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Model-Based Estimation of Contact Forces of a Rotating Blade

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The paper introduces into the concept of model-based force estimation of rotating structures with impacts. Therefore a rotating blade is considered, which periodically contacts with nonrotating grounding like a gasturbine blade with the housing. Using bending information from two measuring locations on the rotating blade a model-based technique is applied to estimate the resulting vertical contact force to the blade tip.
Using the model of the unaffected system, a robust observer is build to reconstruct the contact force history. Experimental results of the observer-based estimation of unknown impacts are presented. The work demonstrates, that the proposed observer technique can be applied to impact estimation affecting rotating blades.

1. Introduction

The control of mechanical structures has been the focus of several scientific and industrial efforts of the last decade in the area of robotics, vehicle- and rotor dynamics [3,5,8,9,14,15]. Actual works are focused on flexible (space) structures concerning aspects of practicability/realizability and precision [4,6]. In all these mentioned areas, the classical fields of mechanics/dynamics and control theory as well as data processing are connected. Advanced control approaches usually need some knowledge about the system to be controlled. This can be realized e. g. by the way of mathematical models (e. g. sets of differential equations) as the base to design observers or regulators/compensators.

Effects of friction, backlash or impacts often occur in elastic structures. Due to these effects, the system behavior in operation may switch between that described by linear and that described by nonlinear formulations. These effects especially aggravate the systematic control of such system types. Impacts have strong effects on the vibrational behavior of flexible structures. As an example, rubbing effects between rotor and seal, or between rotating blades and the inner casing wall should be mentioned. The contribution does not focus on the analysis of these flexible systems with impacts, but on the application of a suitable estimation technique. The typical restrictions of this kind of machines or rotordynamic structures are that measurements of the impact can not directly be taken. This means that impact forces or torques can only be obtained indirectly via the resulting vibration effects.

This contribution deals with the application of the proposed observer technique to estimate contact forces of rubbing turbine blades. A comparable technique was published by Hollandsworth and Busby [7]. Core of their contribution is the use of an inverse technique. Accelerations are used as measurement. Their results show

- that the quality of the impact force estimation is very good if the measurements are done directly at the contact position,
- that the quality of the estimation strongly depends on the measurement position close to the contact point and
- that the estimation of the correct impact force amplitude seems not to be possible using measurements beside the contact point.

This contribution also deals with a model-based approach. Here the PIO-technique is applied to estimate the contact force as well as contact displacements using a minimal number of measurements (one or two sensors) applied beside the contact position. For principal validation of the observer a simply to realize experiment has been done. The system to be considered for the application of the PIO-technique is described in detail in section 3.2.

2. Model-Based Impact Estimation

The idea here is to apply a model-based approach based on a Proportional-Integral-Observer-technique (PIO) to reconstruct unknown external inputs. The base of this technique - the Disturbance Observer (DO) - is well known for the application of modeled disturbances (known external effects acting to the considered system). The PIO technique realizes a special modification of the DO and is used for the detection of faults or for determination of nonlinear effects acting on dynamic systems (like friction, backlash) as a modern Fault-Detection-Isolation (FDI)-scheme. The main important aspect is, that no model for the affecting disturbance is necessary. This allows the application to those problems, where no model is available or the application of a known model is not useful (e. g. because of complexity).

The problem is described by

$$\dot{x} = Ax + Bu + Nn(x, u, \mathrm{t}) \quad , \quad y = Cx \quad , \tag{1}$$

with the state vector x of order n, the vector of measurements y of order r_1, and the known input vector u of order m. The system matrix A, the input matrix B and the output matrix C are of appropriate dimensions. The unknown effects, which affect the elastic structure are considered with the vector function $n(x, u, \mathrm{t})$ of dimension r_2 and describe in general external inputs, here the contact force. In [13] it is shown, that for this application, the number of independent measurements r_2 must be higher than the number of considered external inputs r_1.

The input matrix N locates the disturbance to the system description and is assumed as to be known. The structure of the PIO is given in Fig. 1. From the structure it can be seen that the observer feedback is realized by two loops: a proportional (classical) feedback usually used in the Luenberger Observer approach and a second integral feedback introduced in [11].

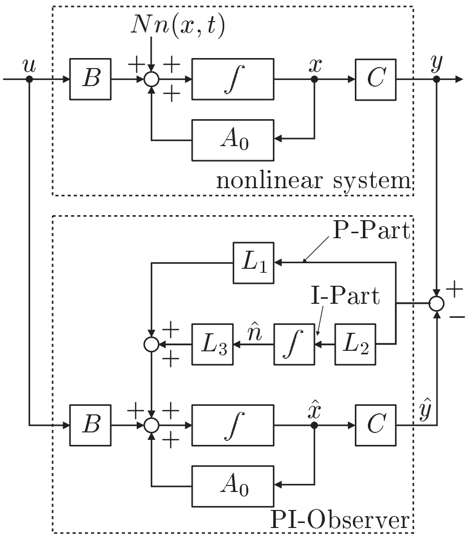

Figure 1: Structure of the PIO.

The main important applications are realized in [10,13]. From the structure of the PIO depicted in Fig. 1 it follows, that the dynamics of the PI-observer is described by

$$\dot{\hat{x}} = A\hat{x} + L_3\hat{n} + Bu + L_1(y - \hat{y}) \quad , \quad \dot{\hat{n}} = L_2(y - \hat{y}) \quad , \tag{2}$$

where $\hat{y} = C\hat{x}$. The integral loop of the PIO is represented by $\hat{n}$. Based on the proof given in [11,13] the observer behavior can be understood in the way, that using high observer gains (due to a special calculation procedure) and also high design values ($L_3 = N$) (with the disturbance input matrix N) the behavior

$$\hat{x} \to x \quad \text{and} \quad \hat{n} \to n \tag{3}$$

results. This means that the observer state $\hat{x}$ reconstructs the modeled system state x and the integral feedback of the PIO represents the external disturbance n. The details are given in [11,13]. Unfortunately this is not a mathematical constructive proof.
Applying a linear disturbance observer technique to a nonlinear problem needs additional explanations:

- The system (Fig. 7) is not considered to be a nonlinear system, but contains a complex contact model.
- The system will be assumed to be a linear system with an unknown force acting on the system. The characteristics of the contact mechanism are assumed to be unknown and nonlinear.
- The assumption of an unknown force acting at a known position onto the linear system allows the application of a linear observer technique, if the observer technique is able to estimate unknown acting effects, cf. [13].
- The observer-technique to be applied must be able to estimate the modeled states (displacements) and the effect (force).
- If the observer technique gives estimations of the displacement at the contact point and of the interacting force, the contact characteristic can be described [12] as a function of these estimated values. In [13] an actual proof of the convergence behavior is given.

The advantages of the proposed Proportional-Integral Observer (PIO) are:

- Estimations of contact displacement and contact force are possible.
- For the observer-design the nominal linear model description of the beam can be used.
- For the design procedure itself classical numerical procedures like LQG Design or Pole Placement approaches can be used.

An application of the PIO-technique is given in [1]. The system to be considered consists of an one-sided clamped elastic beam which is subjected to an impact load at his free end. The resulting movements are restricted due to a contact of the end effector with a stop. The results show that the observer technique works well for this system and is able to estimate the contact force as well as the contact displacement [1].

3. Experimental Setup and Results

3.1 Validation of the observer technique

In a first step the experimental validation of the proposed observer technique for estimation of the contact forces is carried out. Therefore the blade is excited normal to its longitudinal axis at the tip by a impulse force hammer (Fig. 2).

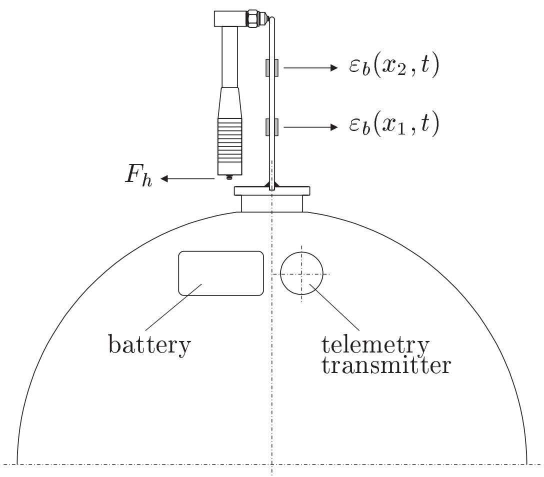

Figure 2: Experimental validation of the observer technique.

The length of the test blade is 100mm. The blade is modeled using finite beam elements of suitable length denoting the measurement positions as modeling nodes. The contact point is at the blade tip, the strain measurements are taken at $x_1 = 35\,\text{mm}$ and $x_2 = 70\,\text{mm}$ from the blade root. The blade material is steel and the rectangular cross-sectional area is of $44.4\,\text{mm} \times 5\,\text{mm}$. The excitation force measurement is realized using a standard impulse hammer. The contact is realized with a set of different tips (steel and two types of vinyl). The measurements have been done at the University of Essen, the application of the PIO to the measurements at the Gerhard-Mercator-University of Duisburg. The experimental results of the first step are given in Fig. 3 and 4. The different tips of the impulse hammer are characterized by different frequency ranges to be excited. The soft tip is valid up to 0.3 kHz and the supersoft tip up to 0.2 kHz. Fig. 3 and 4 shows the comparison between measured and estimated contact force.
For this experiment, the blade has been excited at the tip and the force signal F_h of the impulsehammer is recorded.

The resulting vibrations are measured using the strain-gage full bridges at the blade. Subsequently the signals of the strain gages are transfered via telemetry transmitters, which are supplied by batteries, from the disc to the receivers. The model information and the measured bending strains are used for the observer reconstructing the contact force.

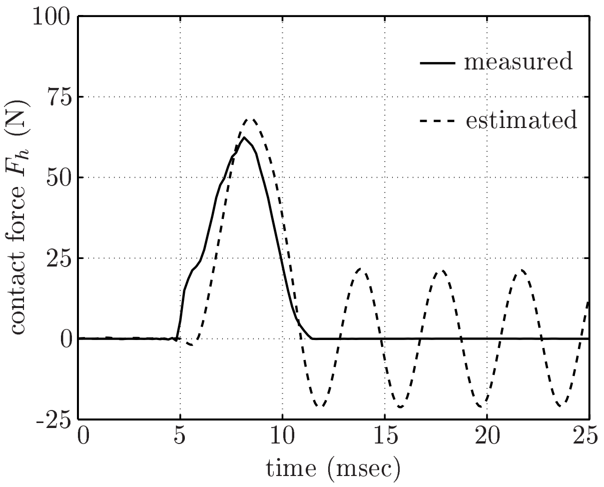

Figure 3: Measured and estimated contact force, supersoft tip

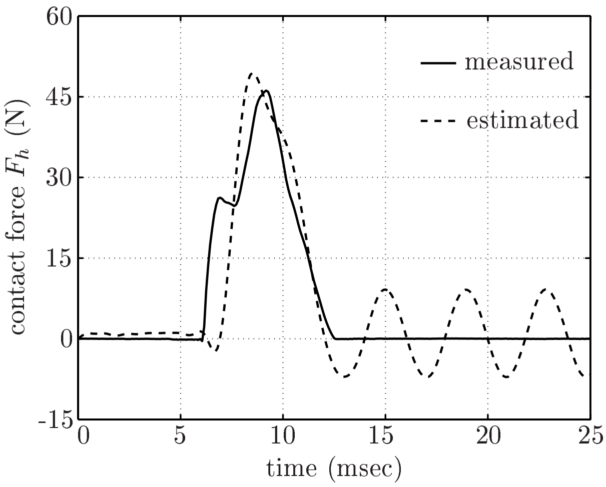

Figure 4: Measured and estimated contact force, soft tip

The estimated contact force is compared with the direct measurement of the contact force to validate the observer quality. The given results show that the observer estimates well the contact force. Here the estimation does not fail related to the force peak value and also to the time history. The remaining vibration in the estimated force after the intrinsic contact is nearly identical to the first eigenmode of the system to be observed at 250 Hz. This observer movement is affected by the external, unknown contact. Designing the observer gains a model of the mechanical model is required. Depending on the inner system dependencies, the measurements, and the design procedure the observer gets also dynamical properties. In this application an undamped mode remains, which is in contrast to other applications [1]. It is not clear in which way the existence of this observer mode can be avoided. Future work will deal with this problem.

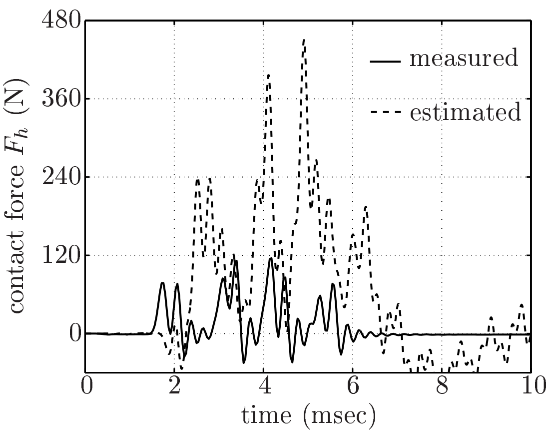

Figure 5: Measured and estimated contact force, steel tip

To examine the quality of the contact force supplied by the observer at multiple contacts and input force signals with a wide frequency range (up to 7 kHz), the beam is excited by the impulse hammer with a steel tip. Fig. 5 shows the comparison between the measured and estimated contact force F_h at the blade tip. Due to the high flexibility at the tip the blade rebound at the hammer and produces four double impacts in a short interval of approx. 4.5 msec. The PI-Observer is able to reproduce this four contacts as well as the double hit characteristic of the force F_h. To reduce the influence of remaining eigenmodes in the estimated signal butterworth bandstop filters, realized in MATLAB, was used. High-frequency components above 4 kHz were removed applying also a digital low pass filter.
Comparing the measured and estimated data, it could be noticed, that there is a slight difference in the time intervals between the both peaks of one double impact. For the application of the observer it is assumed that the effect of the contact instantaneously appear at the measurement location. However, this is practically not the case due to the time required from the bending wave to run from the tip of the blade to the measurement location. Maybe the observer yields therefore a phase distortion in the time history of the estimated contact force.
The double impact characteristic of the second and third contact of the estimated force is not so distinct probably due to the reduction of the second eigenmode by filtering. The amplitude of the applied steel-steel contact force is now overestimated by the PI-Observer. Also this effect will be focussed in future work.

3.2 Application of the observer to blade rubbing processes

For the examination of blade rubbing processes a test-rig was developed and built at the Institute of Mechanics at the University of Essen (Fig. 6).

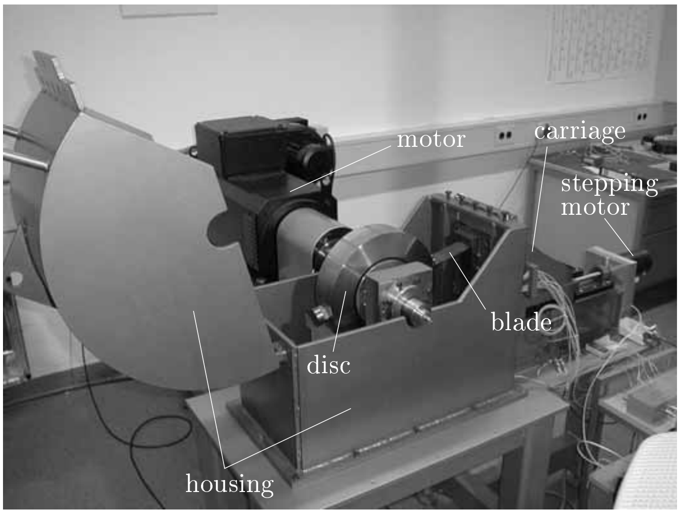

Figure 6: Test rig (Institut of Mechanics, University of Essen)

The test rig consists mainly of two basic structural components, the rotor and the feeding device. The rotor is comprised of a shaft and a disc clamped to the shaft. The dimension of the disc were selected in such way, that the polar moment of inertia is sufficient to hold the rotating speed nearly constant during the rubbing process. The shaft is supported in two angular-contact ball bearings and is driven by a DC-motor via a metal bellow coupling. Due to the high bending stiffness of the shaft, the rotor can be considered as rigid. To ensure a sufficient level of safety during a blade loss, a massive steel housing covers the entire assembly. The feeding device consists of a carriage, which slides on two linear guidances and can be translated by a stepping motor via a spindle. The rubbing surface has been mounted on the supporting plate on the front of the carriage (see Fig. 7). For conducting the experiments, the rubbing surface is moved, via the stepping motor, towards the rotating blade until the blade rubs along the rubbing surface. Since the surface is not penetrated by the blade tip, the blade must shorten through bending and axial deformation in the longitudinal direction in order to pass the rubbing surface. The intensity of the rubbing processes can be determined through the feed rate of the carriage. The control of the stepping motor is performed by a computer and enables various feeding motions, such as ramp or step variation, at different speeds. The movement of the carriage relative to the casing is measured with an eddy-current sensor.

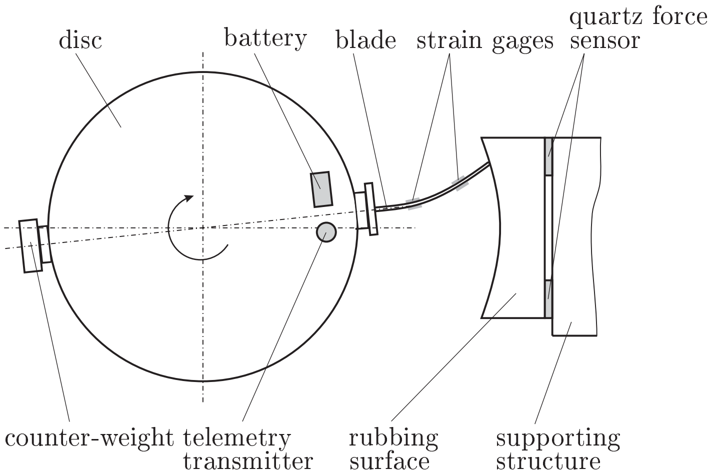

Figure 7: Setup for experimental investigations of blade rubbing processes.

The test blade is screwed into the disc and rubs along the curved rubbing surface once per revolution (see Fig. 7). The imbalance of the blade is compensated by a counter-weight, which is attached to the opposite side of the disc. During the rubbing process the rotating blade is loaded by the contact forces at the tip. The contact forces in the axial and tangential (vertical) direction of the blade are measured by quartz force sensors, which are mounted between the surface and the supporting structure. The measurement method to obtain the bending strains along the beam and the transmission of the signals from the rotating system to the data acquisition board were already described in section 3.1. The experimental setup allows to measure both, the contact force directly via the quartz force sensors and also the resulting system vibrations at points apart from the rubbing blade tip. The structure thus enables to simulate the problems of the mentioned practical applications (i. e. no direct measurements) but also to take additional measurements near the contact point for comparison. A more comprehensive description of the experimental set-up and the applied measuring equipment can be found in [2].

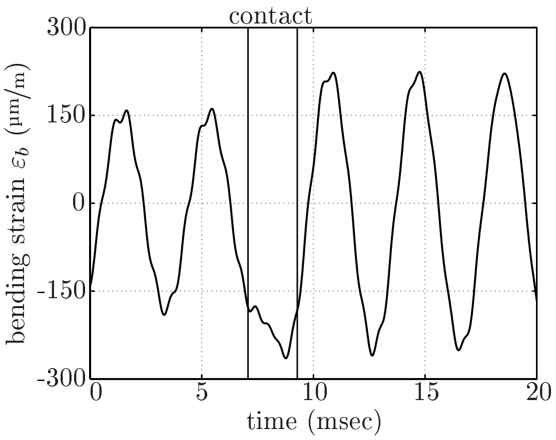

Figure 8: Measured bending strain at $x = x_1$

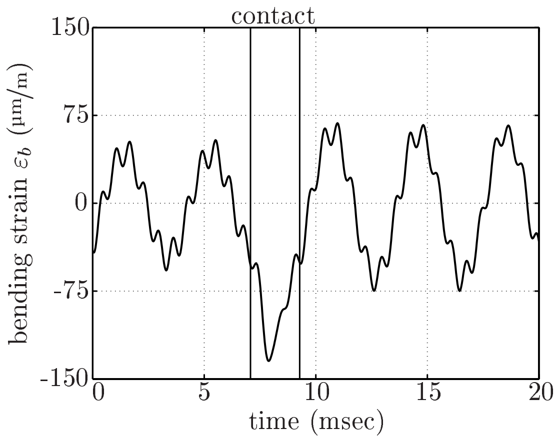

Figure 9: Measured bending strain at $x = x_2$

Figure 8 and 9 show the measured bending strains at both measurement locations x_1 and x_2. Due to the low pass characteristic of the blade at x_1 the signal is mainly determined by the first eigenmode. The additional influence of the second eigenmode on the measured bending strain at x_2 is illustrated in Figure 9. Both strains are used as input signals of the PIO.

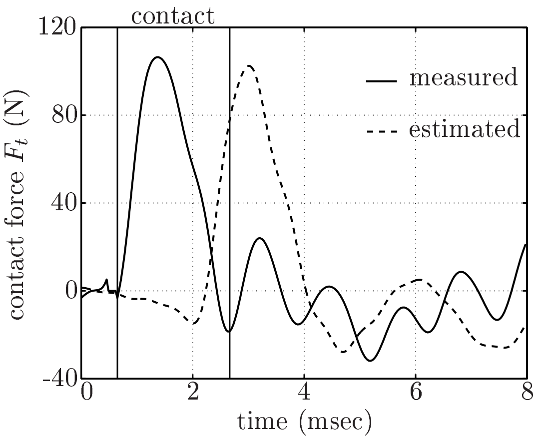

Figure 10: Measured and estimated contact force at the blade tip, measurement 1

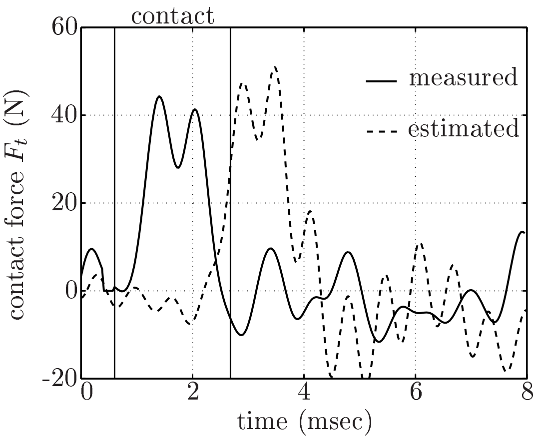

Figure 11: Measured and estimated contact force at the blade tip, measurement 2

Figure 10 and 11 show the time histories of the tangential contact force F_t, as measured directly by the quartz force sensors and reconstructed with the PIO from the bending strains, during two different rubbing processes both at a rotating speed of 1200 rpm. It should be mentioned that the estimated force was scaled with a correction factor, since the PIO overestimate the magnitude of the force, and filtered as described in section 3.1. At the rubbing process in Figure 10 the curve of the tangential contact force F_t is very similar to the shape of a half-sine wave, whereas the ripple at the falling force indicates an influence of the blade flexibility on the contact. Here, the PIO yields a good reconstruction of the characteristic of the contact force and also of the duration of the rubbing process. In Figure 11 it can be clearly seen that the contact force at the blade tip oscillate during the rubbing process. Thereby, the frequency corresponds to the second eigenfrequency of the blade at 1550 Hz. One can see that the PIO is able to reconstruct also the high-frequency influence of this blade eigenmode on the contact force history very well.

4. Conclusions

The paper introduces into the concept of model-based impact estimation of elastic structures; here the contact force estimation of a rubbing turbine blade. A robust linear observer technique is applied to estimate the unknown tangential contact force acting normal to the longitudinal axis at the tip of a rotating blade. The experimental setup shows the observer-based estimation of unknown impacts affecting the flexible turbine blade. The validation of the observer scheme yields that depending on the contact characteristic of the contact itself, the observer produces good estimation for smooth contacts. For stiff contacts the proposed scheme is able to detect the contact force history, the contact time, the contact shape, and also the contact force dimension. The application of the scheme to an rotating turbine blade experiment show the same successful results. Future work has to solve the remaining open questions.

5. References

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