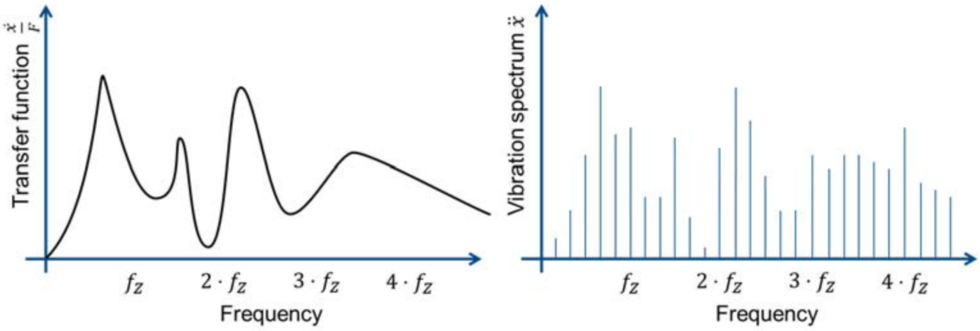

Figure 6. Schematic representation of a transfer function (left) and the vibration response resulting from an excitation spectrum containing sidebands (right)

The second measurand described in Figure 4 is the bearing load measurement using the bearing impedance: The fluctuating contact force is transmitted from the gearmesh to the bearings and changes the bearing load. In practice, the bearing load is difficult to measure, because no discrete force sensor can be integrated at the bearings without major changes to the mechanical design of the gearbox. (Schirra, Martin, Vogel, & Kirchner, 2018) suggested to measure the bearing load by measuring the electrical impedance of the lubrication film, which is influenced by the bearing load, and use it to monitor gearboxes.

The third measurand related to the reduced tooth stiffness, described at the bottom of Figure 4, is the transmission error: A change in tooth stiffness causes an increased deformation of the tooth when it is loaded by the contact force. In consequence, the driven gear‘s rotation does not follow the driving gear exactly, resulting in a „transmission error“. It can be measured by shaft encoders, which measure the shaft rotation angle $\varphi(t)$, and used to diagnose tooth root cracks (Randall, 2010).

Following this description of the relationships, their advantages and disadvantages with respect to the general requirements formulated in section 2 can be discussed: It is difficult to compare the measurands with respect to the first requirement – high signal-to-noise ratio –without detailed knowledge about the disturbances acting on the respective sensors. For the more specific case of comparing different vibration sensor positions, the evaluation with respect to this criterion is conducted in sections 3.2 and 4. For the second requirement – robustness of the physical effects – a comparison of different measurands is possible by a literature study along the signal propagation paths in Figure 4: Both the *transmission error* at the gearmesh and the *parametric excitation* are influenced by the load, by other faults like pittings or production tolerance, and by the internal dynamics (in particular torsional resonances) of the gearbox (Randall, 2010). From the gearmesh, the transmission error is “transferred” directly to the encoder sensor positions at the free (unloaded) ends of the shaft, where it can be measured very accurately (Randall, 2010). Therefore no additional uncertainties arise for this measurand. The force and vibration *transfer functions* are subject to disturbances. They depend in particular on the dynamic stiffness and damping of the bearings, which have a high variance.

Further research is necessary to develop more specific, quantifiable criteria for evaluating the robustness of measurands. Then, the qualitative uncertainty analysis given above needs to be refined in order to compare the measurands with respect to the specific criteria. For vibration-based condition monitoring, specific criteria are derived in the next section.

3.2 Criteria for evaluating vibration sensor positions

Vibration is the most widely used measurand for gear condition monitoring (Randall, 2010). Therefore, we will take a closer look at this example and derive more specific criteria to evaluate vibration sensor positions.
First, the requirement "robust behavior towards disturbances" will be discussed. The vibration transfer function from meshing teeth through the gearbox to the housing has been a research object for some time in the area of machine acoustics. Especially the transfer characteristics of rolling and journal bearings was researched. In general, the transfer characteristics of rolling and journal bearings depend on the operating conditions, in particular on load, speed and temperature, both in theory and in experimental investigations. However, no theoretical model which can accurately predict the transfer characteristics of rolling bearings is available, as can be seen from the literature study and the comparison with experimental results in (Kruk, 2010). Research for rolling bearings of the type 6212 has shown that even for one type of bearing, the transfer function differs for different manufactures significantly, probably due to slight differences in the geometry (Kruk, 2010). This shows that the transfer function depends not only on operating conditions, but on other influences that are difficult to control and lead to uncertainty. The consequences of such uncontrolled influences are shown schematically in Figure 7, using the example of two different operating conditions : If the transfer function from the gearmesh to a sensor position differs between two operating conditions (Figure 7, top right), the same excitation spectrum (the same crack at the same load and speed), which is represented in Figure 7, top right, can yield different vibration spectra at the points of measurements (Figure 7, bottom). This makes it difficult for an algorithm to discern whether a change in the spectrum is caused by a change in the excitation (e.g. a growth of the crack) or a change in the transfer function. With respect to the general requirements from section 2, this means that the transfer function should be robust towards operating conditions and other disturbances. More specifically, the measurements should be taken in places (and frequency ranges) whose transfer function is not influenced much by the bearings.

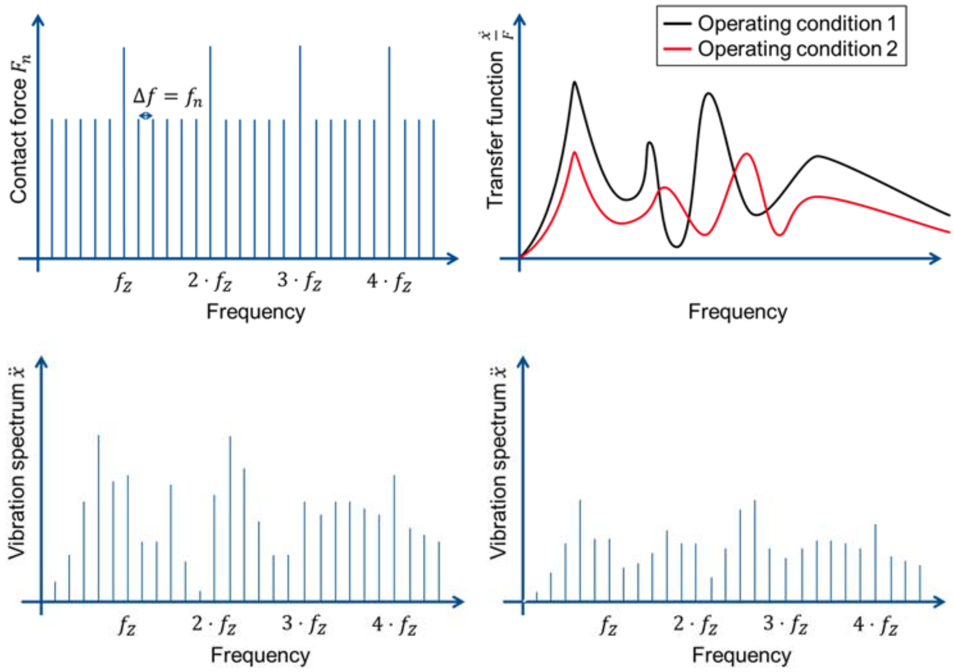

Figure 7. Schematic representation of excitation spectrum (top left), transfer functions at different operating conditions (top right) and resulting vibration response spectra (bottom left: operating condition 1; bottom right: operating condition 2)

Now, the second requirement from section 2 “high signal-to-noise-ratio” will be applied to vibration-based tooth crack monitoring. In order to detect a fault, the fault-related vibration has to “stand out” from the general background noise which is caused by other disturbing vibration sources, e.g. from other machines. Comparing two sensor positions and assuming they have different transfer functions (Figure 8 top), the vibration spectra shown in Figure 8 at the bottom result. Assuming that the noise level at both sensor positions is the same, it can be seen that only some of the characteristic sidebands are obscured by the noise and most sidebands stand out clearly from the noise at sensor position 1. At sensor position 2, almost all sidebands are obscured. More generally, a higher amplitude of the transfer function from the fault location to a sensor location means that a signal originating from the fault has a higher amplitude at the sensor location. With respect to the requirement “high signal-to-noise ratio”, this means that the transfer function from the gearmesh to the sensor position should have a high amplitude in the frequency range of interest. Obviously, the actual noise level at the sensor position has to be considered in this evaluation if possible, but this usually necessitates field measurements or experience from similar cases, which are not always available.

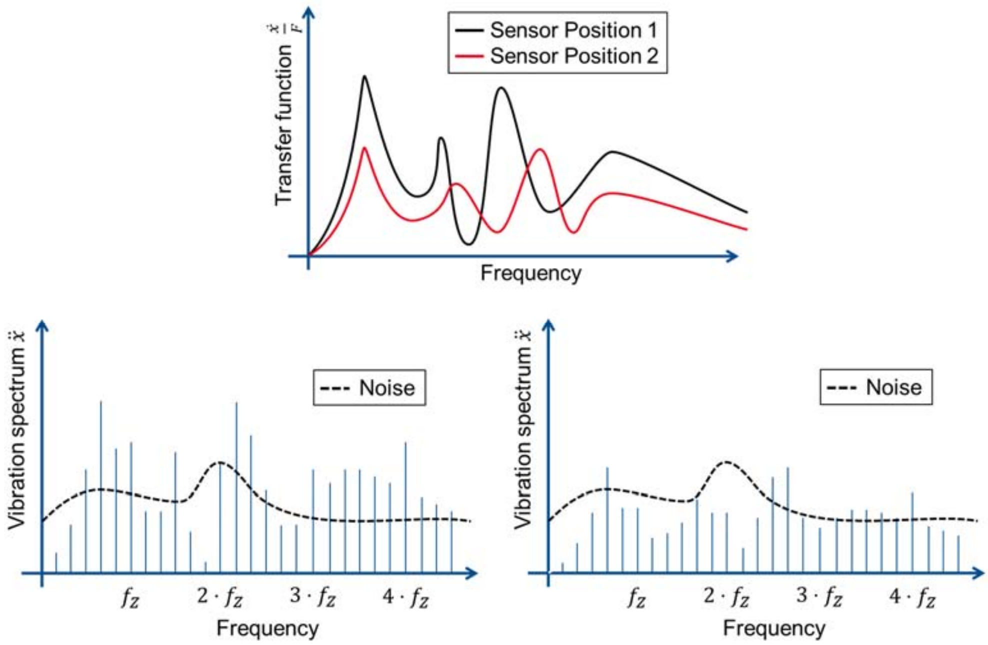

Figure 8. Schematic representation of the transfer function to two sensor positions (top), and resulting vibration at sensor position 1 (left) and 2 (right), including noise

4 Theoretical model and measurement of vibration transfer functions

The discussions in section 3 resulted in the conclusion that the vibration transfer function to the sensor positions should not be influenced by the bearings and have a high amplitude. In order to evaluate this criterion the transfer functions for the driven shaft of a single-stage gearbox (SEW RX57) were modelled using a simple two-degree-of-freedom model, shown in Figure 9, and then measured.

4.1 Theoretical model

The driven shaft is modeled as a rigid body (mass $m = 0.494\,kg$, moment of inertia $\Theta^S = 7.46\,kg \cdot m^2$). The bearings are modeled with stiffness and damping ($k_A = 7.33 \cdot 10^7\,N/m$; $k_B = 1.18 \cdot 10^8\,N/m$; $b_A = b_B = 1{,}000\,Ns/m$) at the distances

$a_A = 9.7\ mm$ and $a_B = 53.3\ mm$ from the center of mass. The shaft is loaded by a force F, acting at the distance $a_F = 68.8\ mm$ from the center of mass both for the calculation of bearing preload and for the calculation of the transfer function. The values of mass, inertia and the geometrical distances are taken from measurements at the actual shaft of the gearbox, which is shown in figure Figure 11. The stiffness of the bearings is calculated according to (Dahlke, 1994) for the respective bearing types. Bearing A is a ball bearing type 6305, bearing B is a ball bearing type 6304. A preload of $F = 1{,}290\ N$ was assumed, corresponding to the static gear contact force acting at the maximum rated torque of the gearbox. The damping value is based on the literature study and the measurements in (Backhaus, 2008), stating a range of the damping between $500 Ns/m$ and $3600 Ns/m$. As mentioned in chapter 3, the stiffness and damping of bearings have a high degree of uncertainty, so the model can only be taken as a rough estimation of the actual behavior.

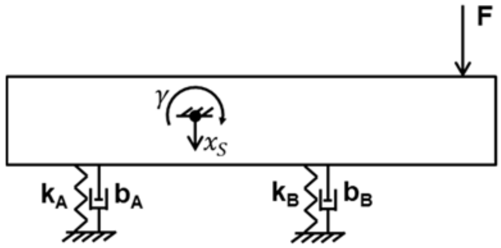

Figure 9. 2DOF model of the driven shaft of a single-stage gearbox

Figure 10 shows the magnitude of the acceleration transfer function to the center of mass $|\ddot{x}/F|$ and of the force transfer functions$|F_A/F\ |$ and $|F_B/F|$. F_A and F_B are the forces transmitted through the stiffness and damping to the foundation: $F_A = F_{kA} + F_{bA}$ and $F_B = F_{kB} + F_{bB}$. It can be seen that the behavior is dominated at lower frequencies by the two resonance frequencies at approximately 1800 Hz and 4200 Hz. At higher frequencies, the magnitude of the acceleration transfer function converges to a constant value because the behavior is dominated by the inertial mass, and the bearing stiffness and damping have little influence. The magnitude of the force transfer functions clearly shows vibration isolation: At higher frequencies, the force transmitted through the bearings decreases, meaning that the vibration excited in the surrounding structure would also decrease. Applied to the evaluation criteria formulated in section 3, the following conclusions can be drawn: At low frequencies, the transfer functions are strongly influenced by the stiffness and bearing of the damping, which are difficult to determine and subject to disturbances as explained in section 3. Therefore, neither the measurement at the housing nor at the shaft is robust towards disturbances at low frequencies. At high frequencies, the transfer functions depend less on the bearing properties. However, very little vibration is transmitted to the housing due to the vibration isolation effect. Therefore, the characteristic sidebands can be easily masked by noise and a low signal-to-noise can result. On the other hand, the transfer function for measurement on the shaft has a high amplitude at high frequencies, indicating a high signal-to-noise ratio. Summing up, the theoretical investigations suggest that a sensor mounted on the shaft would be better suited for condition monitoring of tooth root cracks than a sensor mounted at the housing. It should be noted that vibration isolation also works in the other direction: Disturbing high frequency vibrations from the surroundings, which contribute to the background noise level, are isolated from the shaft, contributing even more to a better signal-to-noise ratio.

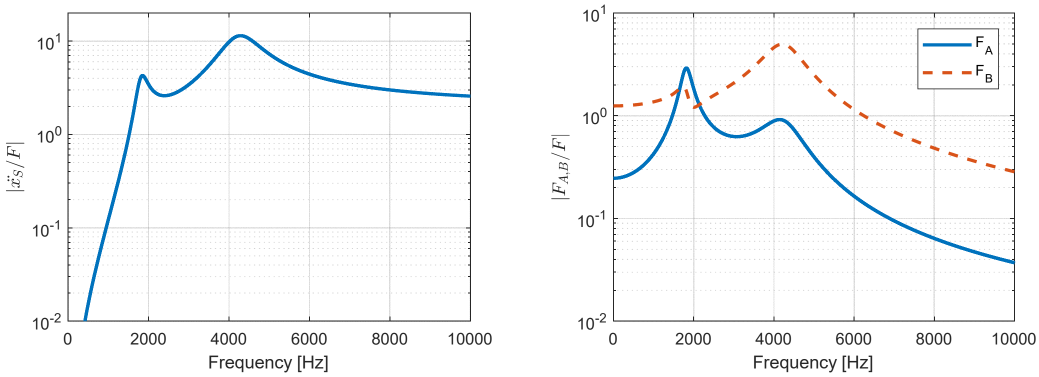

Figure 10. Acceleration transfer function (left) and force transfer functions (right), based on the theoretical model

4.2 Experimental investigation

For the experimental investigation of the transfer function, the shaft was mounted in its original position in the housing. A static load of $F_G = 1000N$ was applied at the end of the shaft in order to preload the bearings, simulating the static gear contact force. Due to reasons of accessibility it was not feasible to apply the preload at the actual position of the gearwheel. The shaft was excited at frequencies up to 5000 Hz by an electrodynamic shaker, acting in the same plane as the preload. Piezoelectric accelerometers were mounted on the shaft and housing at the positions specified in Figure 11. A sketch of the experimental setup can be seen in Figure 11. The shaft was stationary while the measurements were conducted. Since the bearing stiffness and damping depend on the speed – e.g. no lubrication film is present between balls and raceways at stationary conditions – the transfer function at realistic speed will be different. However, the setup is considered suitable for a qualitative validation of the theoretical results.

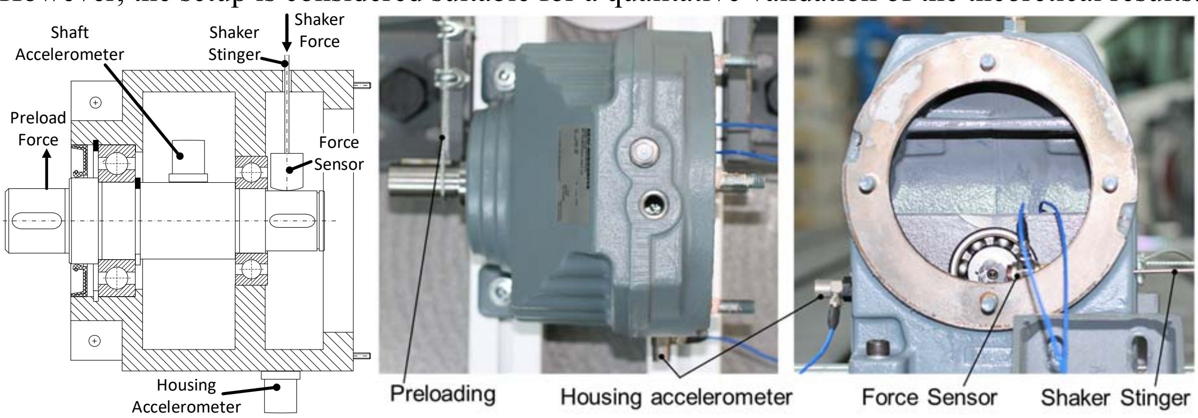

Figure 11. Sketch and photographs of the experimental setup including sensor positions

Figure 12 (left) shows the vibration transfer function for the sensor positions. It can be seen that the amplitude of the transfer function is similar for the two sensor positions frequencies below approximately 1000 Hz, and that the amplitude is higher at the shaft sensor for frequencies above approximately 1500 Hz. This corresponds to the results from the theoretical model that less vibration is transmitted from the shaft to the housing at higher frequencies.

Figure 12 (right) shows the coherence between the excitation force signal and the acceleration response signals. The coherence is a measure of the measurement error, in particular the linearity, of the relationship between excitation and response (Randall, 2010). A coherence of 1 means that the relation between force and acceleration is exactly linear. It can be seen that the coherence is higher for the shaft sensor at nearly all frequencies, compared to the housing sensor. This can be attributed to the nonlinear properties, especially the nonlinear stiffness, of the bearings. In particular, the coherence for the shaft sensor is greater than 0.8 between approximately 400 Hz and 4000 Hz, indicating a low uncertainty of the transfer function.

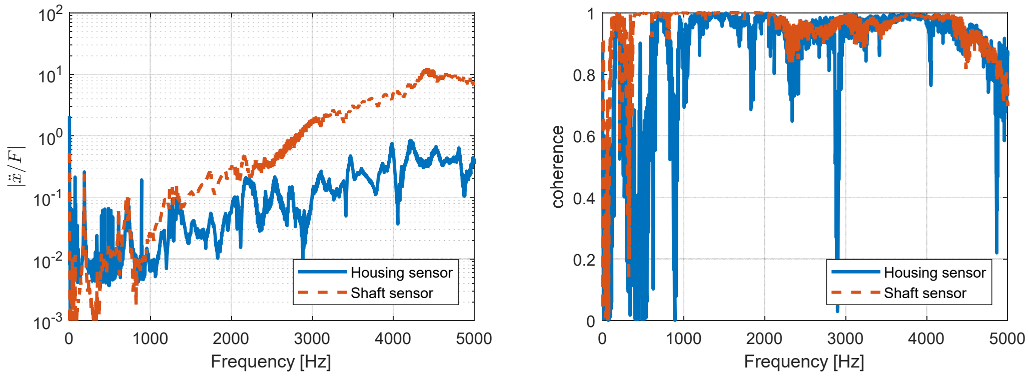

Figure 12. Acceleration transfer function (left) and coherence (right) for sensor positions on the shaft and housing

Summing up, the measurements indicate a more linear transfer function with lower uncertainty, and a higher signal-to-noise ratio due to the higher amplitude of the transfer function, for a sensor placed at the shaft in comparison with a sensor placed at the housing. The quantitative values of the effects in real world conditions are subject to many influences that could not be examined within the scope of this paper, like rotational speed, load, temperature and the scaling of the system. However, the qualitative behavior, which is expected in all conditions, could be verified.

5 Conclusion and Outlook

This paper dealt with the questions of choosing a measurand for vibration monitoring of gears and of choosing the sensor position. General requirements for choosing measurands and sensor positions were presented in section 2, showing that the sensor data used for condition monitoring should have a high signal-to-noise ratio with respect to the fault-related signal, and that the physical effects relating the measurand to the fault should be robust towards disturbances. For the first question – choosing a measurand – the disturbances acting on common measurands were analysed qualitatively in section 3.1, and significant differences were found, but no final conclusions could be drawn. A more detailed analysis in the future should incorporate established methods for considering disturbances, such as proposed by (Freund, Würtenberger, Calmano, Hesse, & Kloberdanz, 2014).
The second question – choosing the sensor position – was investigated more specifically for vibration-based monitoring of tooth root cracks in section 3.2. The transfer function from the toothmesh to the sensor position should have a high amplitude and be independent of the bearing transfer characteristics in order to acquire high-quality sensor data. This criterion can be used to compare different sensor positions and was applied to an industrial gearbox in section 4. The theoretical and experimental investigations in section 4 suggest that high-quality sensor data can be expected if the sensor is placed on the shaft and a sufficiently high frequency range is considered. The reason for this is the vibration isolation at high frequencies and the nonlinear, disturbance-sensitive behavior of the bearings. The quantitative results of the model and the measurement cannot be generalized to other operating conditions and gearbox designs. It is expected, however, that the general effects of vibration isolation and nonlinear bearing transfer behavior are present in all conditions and types of gearboxes. Further research is planned to clarify this generalization. To this end, transfer functions should be measured in a rotating system, including different load and speed configurations, in order to quantify the influence of

operating conditions. Finally, a condition monitoring system should be implemented for promising sensor positions. Based on the preliminary results, we propose to develop solutions that allow the integration of vibration sensors close to fault locations, especially in rotating systems.

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