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Bearing Stiffness Determination through Vibration Analysis of Shaft Line of Bieudron Hydro Powerplant

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1 Introduction

Bieudron power plant houses three turbines and associated hydro generators which set records in terms of Pelton head (1883 m), output (465 MVA) and generator capacity per pole (37 MW).

Each vertical-shaft synchronous machine has a two-bearing arrangement with a guide bearing above the stator and a guide bearing below the stator. The thrust bearing designed to carry the entire weight of the rotating mass of approximately 540 t, is combined with the upper guide bearing. The coupling between the generator shaft and the Pelton turbine is immediately below the lower guide bearing.

The shaft line was designed to have a first bending critical speed between load rejection speed and runaway speed. The critical speed is determined by the structural properties of shaft line and bearings which are independent of speed and the speed dependent characteristics, e.g. oil film elasticity and damping, gyroscopic effect, magnetic pull.

In general, the structural properties of the shaft line are well defined in terms of its material elasticity and mass distribution. On the other hand, accurate values for the bearing elasticity are rarely available and estimates must be made. This is an unsatisfactory situation since it may lead to erroneous predictions of the shaft critical speed. In the case of Bieudron power plant the bearing elasticity was determined through vibration measurements using the modal analysis technique and subsequent finite element calculations.

2 Problem definition

The bending and torsional natural frequencies of the unit no. 3 shaft line were measured at the Bieudron power plant.

General data:

Rated capacity:	465 MVA
Nominal speed:	429 min^{-1}
Load rejection speed:	486 min^{-1}
Runaway speed:	800 min^{-1}
Total weight:	528 t
Total polar moment of inertia:	1,545 tm^2
Total length:	11.7 m

The bearing elasticities should be determined from the measured frequencies and transfer functions with the aid of a calculation model:

I. Using a modal analysis on the basis of the ABB standard input data set isotropic bearing elasticities were adjusted so that the measured frequencies resulted.

II. The transfer function was calculated using a harmonic analysis on the basis of the measured damping values and compared with the measured curve. The bearing elasticity was estimated by extrapolating the kinetic elasticity to zero frequency.

3 Measurement setup

The complete shaft line with generator rotor, Pelton wheel and bearings was installed. The frequencies of the bending modes were determined for the following bearing condition:

- DE and NDE bearings dry and wedged. In both bearings the tilting pads were pressed against the shaft using a wedging device

To determine the torsional natural frequency, the NDE bearing was under oil and the wedges on the DE bearing were removed.

The natural frequencies were determined using an impact test. The bending vibrations were registered in two measuring planes with three measuring points at positions 1, 2 and 3 (s. Fig. 1). The impact location was at position 3. To determine the torsional natural frequency, the tangential accelerations were measured on the impact point position 4 used for torsional excitation and on the outer diameter of the Pelton wheel, position 6. A HP four channel signal analyzer and PCB vibration accelerometers were used to measure and record the vibration data.

4 Results and comparison with the calculation

The amplitudes of the response spectra are shown in Fig. 2. The comparison for both measurement planes revealed an isotropic behavior of the shaft line. The bending natural frequencies are summarized in the following table. A natural frequency of 40.0 Hz was measured for the 1st torsional mode where the Pelton wheel performs a motion of revolution against the fixed generator rotor. The calculated value for the 1st torsional mode was 40.9 Hz using the finite element model shown in Fig. 3.

Mode	Natural frequency [Hz] DE & NDE bearing dry and wedged	Mode shape
1	13.5	1 st bending mode of shaft
2	16.3	transverse mode of NDE bearing bracket
3	25.3	2 nd bending mode of shaft
4	32.3	tilting mode of bearing bracket
5	41.5	3 rd bending mode of shaft

Table 1: Bending natural frequencies in [Hz] and mode shapes

Mode shapes were determined from the measured amplitudes and phases of the response spectra and interpolated with a cubic spline between the measurement points 1, 2 and 3. The mode shapes are shown in Fig. 4 to Fig. 6 at an interval of one quarter of a time period.

A finite element model (Fig. 3) was created for the shaft line on the basis of the ABB input data set and a modal analysis carried out. For comparison with the measurement the speed dependent oil film elasticity and damping as well as the gyroscopic effect and magnetic pull were not considered. The effect of the rotary inertia and the shearing deformation was included in the calculation.

The FE-modal analysis was carried out for varying bearing elasticities and compared with the measurement. The mode shapes calculated are shown in Fig. 4, Fig. 5 and Fig. 6 for comparison with the measured mode shapes. The results for the natural frequencies are summarized in the following table.

Mode	Measurement	FE-modal analysis				
		Bearing elasticities in mm/MN				
	DE & NDE wedged	DE: 0.54 NDE: 0.81	DE: 0.27 NDE: 0.40	DE: 0.13 NDE: 0.20	DE: 0.13 NDE: 0.26	DE: 0.2 NDE: 0.2
1	13.5	9.8	11.7	13.0	12.7	12.8
2	16.3	-	-	-		
3	25.3	16.4	20.8	25.0	25.0	22.7
4	32.3	-	-	-		
5	41.5	35.2	37.5	38.8	38.5	38.7

Table 2: Comparison of measured and calculated natural frequencies [Hz]

The best agreement between measured and calculated natural frequency resulted for a bearing elasticity of 0.13 mm/MN for the DE bearing and of 0.20 mm/MN for the NDE bearing. These values describe the mechanical elasticity of pedestal, attachment of pedestal to foundation and surrounding foundation. They do not include the oil film elasticity.

In order to confirm the results for the bearing elasticities based on modal analysis the frequency response functions were calculated applying a harmonic analysis and compared to the measured spectra. Fig. 7 shows that good agreement was obtained between measured and calculated acceleration peaks for 1st and 2nd bending natural frequency. The resulting kinetic elasticity is also shown in Fig. 7. The bearing elasticities can be obtained from the kinetic elasticity by extrapolating to zero frequency:

- DE bearing: 0.15 mm/MN
- NDE bearing: 0.21 mm/MN

A difference of only 13% is found between the calculated bearing elasticities based on vibration measurement and finite element

- modal analysis
- harmonic analysis.

The results for the bearing elasticities using two different calculation methods provide a valuable basis for a precise quantitative prediction of the critical speed. In a 2nd step the speed dependent elasticity of the oil film will be calculated. Modern numerical methods will be used to account for type of journal bearing, clearance, speed and bearing force.

Biographical details of the authors

Pierre Loth graduated from the Ecole Polytechnique Federale de Lausanne, Switzerland, in 1967 and has been mostly active in the design of hydroelectric schemes and equipment. He is now head of the hydromechanical section of Cleusson-Dixence scheme at EOS.

Heinrich Sprysl graduated in Mechanical Engineering from the Czech Technical University of Prague and received his doctorate from the Ecole Polytechnique Federale de Lausanne, Switzerland. He has worked under Prof. Kellenberger as assistant at ABB's Department for Mechanical Studies. He is responsible for mechanical calculations for hydro and diesel generators.

Günter Ebi received his M.S. degree in Mechanical Engineering from the University of Karlsruhe, Germany in 1981 and in Material Sciences from the University of Connecticut, USA in 1982. In 1987 he received his Ph.D. from the University of Aachen, Germany. He has worked for the Max-Planck-Institut at Duesseldorf, Germany, for the ETH Zuerich, Switzerland, and for ABB Baden, Switzerland. He is now managing director of SENSOPLAN, Germany, an engineering company offering field testing, finite element calculations, design and failure analysis for large machinery.

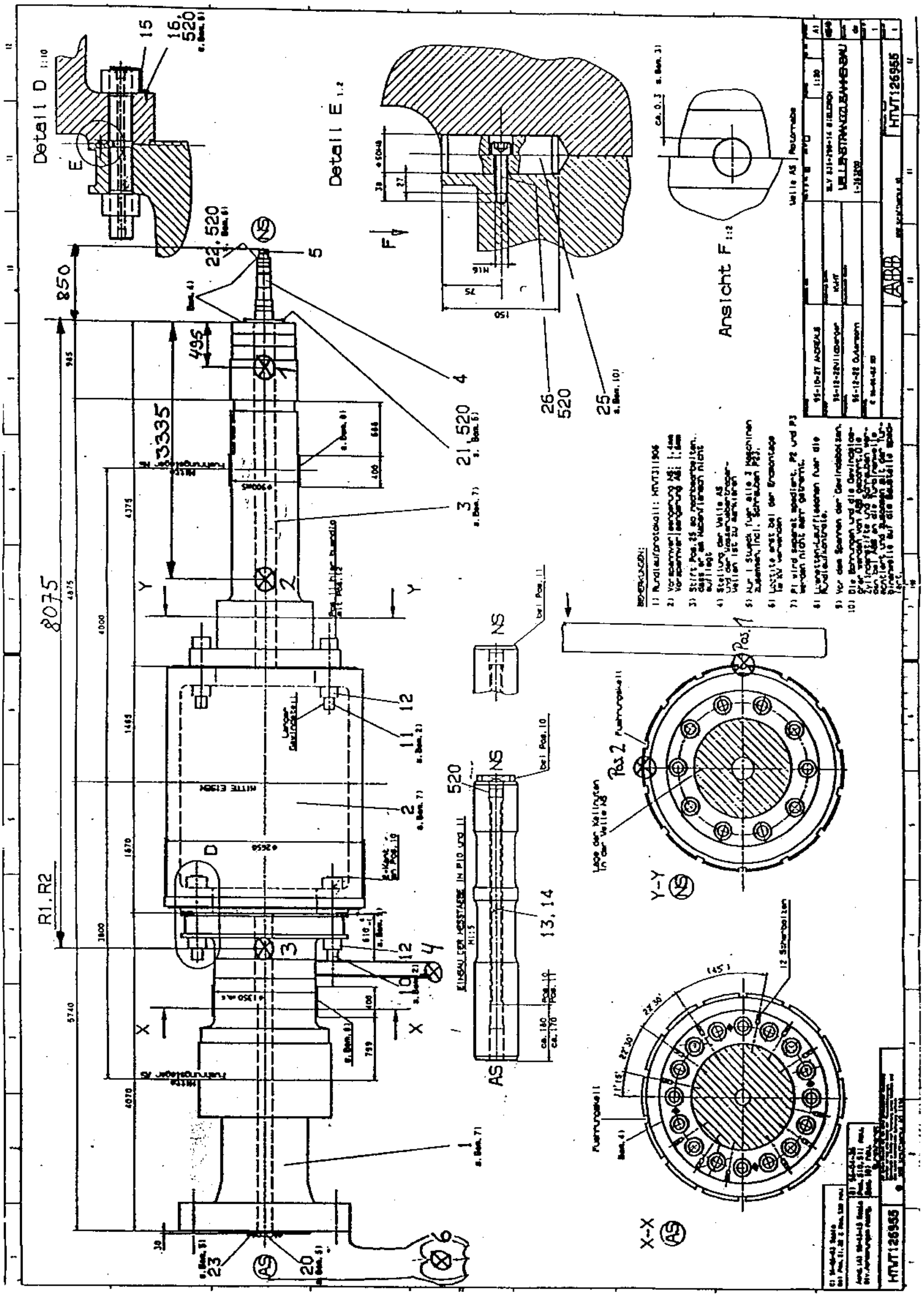
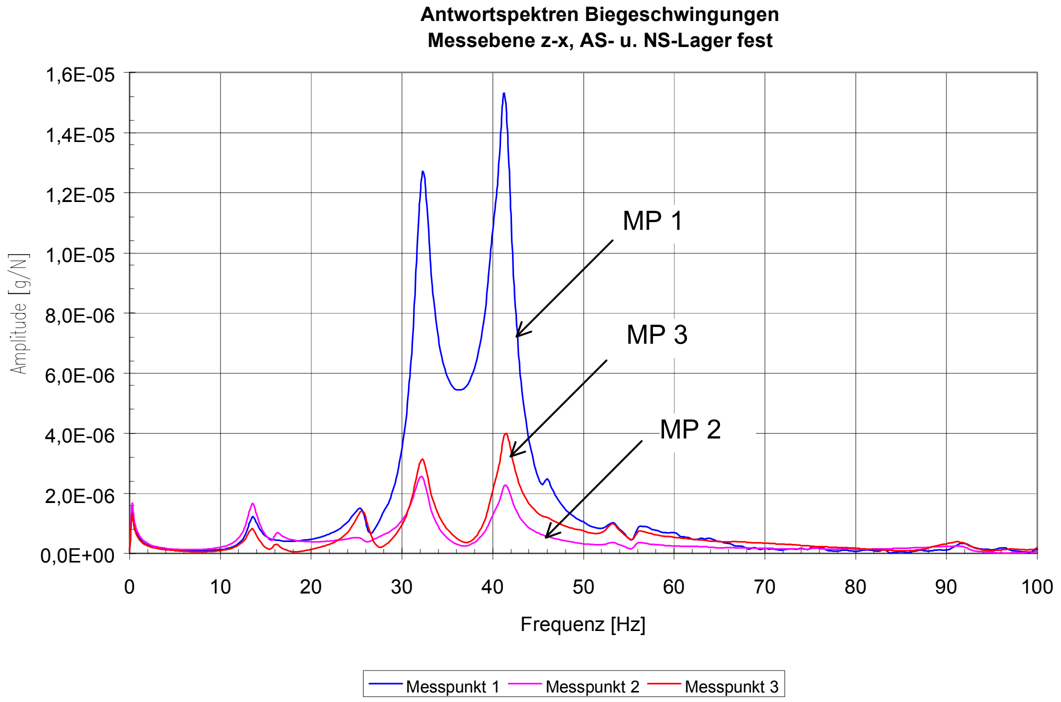


Fig. 1: Arrangement of measuring points 1-6



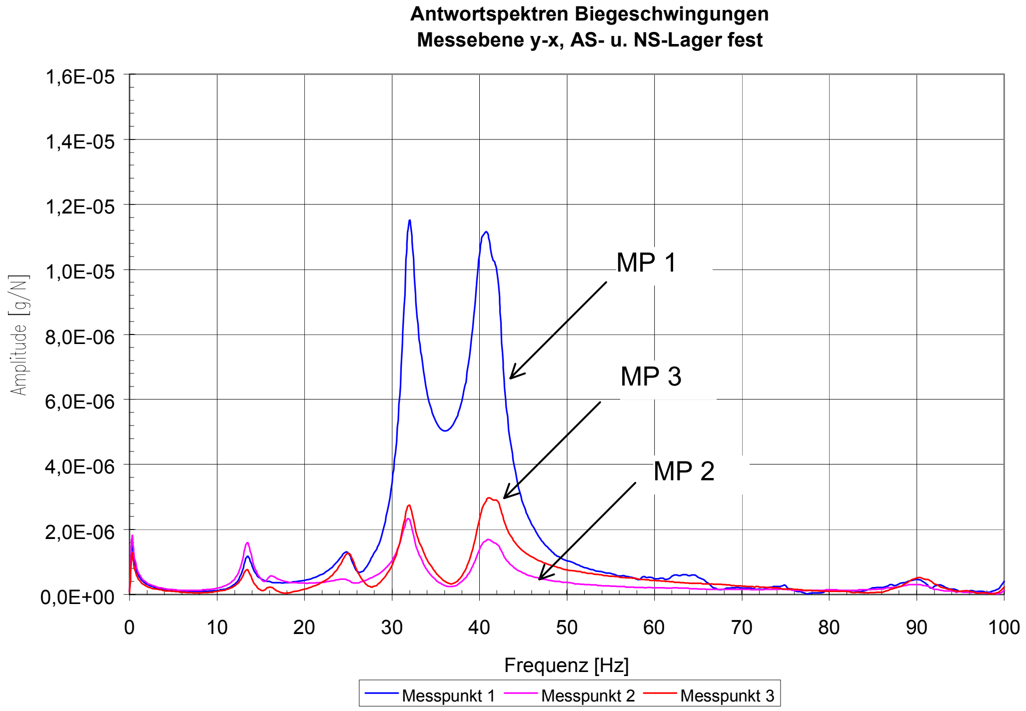
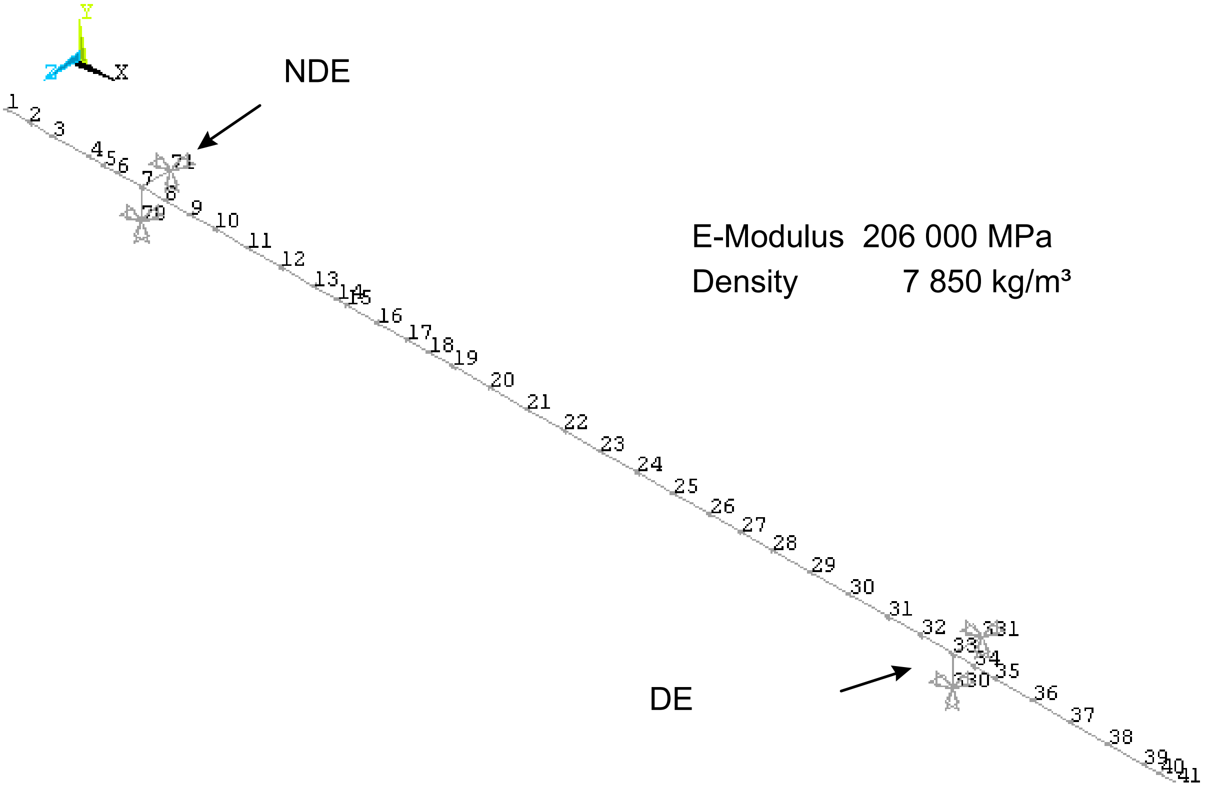


Fig. 2: Measured response spectra for bending, DE and NDE bearing fixed, top: measuring plane z-x, bottom: measuring plane y-x



E-Modulus 206 000 MPa

Density 7 850 kg/m³

Total mass: 528 012 kg

Total moment of inertia: 1 545 809 kgm²

Fig. 3: FE-model for Bieudron shaft line

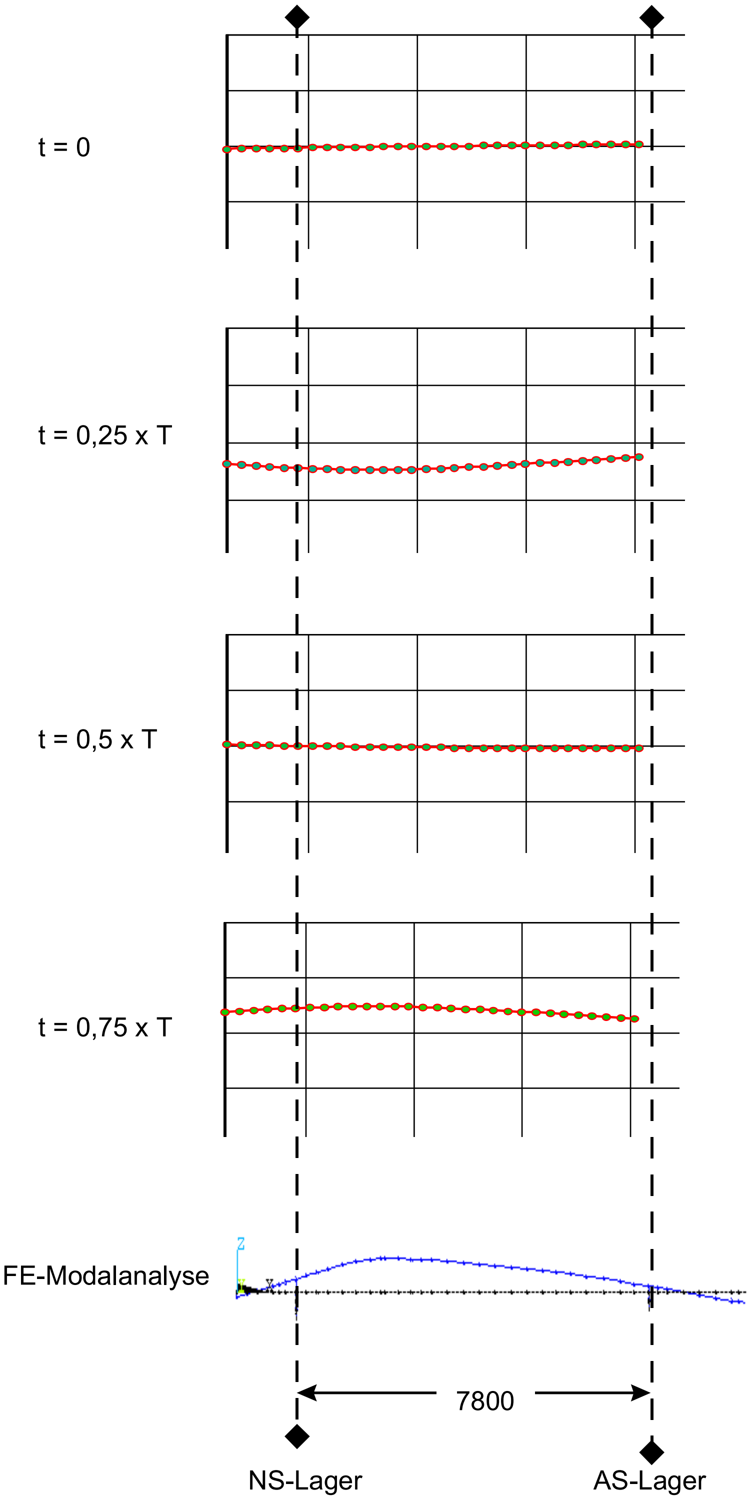


Fig. 4: 1st shaft bending mode; 13.5 Hz: measured (circles) and with ANSYS calculated mode shape (bottom)

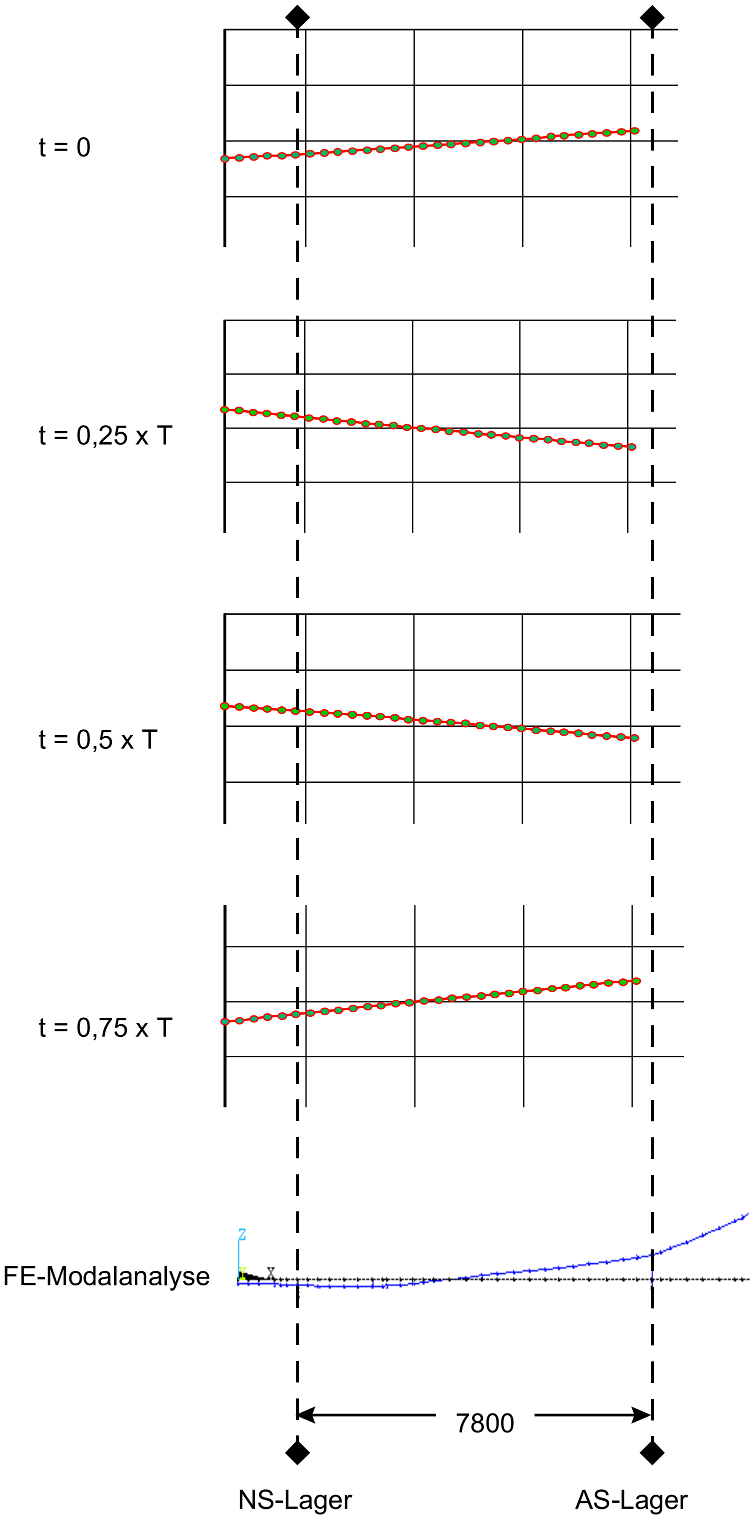


Fig. 5: **2nd shaft bending mode; 25.3 Hz: measured (circles) and with ANSYS calculated mode shape (bottom)**

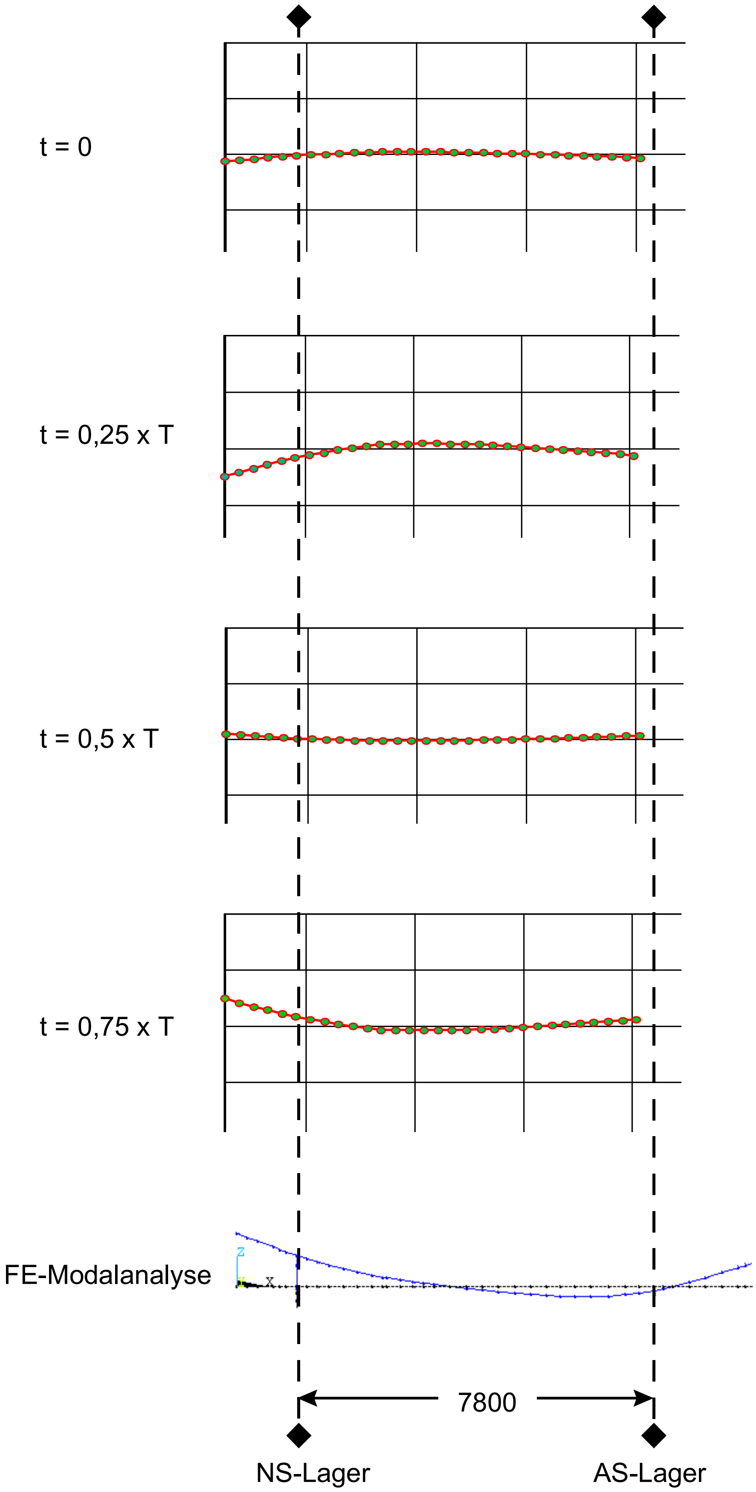
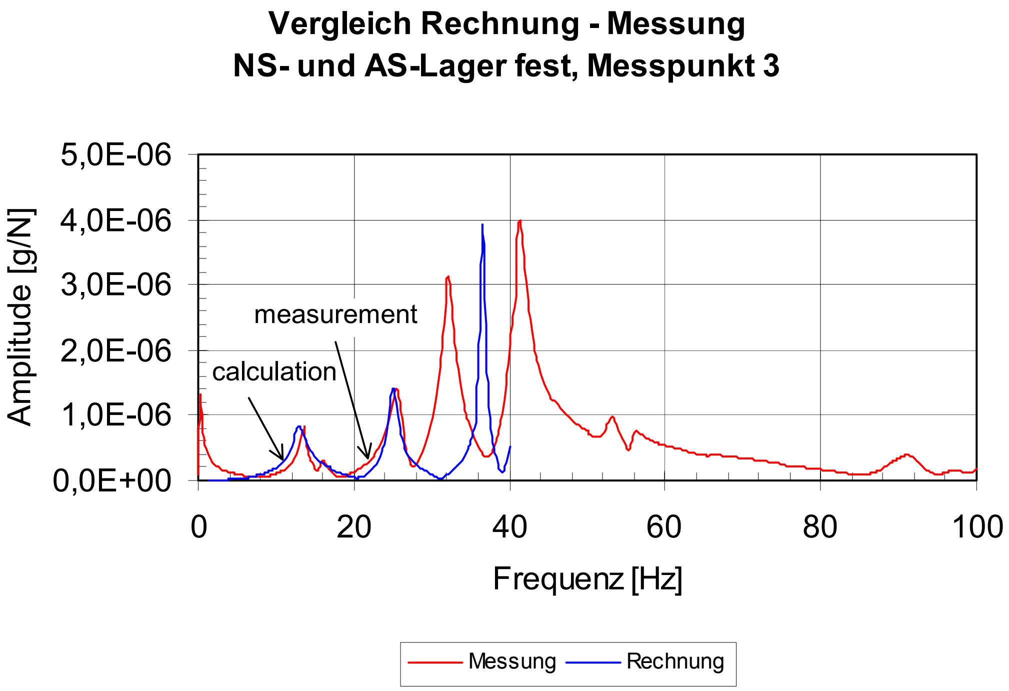


Fig. 6: 3rd shaft bending mode; 41.5 Hz: measured (circles) and with ANSYS calculated mode shape (bottom)



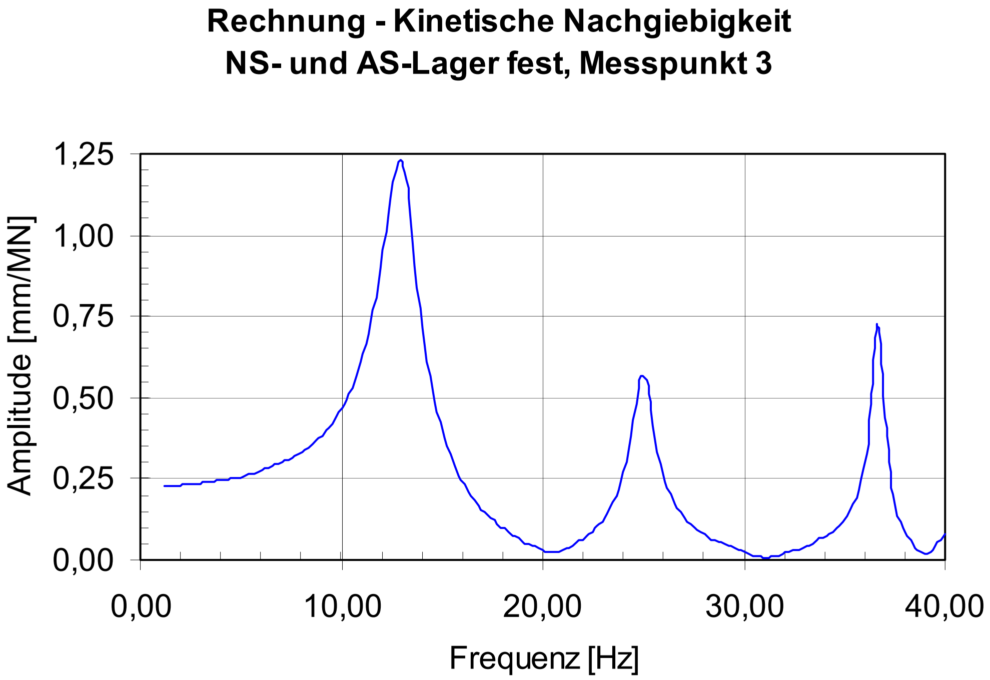


Fig. 7: Comparison of measured and calculated acceleration spectrum (top) and calculated kinetic elasticity (bottom)