Forsch Ingenieurwes
https://doi.org/10.1007/s10010-023-00636-z

ORIGINALARBEITEN/ORIGINALS

Comparison of transfer path characteristics for different wind turbine drivetrain variations

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Received: 7 November 2022 / Accepted: 19 January 2023
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Abstract
To reduce acoustic emissions of a wind turbine (WT), the source of the vibrations (e.g. blades, gearbox), the emitting surfaces (e.g. blades, tower, nacelle cover) and the transmission between the source and surface have to be studied. The focus of this paper lies on a method to identify relevant transfer paths between the WT drivetrain and the sound-emitting surfaces and their respective contribution. The identified transfer paths can be used to improve turbine acoustics, especially by identifying problematic transmission areas.
This paper will cover the application of transfer path analysis methods for different wind turbine drivetrains. The method will be applied to an MBS model of a 3 MW turbine. Different Bearing arrangements of the drivetrain and their influence on the Transfer path are compared. With the result of the TPA, developers will be able to determine critical contributions to the tonal behavior of the turbine and optimize either the transfer characteristics or the excitation specifically responsible for tonal behavior. The methods described can be applied to field measurement as well as an MBS-Model of the turbine to be able to optimize transfer characteristics during development.

Vergleich der Transferpfade für verschiedene Antriebsstränge in Windenergieanlagen

Zusammenfassung
Um die Schallemissionen einer Windenergieanlage (WEA) zu reduzieren, müssen die Quelle der Schwingungen (z. B. Blätter, Getriebe), die emittierenden Flächen (z. B. Blätter, Turm, Gondelabdeckung) und die Übertragung zwischen Quelle und Fläche untersucht werden. Der Schwerpunkt dieser Arbeit liegt auf einer Methode zur Identifizierung relevanter Übertragungspfade zwischen dem WEA-Antriebsstrang zu den schallabstrahlenden Flächen. Die identifizierten Übertragungswege können zur Verbesserung der Turbinenakustik genutzt werden, insbesondere durch die Identifizierung problematischer Übertragungsbereiche.
In diesem Beitrag wird die Anwendung von Methoden zur Analyse von Übertragungswegen für verschiedene Antriebsstränge von Windkraftanlagen behandelt. Die Methode wird auf ein MKS-Modell einer 3-MW-Anlage angewandt. Es werden verschiedene Lageranordnungen des Antriebsstrangs und deren Einfluss auf den Übertragungsweg verglichen. Mit dem Ergebnis der TPA können die Entwickler die kritischen Beiträge zum tonalen Verhalten der Turbine bestimmen und entweder die Übertragungseigenschaften oder die für das tonale Verhalten verantwortliche Erregung optimieren. Die beschriebenen Methoden können sowohl auf Feldmessungen als auch auf ein MKS-Modell der Turbine angewendet werden, um die Übertragungseigenschaften während der Entwicklung zu optimieren.

1 Introduction

An increase in energy from renewable sources is crucial for a successful transition from fossil fuels to less CO2 emissions. Wind energy takes on a leading role, showing a trend towards larger and more efficient Wind Turbines (WT) over the last decades.

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Published online: 20 March 2023

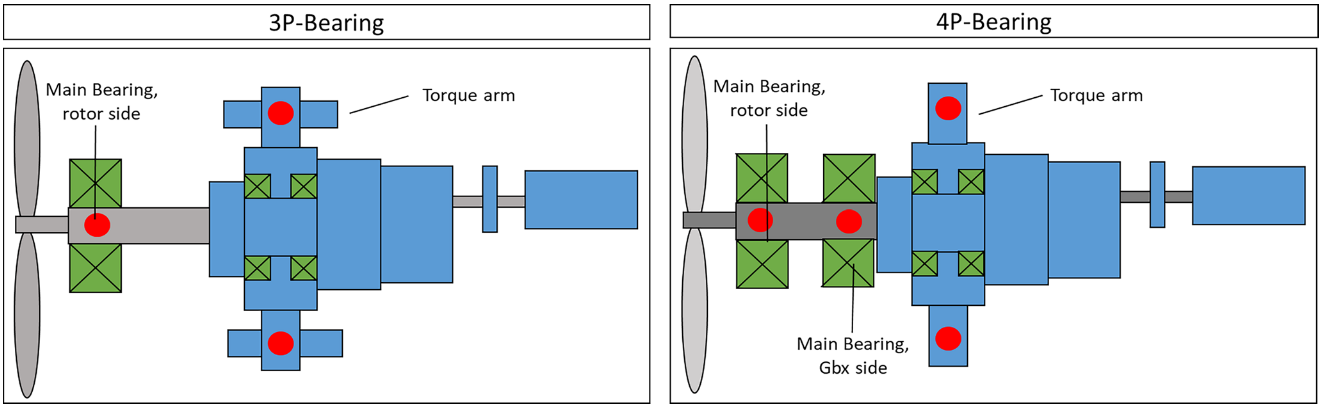


Fig. 1 Comparison of 3P and 4P arrangement

With the development of ever-larger turbines, various turbine configurations have also been developed, each of which is intended to achieve the optimum in terms of performance and costs. Generally speaking, wind turbines can be divided into direct drive turbines and turbines with a gearbox. Direct Drive Turbines are built without a main gearbox, the Torque is directly applied to a generator. The generator for Direct Drive Turbines tends to be very large. Adding a gearbox between the rotor and generator allows for a much smaller generator but adds an additional component to the turbine. Gearbox systems can be further categorized by the drive train's arrangement and the number of bearing points.

Most WTGs have a so-called 3-point (3P) or 4-point (4P) bearing arrangement (Fig. 1). In a 3P variant, the drive train is supported by a main bearing near the hub and by two torque arms of the gearbox. In this case, the generator is separated from the rest of the drive train by a coupling and mounted separately. In the 4P arrangement, an additional main bearing is inserted on the shaft near the gearbox.

A different design, which is applied in some newer wind turbine models, is an integrated drivetrain concept. This is characterized by a medium-speed generator that is directly coupled to the gearbox housing. The gearbox housing, in turn, is directly connected to the bearing unit of the main bearings and is not mounted separately. This allows a very compact design of the entire drive train, which positively affects weight, costs, and transport. On the other hand, this results in a very stiff connection between all turbine parts.

Aside from the development of larger and more efficient turbines, regulations for new turbine sites significantly reduce this technology's expansion. Such strict regulations are intended to reduce the negative impact of acoustic emissions from WT on residents nearby. To increase the number of potential areas for wind energy, the acoustic emissions of new turbines must be reduced. To achieve this reduction, developers aim to simulate the acoustic emissions of a WT early in the development process and optimize the WT accordingly. To reduce acoustic emissions of a WT,

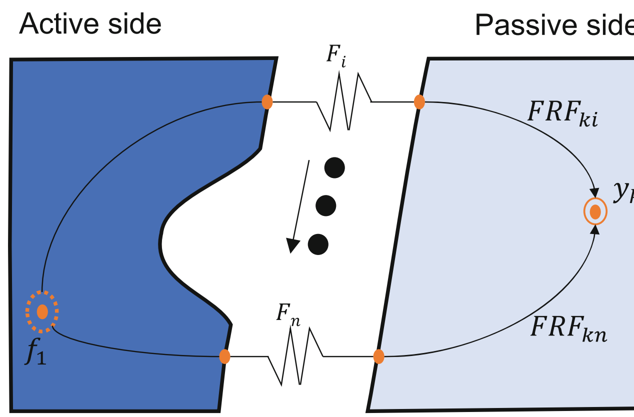


Fig. 2 Schematic of TPA according to [5]

the source of the vibrations (e.g. blades, gearbox), the emitting surfaces (e.g. blades, tower, and nacelle cover) and the transmission between source and surface have to be studied. The Transfer Path Analysis (TPA) can be used to determine the relevant acoustic transfer paths between an active and passive part of an exited structure and so be able to optimize the structure accordingly.

The focus of this paper lies in the identification of relevant transfer paths between the WT drivetrain and the mainframe. It will compare the transfer path for the 3P, 4P, and an integrated drivetrain design variation. The identified transfer paths can be used to improve turbine acoustics, especially by identifying problematic transmission areas [1].

2 Method

The TPA aims to study the transmission of vibrations from an active source to a passive structure. This can be a point of Interest, a surface of radiation, or an acoustic measurement point. The transfer paths are the coupling points between the active and passive sides (Fig. 2). To describe the transfer characteristics, the excitation and transfer characteristics are described separately. The excitation is measured with the interface forces at the transfer path (F_i) from the dynamic simulation, while the transfer characteristic is described by separately measuring the Frequency Response Function (FRF) or determining the transmissibility matrix of the structure [1].

In the literature, different types of transfer path analysis are distinguished [1]. There is the classical TPA, the component-based TPA, and the operational TPA. In classical TPA, the FRF is determined on the entire system, and the forces are measured during operation or determined via a mount stiffness or Matix Inversion Method. In component-based TPA, the forces are measured only on the active side, and the FRF is measured only on the passive side. This allows easy calculation of different variants with changing components. In Operational TPA (OTPA), the transfer properties are determined only by measurements from the operation, thus eliminating the need for separate measurements of the FRF. For the transfer properties, the transmissibility matrix is calculated from measurement data [1, 4].

The individual path contributions are calculated by multiplying the FRF and the Interface Force. The sum of the path contributions can be compared here with the measurements at the reference point.

$$y_k\,(f) = \sum_{i=1}^{n} \mathrm{FRF}_{ki}\,(f) * F_i\,(f) \tag{1}$$

The OTPA is based on the determination of a transfer matrix H_{yu} from the relationship between the input signal ($x^{(m)}(f)$) and output signal ($y^{(k)}(f)$). The input signals are usually the acceleration or forces at the transfer paths (a_{ai} or F_i), and the output signals are the accelerations or sound pressures at the reference point (u_q, p_q). The transfer matrix is based on the consideration of the machine structure at different load conditions e.g. depending on the speed range or torque (Indize r).

Thus, the determination of the transmission matrix is done by the least square (LS) calculation, which resolves Eq. 2 to H. The number of measurement blocks must be larger than the number of measured nodes u_q and the calculation of the transmission matrix must be performed for each frequency in the spectrum of the signal [2, 3].

$$\underbrace{\underbrace{\begin{bmatrix} H_{11}(f) & \cdots & H_{1m}(f) \\ \vdots & \ddots & \vdots \\ H_{k1}(f) & \cdots & H_{km}(f) \end{bmatrix}}_{\underline{\widehat{H}}} \times \begin{bmatrix} x_1^{(1)}(f) & \cdots & x_r^{(1)}(f) \\ \vdots & \ddots & \vdots \\ x_1^{(m)}(f) & \cdots & x_r^{(m)}(f) \end{bmatrix}}_{\underline{X}} = \underbrace{\begin{bmatrix} y_1^{(1)}(f) & \cdots & y_r^{(1)}(f) \\ \vdots & \ddots & \vdots \\ y_1^{(k)}(f) & \cdots & y_r^{(k)}(f) \end{bmatrix}}_{\underline{Y}} - \begin{bmatrix} \mu^{(1)} \\ \vdots \\ \mu^{(k)} \end{bmatrix} \tag{2}$$

The solution can be done either by H1-calculation (Eq. 3) or by H2-calculation (Eq. 4). Both variants differ in the weighting of noise in the input or output signal. As a shorthand notation, in Eq. 3 and 4 the variables X and Y stand for matrices in complex form, X^T and Y^T are the complex conjugate matrices, G_{yy} and G_{xx} stand for the spectral power density and G_{xy} and G_{yx} for the cross-power spectrum (CPS) [3]. It has been shown that the H2 method produces better results when applied from simulation data.

$$\underline{\widehat{H}_1} = \underline{Y\,X}^T * \left(\underline{X\,X}^T\right)^{-1} = \frac{G_{yx}}{G_{xx}} \tag{3}$$

$$\underline{\widehat{H}_2} = \underline{Y\,Y}^T * \left(\underline{X\,Y}^T\right)^{-1} = \frac{G_{yy}}{G_{xy}} \tag{4}$$

The path contribution is finally determined using Eq. 5. For the calculation of the path contributions, a measurement of the FRF is therefore no longer necessary.

$$\underline{y_k}(f) = \sum \underline{\widehat{H}_{yu}}(f) * \underline{u_q}(f) \tag{5}$$

The graphical representation of the path contributions is done using a Partial Path Contribution Plot (PPCP). This allows the amplitude of the path contributions for each path to be plotted on top of each other depending on the speed, and frequency at an operating point or order.

3 Modelling approach

The structure of the reference model and the variations are described below. A turbine model with a 3P-bearing arrangement is selected as the reference model. This model was subsequently extended to include components to be able to represent the 4P-bearing arrangement and the integrated drivetrain design.

3.1 Reference model, 3P-bearing

The reference turbine has a rated power of 3 MW. The gearbox is a 3-stage planetary gearbox (2 planetary stages + one spur stage). The turbine has a rated generator speed of 1450 rpm. To consider the NVH performance of the turbine, all major structures of the turbine are modeled as flexible bodies. This included the blades, rotor shaft, hub, machine frame, tower, and gearbox. In the gearbox, all gears are further modeled as flexible bodies, and tooth contact considers the gear wheel deformation as well as tooth bending and contact stiffness to accurately model the tooth mesh frequencies. The bearings in the gearbox and for the gearbox support are integrated via spring-damper elements with stiffness and damping properties, main bearings are rep-

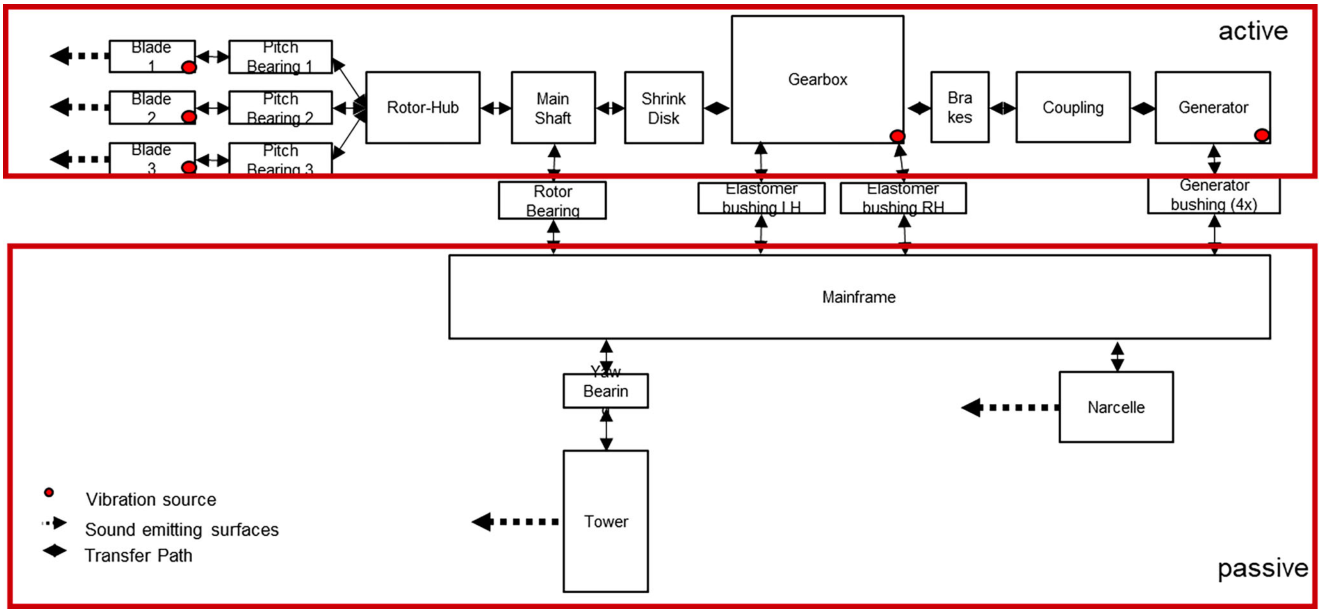


Fig. 3 TPA of a WT-Drivetrain

resented via a user-subroutine force element with bearing stiffnesses and reaction forces.

For the TPA of a WT drivetrain, the system is first divided into an active and a passive side. The active side consists of the drive train with a gearbox, generator, and main shaft as well as the hub and the blades. The passive side is thus formed by the Mainframe and the tower. This results in the transfer paths considered via the main bearing, the gearbox bearing of the torque arms, and the generator bearing (Fig. 3). A measuring point at the tower head is selected as the reference point for the analysis.

3.2 4P-bearing

For the 4P bearing arrangement, the model can be extended by an additional main bearing. For this purpose, the FEM model of the mainframe is extended by a bearing housing similar to that of the rotor-side bearing. For the bearing on the gearbox side, only the radial stiffnesses of the main bearing are considered. In the axial direction, the bearing is defined as soft. To reduce additional forces due to overconstraints, the Gearbox support is modeled to only support the torque loads, with reduced stiffness in the axial and horizontal direction.

3.3 Semi-integrated drivetrain

For the integrated drive train design, the housing of the gear unit was coupled directly to the bearing housing of the main bearing on the gearbox side. This eliminates the need to support the transmission via the torque arms. However, as part of the rebuild of the model, the generator was not integrated with the gearbox unit. With a separate Generator, this modeling approach represents not a fully integrated drivetrain, rather it can be described as a semi-integrated drivetrain. The focus of the research is on the influence

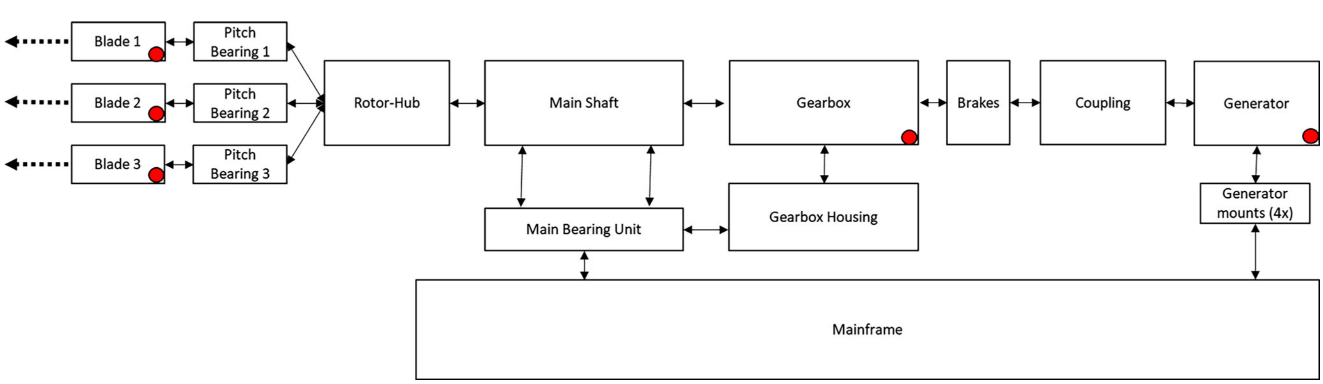


Fig. 4 schematic of the semi-integrated Drivetrain

of gearbox excitation on the structure, so this modeling approach is sufficient. The gearbox housing was rebuilt as a FEM model and modally reduced. The connection with the bearing housing was made in the MBS model via a joint with 0 DOF (Fig. 4).

4 Results

In the following, the results of the system variations are presented. First, the waterfall plot of the accelerations at the reference point at the tower head in the horizontal direction is shown here as an overview, then the PPCPs for the nominal speed and at the first and third tooth mesh frequency of the IMS (Order 4.73 and Order 14.19). These have proved to be particularly relevant in the analysis of system behavior.

4.1 Reference model, 3P-bearing

In the waterfall plot of the 3P bearing, the tooth mesh frequencies of the 2nd gear stage can be seen very clearly. The tooth mesh frequency of the 1st harmonic of the HSS stage is also clearly visible.

The PPCP at nominal speed show as dominant transfer paths the paths via the gearbox bearing in radial and vertical direction as well as via the main bearing. The transfer paths via the generator have only minor contributions. Also, the excitation by the gear mesh frequencies is shown to be dominant in the PPCP. Looking at the PPCP for the individual orders, for order 4.73 only the transfer path via the gear support in the vertical direction and at higher frequencies is relevant, for order 14.19 again all TP via the gearbox and the main bearing in the vertical and radial directions (Fig. 5).

4.2 4P-bearing

The waterfall plot shows a generally lower amplitude of the oscillations at the reference point. Especially the 1st-order IMS and 1st-order HSS show a much lower oscillation. This is also evident in the forces at the interfaces, these are much lower in the region of excitation of the 1st IMS than still with the 3P-bearing.

For the PPCP, it can be seen that the path contribution via the floating bearing on the gearbox is very small compared to the fixed bearing. The majority of the vibrations are therefore transmitted via the rotor-side bearing and the gearbox supports, as in the case of the 3P-bearing arrangement. The PPCP at the orders also generally show the same behavior as the 3P model. At order 4.73, however, the TP via the main bearing is now also more involved (Fig. 6).

4.3 Semi-integrated drivetrain

The results of the integrated drivetrain design initially show a significantly higher amplitude of the vibration behavior at the reference point as well as on the entire model.

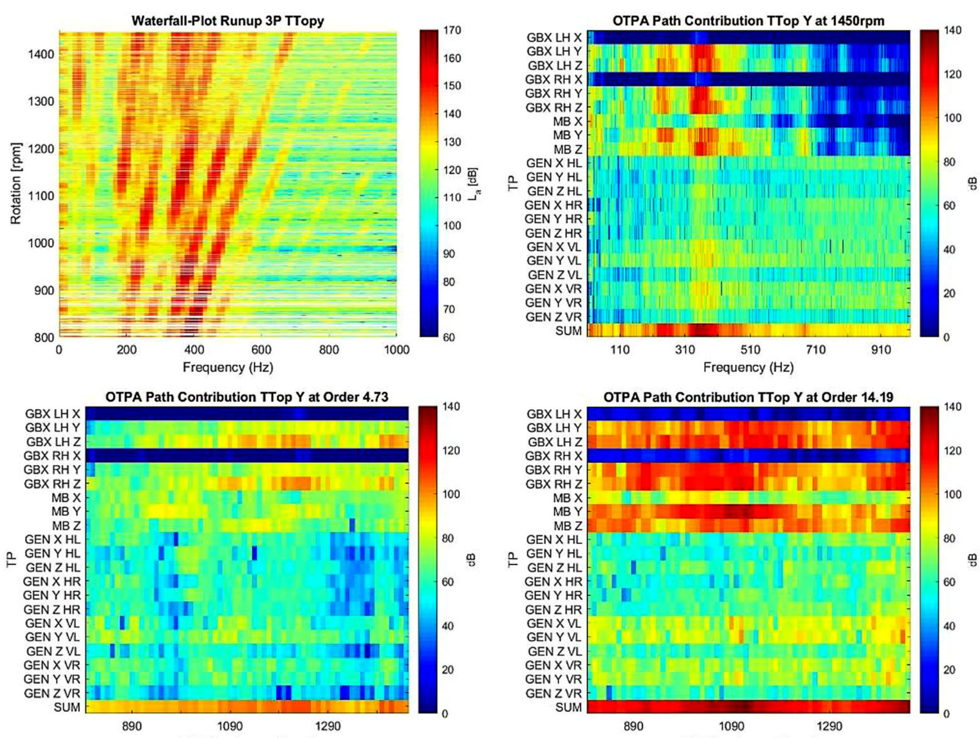


Fig. 5 Results 3P-Bearing

Fig. 6 Results 4P-Bearing

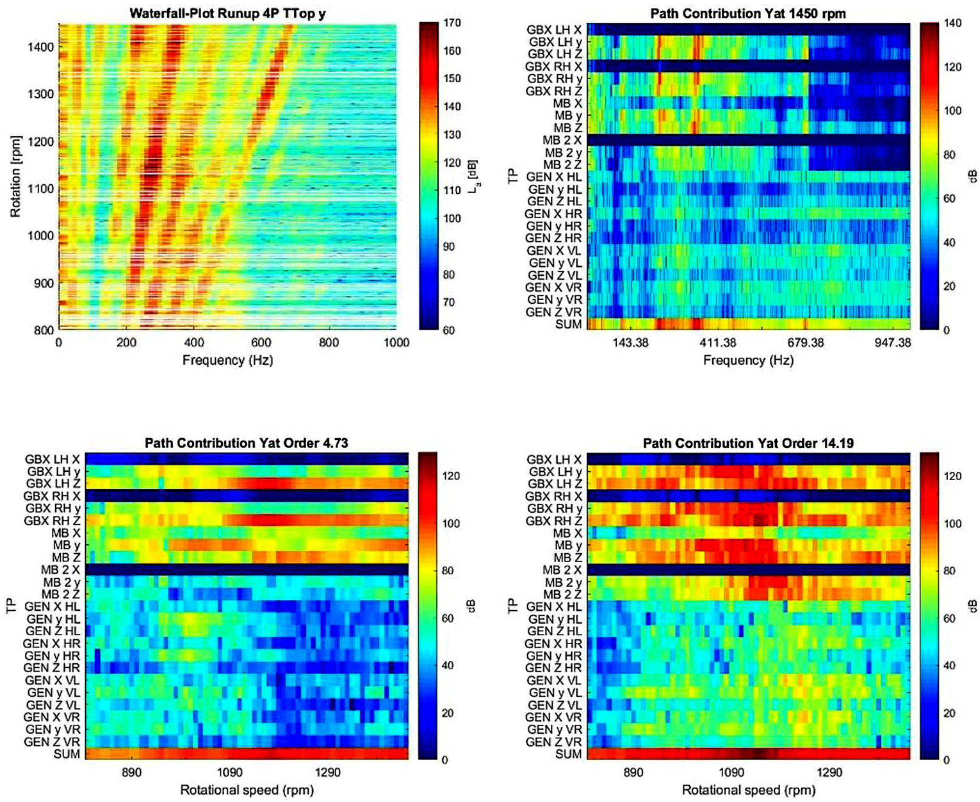
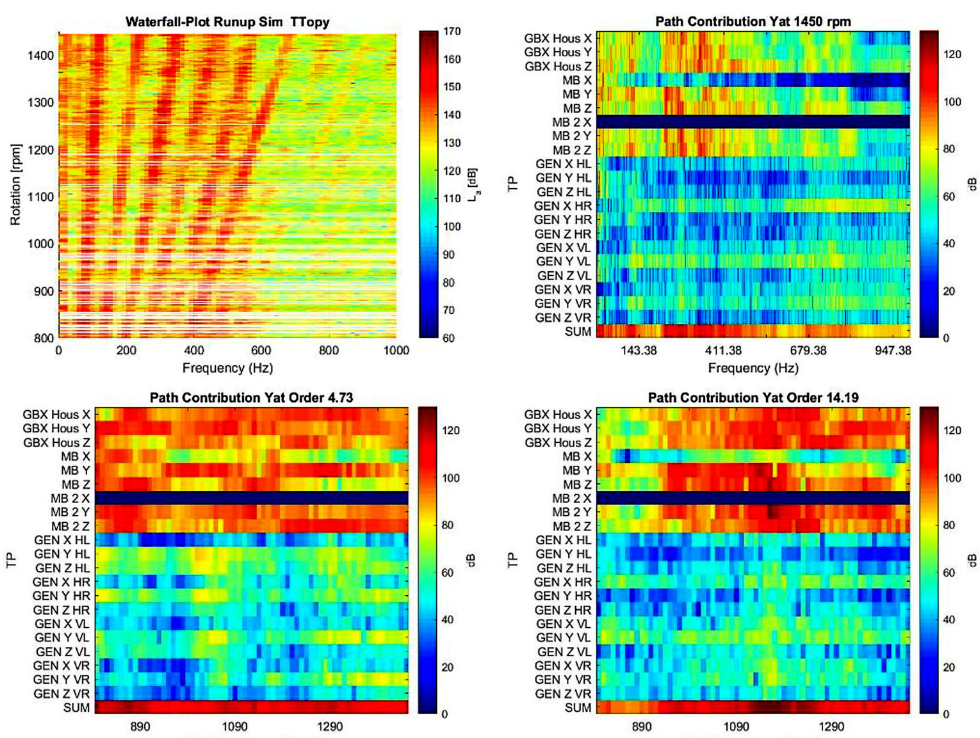


Fig. 7 Results, integrated Drive-train



The forces induced by the drive train are also significantly higher than for the 3P and 4P bearings. The PPCPs also show the Transfer path between the Gearbox housing and Mainframe as the relevant path. A reason for the higher vibrations may be the direct connection between the gear housing and the machine carrier. While these two components are decoupled via elastomer bearings and roller bearings in the 3P and 4P bearing arrangement, this is directly connected to the integrated drive train design (Fig. 7).

5 Conclusion

In this paper, the method of TPA was first explained and OTPA was specifically discussed. The advantage of OTPA is that no separate measurement of the FRF is necessary and therefore it can be easily applied to measurements of a WT in operation. The method was applied to the results of a simulation model of a WT with a 3P bearing. It is shown that for a WT drivetrain the transfer paths via the gearbox bearing in the horizontal and vertical direction and via the main bearing in the radial direction show the largest path contribution. Furthermore, the path contributions were also investigated depending on the speed or order.

The design of the wind turbine was varied based on the reference model to be able to represent other turbine configurations. Thus, a 4P bearing and an integrated drive train design were simulated and compared with the reference model. It was shown that the total vibrations decreased for the 4P bearing, but the transfer path via the rotor-side main bearing was still more relevant than via the new gearbox-side bearing. For the integrated drive train concept, the total vibrations were significantly higher than for the 3P or 4P bearing, and the transfer path via the direct coupling between the gearbox housing and mainframe is now also significantly more relevant.

Funding This research is part of the Project "TraWin" which is funded by the Federal Ministry for Economic Affairs and Energy of Germany. We also thank our project partners, who provided equipment, insight, and expertise that greatly assisted the research.

Funding Open Access funding enabled and organized by Projekt DEAL.

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