

## DESIGN OF EQUIPMENT SUITABLE FOR MEASURING THE NATURAL FREQUENCY OF A ROTATING SHAFT

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**Abstract:** In this paper, the construction and operation of a measuring bench for measuring the vibration state of a rotating shaft and the evaluation of the measurement results are presented. The design of the measuring bench was aimed at measuring the vibration generated by the excitation effects on the shaft. The effect of the imbalance and the position of the disc on the vibration characteristics is shown in diagrams. It is also observed that at higher speeds the vibration plotted shows an irregular shape. This is due to the excitation effects on the whole structure.

**Keywords:** *natural frequency, critical speed, shaft*

### 1. INTRODUCTION

It is advisable to test the rotating parts of machines used in the technical field with experimental and practical methods. Energy accumulates in these components, and these energies force the system to vibrate. In the alternating evolution of potential and kinetic energies, such as in the case of vibration of a system or a structural element, the frequency of the excitation effect determines the speed of the energy transitions. Depending on the excitation effects on the drive chain, the amount of stored energy can be increased. The time-varying excitation effects cause a vibrating motion of the structure, which is damped either by the material or by the damping of the structure.

The vibration amplitude of the drivetrain depends on the excitation effects, the degree of damping, and the proximity of the excitation to the natural frequency. The excitation effects are caused by the operating characteristics of the drive chain (e.g., universal joint or any reciprocating element) and manufacturing or assembly error causing the eccentricity of the rotating elements. Resonance can be avoided by adjusting the eigenfrequencies of the drive train and excitation frequencies far from each other.

The purpose of building the test bench was to measure the effect of excitation on the vibration level. A test bench appears to provide measurement results in operation, is subject to mischaracterization in practice, and effectively reduces the corresponding capabilities. This provides a visual representation of the conflicting placement that allows for practical use and results in satisfactory accuracy.

When designing rotating shafts, care must be taken to reduce vibrations, considering their sources. In addition, the intensity of vibration, the critical speed of the shaft, stability and other parameters affecting the properties of the system must be considered. The effects of the mentioned parameters should be examined especially at the critical speed, since the vibration properties of the system change significantly, which can cause damage to the shaft, premature failure of rotating parts and bearings [1].

Vibrations affecting the service life of shafts include torsional vibrations of varying degrees. High levels of torsional vibration can damage or cause rotating equipment to fail, causing costly downtime. Comprehensive torsional vibration analysis is the typical method for designing a torsional system that eliminates such problems. Torsion system design requirements are defined by various standards; however, a certain degree of uncertainty is always present in analytical data, modelling techniques, and excitation and damping assumptions [2].

The predictive calculation of the torsional natural frequencies of the shaft is performed to avoid torsional resonance problems. However, it is often delicate to select the appropriate excitation frequencies and to determine the modal damping factors that must be considered in the calculation. A further inaccuracy is caused by neglecting the elasticity of the bearings in the boundary conditions, which makes the results uncertain. In practice, it is therefore often useful to experimentally measure the torsional natural frequencies to validate the calculations [3].

Every elastic system is characterized by its own period of oscillations, which is determined by its stiffness and its own mass. If such a system is subjected to forced vibrations because of forces and torques changing due to unbalanced masses or other reasons, and if the frequency of change of these forces and disturbing moments is equal to the frequency of the vibrations or a multiple thereof, then the amplitude of the vibrations increases rapidly, and resonance occurs. In the case of shafts, longitudinal, torsional, and transverse vibrations may occur depending on the forces and disturbing torsional moments. From a practical point of view, it is most often only the latter two types of vibration, because in general the longitudinal restoring forces are very large, and the amplitude of the longitudinal vibrations is small. In most cases, transverse vibrations occur due to transverse forces, which are periodically repeated [4]–[6].

In practice, there are also tasks where the critical speed of axes with asymmetrically concentrated mass (e.g., gear shafts, main shafts of cone crushers) must be determined [7].

Analytical methods may also be suitable for testing torsional vibrations. Two methods were developed to examine the power transmission shaft of a ship propeller, of which the shaft line was modelled as a two-mass system in the first, approximate method. In the second procedure, the multi-degree-of-freedom problem of the entire system was solved using the Rayleigh-Ritz method. The outlined analytical procedures can be used to estimate shaft torsional vibrations in the conceptual design phase, as well as for equipment already in operation [8].

Torsional vibrations are inherently present in all rotating drivetrains. Torsional vibrations can be significantly amplified in resonant conditions. A typical method of reducing torsional vibration, especially at resonance, is to modify the torsional natural frequencies by designing the component. In general, a simple way of modification is to adjust the torsional stiffness of the elements incorporated in the drive [9].

## 2. CONSTRUCTION OF THE LOAD BENCH

Based on the studies and research reviewed in the introduction, the examination of the critical speed of the rotating shafts and the various vibrations of the shafts is an extremely important task. The experimental measurements were carried out on the load bench shown in *Figure 1*. As shown in *Figure 1*, a disc can be mounted on the shaft along the longitudinal axis, which can be fixed in any position on the shaft. There is a  $15 \times \text{Ø}150 \times \text{M8}$  threaded hole on the disc, with the help of which the mass mounted on the rotating shaft can be placed, because of which the moment of inertia and imbalance of the system can be changed.

The mass and moment of inertia of the disk, as well as its location on the shaft, are parameters that influence the natural frequencies of the shaft. The imbalance of the disk ensures the excitation effects when the shaft rotates.

The test bench is driven by a 300 W electric motor, the speed of adjusted using a variable frequency drive (VFD) in the range of 0–5000 rpm. The load of the drive chain is provided by a car generator with a power of 500 W, which can be increased with 3 additional loads of 25 W in discrete steps. When the test bench is in operation, a Hottinger Baldwin Messtechnik Ltd. data acquisition system ensures the registration of measurements and the collection of measurement data. The basis of the measurement is a Spider8 device, which is a multi-channel PC measuring electronics for the computerized collection of parallel, dynamic measurement data. Spider8 contains everything needed for measurement in a compact design.

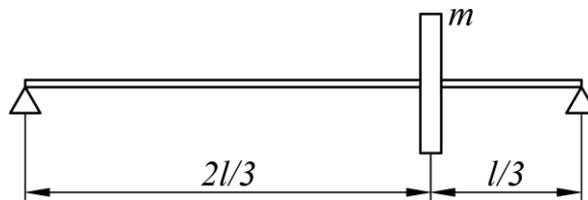


*Figure 1. The design of the measuring bench and the measurement data collection system*

Such a measurement system does not require additional connection and wiring installations, nor large configuration presets. The processing and direct display of the measured data is made possible by a CATMAN EASY measuring software. The system offers the user several prepared measurement programs with evaluation diagrams, which speeds up the evaluation of the measured data. The bending vibration of the shaft placed on the load bench can be measured in two ways. In one method, the values occurring during loading can be measured along three axes with a piezoelectric accelerometer, which is fixed on the two end plates of the stand. In the other measurement method, a complete bridge created with 4 strain gauges measures the changes in shape due to vibration. During measurement, depending on the angular velocity relative to the natural frequencies of the shaft, the transducers generate an electrical signal proportional to the amplitude of the vibration. The natural frequencies of the shaft can be measured by increasing the speed of the shaft and checking the amplitude of the vibration. The natural frequency (or critical speed) is the angular velocity at which the maximum value of the vibration occurs.

### 3. MEASUREMENT RESULTS

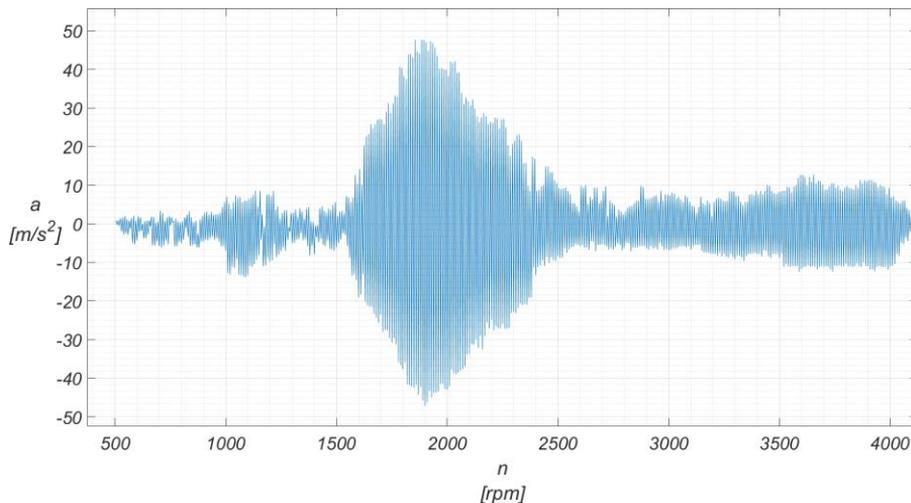
The measurements were carried out in various arrangements, the direction  $x$  acceleration of the piezoelectric accelerometer, as well as the magnitude of the vibration amplitude as a function of the speed were measured during the measurements. The piezoelectric accelerometer and strain gauges were fixed so that their measurement direction was the same (direction  $x$ ).



**Figure 2.** Schematic figure of the measurement arrangement I

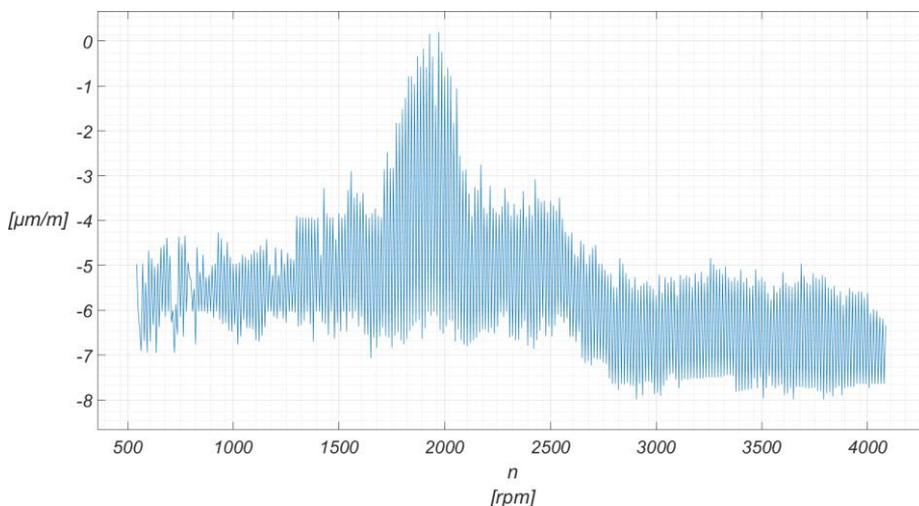
#### 3.1. Measurement result I

In this measuring arrangement, the disk was fixed at  $1/3$  of the length of the shaft (Figure 2), the fastening elements were installed in the threaded holes of the disk, thus the assembled mass of the disk was 300 g. The measurement result shows that the maximum vibration acceleration reaches its highest value at 1950 rpm. The magnitude of the vibration amplitude can be inferred from the electrical signal provided by the strain gauge stamps proportional to the displacement.



**Figure 3.** The vibration acceleration as a function of the speed (Measurement I)

Figure 3 shows the measurement result of the vibration acceleration as a function of the speed. The measurement was performed by fixing the disc at 1/3 of the shaft length and removing 3 fasteners from the disc, which caused the disc to be unbalanced and resulted in reduced weight. The figure shows that due to the reduced mass, the natural frequency was generated at a higher frequency (~2000 rpm), and due to the increased excitation effects, the vibration amplitude decreased minimally after the critical speed was exceeded.



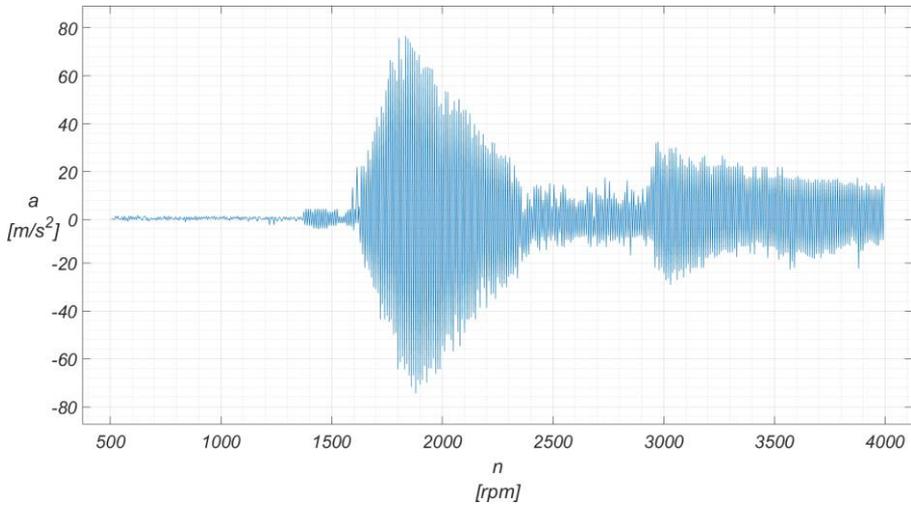
**Figure 4.** Vibration amplitude as a function of speed (Measurement I)

Figure 4 shows the measurement result of the vibration amplitude as a function of the speed. At higher revolutions the measurement result shows an uncertain

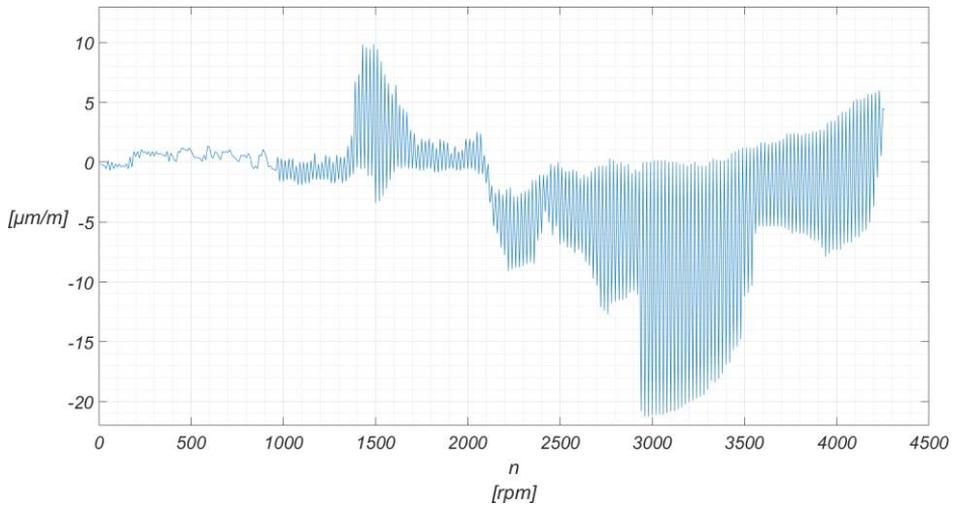
vibration range. The reason for this may be that the whole structure was affected by the excitation effect.

### 3.2. Measurement result II

In this case, the arrangement of the measurement is the same as the measurement described in the previous case. However, the measured system differs from the previous one in that the disc here is unbalanced and weighs 290 g. *Figure 5* shows that the value of the maximum vibration acceleration already appears at ~1800 rpm and a slight increase in deflection can be observed from ~3300 rpm.



**Figure 5.** The vibration acceleration as a function of the speed (Measurement II)



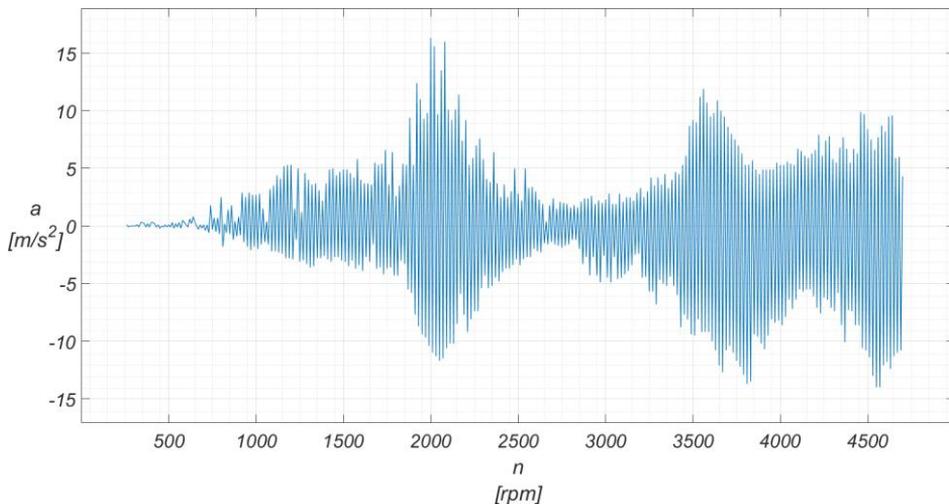
**Figure 6.** Vibration amplitude as a function of speed (Measurement II)

Figure 6 illustrates the measurement result registered by the measuring bridge formed by the strain gauge stamps for the same measurement arrangement. It can be observed that even with this configuration, the largest vibration deflection occurs at  $\sim 1800$  rpm. Compared to Figures 3 and 4, the measurement results show higher and more irregular vibration acceleration and vibration deflection, which can be explained by the fact that the pulley became unbalanced when the fasteners were removed, which significantly increased the vibration of the shaft. It can also be seen in this figure that at  $\sim 3300$  rpm the measured maximum value at which the vibration peaks occur is displayed.

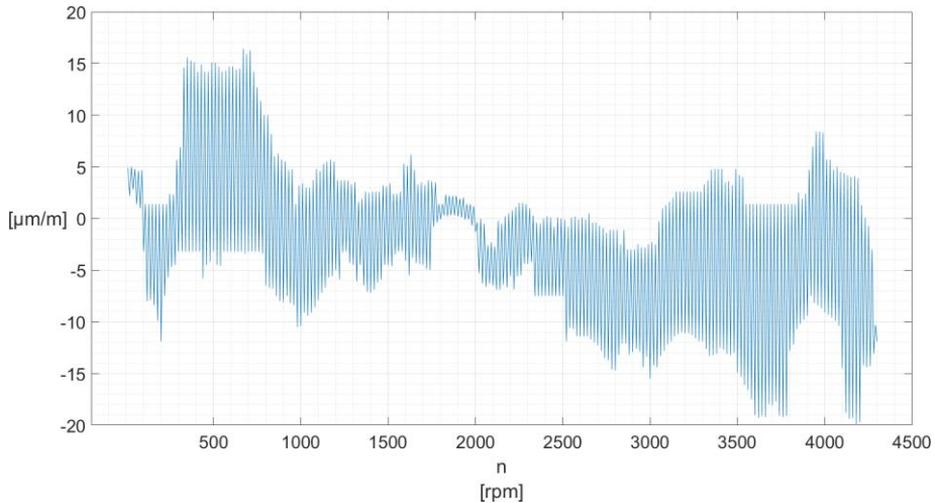
### 3.3. Measurement result III.

In this measurement setup, it shows acceleration in the x direction as a function of speed. The vibration maxima are clearly visible in Figure 7. The graph recorded by the piezoelectric accelerometer shows that there are several maximum accelerations in the range of  $\sim 3500$  rpm and  $\sim 4700$  rpm. The measurement results detected by the strain gauges are illustrated in Figure 8.

It can be observed that the diagram registered by the strain gauges gives a slightly more accurate picture of where the frequencies are where the system tends to resonate. The increase in acceleration and vibration values already occurs at lower speeds. From this, it can be concluded that the eccentric disc is located half-way between the two clutches, affecting the vibration conditions of the shaft, as when it was placed closer than one third to the bearing bracket. The displacement – speed diagram shows that the vibration peaks at low speed already appear around  $\sim 1000$  rpm.



**Figure 7.** The vibration acceleration as a function of the speed (Measurement III)



**Figure 8.** *Vibration amplitude as a function of speed (Measurement III)*

#### 4. SUMMARY

In this article, the construction and operation of a measuring bench suitable for measuring the vibration state of a rotating shaft, as well as the evaluation of the measurement results were presented. The purpose of the measuring bench was to measure the vibration caused by the effects that excite the shaft. The effects of unbalance and disc position on vibration characteristics are shown in the diagrams. It can also be observed that at higher revolutions the depicted vibration shows an irregular shape. The reason for this is that the whole structure is affected by the excitatory effects.

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