Evaluation of the dynamic behavior of a Pelton runner based on strain gauge measurements

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Abstract. A reliable mechanical design of Pelton runners is very important in the layout of new installations and modernizations. Especially in horizontal machines, where the housing is not embedded into concrete, a rupture of a runner bucket can have severe consequences. Even if a crack in the runner is detected on time, the outage time that follows the malfunction of the runner is shortening the return of investment.

It is a fact that stresses caused by the runner rotation and the jet forces are superposed by high frequent dynamic stresses. In case of resonance it even can be the dominating effect that is limiting the lifetime of a runner. Therefore a clear understanding of the dynamic mechanisms is essential for a safe runner design.

This paper describes the evaluation of the dynamic behavior of a Pelton runner installed in a model turbine based on strain gauge measurements. Equipped with strain gauges at the root area of the buckets, the time responses of the strains under the influence of various operational parameters were measured. As a result basic theories for the jet bucket excitation were verified and the influence of the water mass was detected by evaluating the frequency shift in case of resonance. Furthermore, the influence of the individual bucket masses onto the dynamic behaviour for different mode shapes got measured.

1. Introduction

Hydropower machines are designed to be a reliable and profitable investment for the power utility and its owners. Their application varies from base load operation to fast regulation tasks to stabilize the network frequency. It is therefore essential to understand all physical phenomenon that influence the structural integrity [1]. Especially the rotating equipment bear a number of dynamic effects that can strongly limit the lifetime of the machine.

The buckets of a Pelton runner, for example, experience a strong load change during the working cycle. On top of the load generated by the jet bucket excitation severe additional stresses can appear if harmonics of the jet passing frequency are close to the natural frequencies. During the design phase it is therefore essential to check for the margin against a possible resonance scenario and to change the design if the safety distance is not sufficient.

However, this presumes, that all the parameters influencing the natural frequencies are known and well understood so that changes in the geometry can be properly evaluated. The aim of the measurements described in this paper were exactly for this purpose: to evaluate theories and models on their validity and to check the influencing parameters so that they are considered correctly during the design phase.
The current theories and design limits are based on theoretical and numerical models, which are checked on the results of experimental modal analyses in prototype runners including the shaft line. Refinements on the theories were made with strain measurements in rotating prototype runners [2] and [3].

However, as the prototype runs at constant speed, the excitation frequency is limited to only discrete values depending on the number of jets in operation. An analysis on the damping of the structure is therefore only possible with measurements during the coast down of the unit. Whether these results are also representative for the regular operation with water is unclear. Moreover, investigations on the influence of the water mass onto damping and shift in the resonance frequency is not possible at all.

Because of this, mode tests with an instrumented integral runner had been carried out. This allows to vary the excitation frequency and shape as well as the water flow to evaluate the damping of the structure and the shift in the natural frequency. Furthermore, the influence of individual bucket masses had been measured to check the possibilities of mistuning onto the stress amplitudes in case of resonance.

In the following the test rig setup and the instrumented runner are explained. After that, the results of the experimental modal analyses are shown and compared to numerical predictions, before some results of the stress measurements for the influence of the water mass and the mode shapes are discussed in more detail.

2. Test setup

The measurements were carried out on the vertical model test rig located in the corporate research center of Voith Hydro in Heidenheim, Germany. This test rig is commonly used for hydraulic investigations and model acceptance tests. For the dynamic measurement of the stresses, however, adaptions to the setup were necessary in the rotating equipment.

![Figure 1. Vertical shaft test rig for the measurement of the dynamic stresses in a model runner. Below the runner a rotating housing is mounted, containing the transmitter equipment.](image-url)

Figure 1 illustrates the setup used for the measurements. The vertical shaft arrangement consists of the eddy current brake supported in a hydrostatic bearing on the top, the Pelton runner mounted on the rotating part of the brake, the transmitting system attached below the
runner and a 6 jet model turbine around the runner holding the receiver system on the very low part of the rig.

The runner itself used for the measurements was made as an integral runner out of aluminum bronze. This material was chosen to bring the natural frequencies of the buckets into a region possible for a reasonable sampling rate during the measurement. Special faces were applied on the bucket outside onto which additional weights got mounted during the measurement phase.

![Figure 2. Integral model runner manufactured out of aluminum bronze](image)

For the stress measurements the model runner had to be instrumented with strain gauges in the root area, as shown in Figure 3 on the left picture. All 20 buckets were eqipped at the point, where the maximum stress amplitudes were expected. To ensure that each miniature strain gauge got located at exactly the same position a template was created that also served as a support tool for the application. The strain gauges used for this application were encapsulated and water proof to protect them from straying water. The cables of 5 adjacent buckets were collected, protected with epoxy resin and led into the attached housing containing the transmitting system.

![Figure 3. Application of the strain gauge onto the root area (left). Cables collected for 5 sensors and protected with epoxy resin (middle). Measurement system installed into turbine ready for test (right).](image)

The data transmission system sampled the strain gauge signals with a rate of 5 kHz. Inside the sender system it got further amplified, converted into frequency and prepared for transmission. Also the calibration with a high precision shunt is done in the rotating system. The data are then transmitted to the data aquisition system. In here the signals from the rotating system were combined with the speed measurement located on the shaft and a trigger signal to record the position of the runner. The right picture in Figure 3 shows the instrumented runner together with the measurement equipment.
3. Preparatory Investigations

Many dynamic effects at Pelton runners can be detected by the measurement of resonance curves. Resonance curves can be determined by shifting an exciting frequency gradually over a natural frequency of the runner. Practically the measurement starts at a defined rotational speed and is increased until the resonance condition is passed. To identify resonance curves a detailed knowledge of the dynamic behavior of the Pelton runner is required. So before the measurement has started investigations about the natural frequencies and the corresponding excitation frequencies have been done.

Already during the development of the runner geometry numerical modal analyses have been done simultaneously to identify the natural frequencies and the corresponding mode shapes. The attendant calculations guaranteed that the runner has suitable dynamic properties. So even before the strain gauge measurements have been started expectations could be formulated.

Before the runner was mounted into the test rig an experimental modal analysis was carried out. This experimental modal analysis was necessary to adjust the calculated frequencies to the results determined for the real modelrunner. To detect the influence of the shaft and the measurement equipment an experimental modal analysis was repeated at the mounted state. Figure 4 shows the mounted configuration with buckets equipped with uniaxial accelerometers.

![Figure 4. Experimental modal analysis at the mounted Pelton runner](image_url)

Figure 5 and 6 show a typical example of a k=5 mode. The mode shape of the finite element calculation in Figure 5 can be detected easily. The corresponding mode shape from the experimental modal analysis in Figure 6 is in good agreement.

For the measurement of resonance curves a detailed understanding of the expected resonance frequencies for the different modes is necessary. For this, a numerical resonance check was made, in which a fixed excitation pattern for a fixed rotational speed is analysed on its fit into the spread angle \( \frac{360}{k} \) of the different mode shapes. Into this analysis the natural frequencies determined in air are plotted to check the distance to possible resonance scenarios. Figure 7 shows the natural frequencies of the numerical and experimental modal analysis. Additionally the excitation frequencies and mode shapes for a symmetric 3 nozzle operation for a fixed rotational speed are included. Thereby the black crosses show the excitation frequencies on the vertical axes and the appendant mode shapes of excitation on the horizontal axes. It is obvious that the constellation of Figure 7 would cause resonance by exciting the runners k=2 mode at an excitation frequency of around 1450 Hz. To measure resonance with different excitation modes the rotational speed has to be increased until the corresponding excitation frequency fits
to the resonance frequency. Based on the knowledge of the dynamic properties of the runner an adequate measurement program was designed.

4. Results

4.1. Resonance Examination

Resonance occurs only when the natural frequencies of the runner fit to the excitation frequencies and additionally when the excited mode shape of the runner fits to the mode shape of the excitation. The mode shape of the excitation is a function of the number of buckets, the nozzle configuration and the rotational speed of the runner.
Figure 8 and 9 show two strain time signals of one bucket in six nozzle operation for one revolution of the runner. In operation out of resonance the strain signal follows the load of the jets smoothly. The influence of harmonics is marginal. In resonance condition of Figure 9 an overlay of a dynamic strain signal to the strain signal of the out of resonance operating condition is visible.

Figure 8. Time signal of strain at bucket root at out of resonance operating condition

Figure 9. Time signal of strain at bucket root at resonance operating condition

Figure 10 and 11 show the corresponding spectra of the strain time signals. Here a signal length of 6.5 seconds is used. The spectra show the jet passing frequency and its harmonics which fits to the assumption that the runner is excited by a periodic force time signal. The first frequencies describe the strain signal for the working cycle of one bucket. In the out of resonance operating condition of Figure 10 at the 16th harmonic small dynamic effects can be seen. The low strain amplitudes at these frequencies characterize the low dynamic effects seen in Figure 8. Compared to the amplitudes of the lower harmonics, these effects do not have a significant influence on the time signal. Figure 11 shows the spectrum in resonance condition. As seen in Figure 9 a large increase of the strain amplitude at the 16th harmonic can be detected. Compared to the lower harmonics the dynamic strain amplitude is even higher.

Figure 10. Frequency spectrum of time signal shown in Figure 8

Figure 11. Frequency spectrum of time signal shown in Figure 9

Visible amplitudes acting close to resonance frequencies can become fatigue relevant. A strain amplitude at the 16th harmonic, which is comparable to the strain amplitude of the jet passing frequency, would increase the damage rate significantly. This result shows the relevance of experimental modal analyses and resonance checks in design process to get a safe runner design.
4.2. Influence of Water Mass

Because of theoretical considerations it is expected that in operation different effects cause a frequency shift of the natural frequencies. The influence of the water flowing through the buckets as reason for the shift is a theory used to describe this effect. An increase of the bucket mass by the water would result in a reduction of the natural frequencies and so in a reduction of the rotational speeds where resonance occurs. This measurement gives information for actual research programs how high these frequency shifts are.

To get that information several measurements at different rotational speeds have been done. Two significant results are shown in Figure 8 and 9 already. After the measurement the strain amplitudes have been determined and resonance curves could be derived. To detect the influence of the water mass the discharge was modified and the measurement was repeated. Afterwards the frequency shift of the maximum strain amplitudes were compared to the corresponding natural frequency of the runner in air, determined in the modal analyses. Figure 12 shows one typical result.

![Influence of water mass](image)

**Figure 12.** Shift of natural frequency at different runner loads caused by considered water mass

The resonance curves of Figure 12 show the dynamic amplitudes close to a resonance frequency. The vertical line shows the natural frequency at standstill derived by an experimental modal analysis. With increasing discharge a reduction of the natural frequency can be detected. At minimum load the position of the resonance maximum is close to the considered natural frequency in air. So with minimum load no significant shift of the natural frequency can be measured. With increasing discharge a part of the water mass flowing through the bucket can be added to the bucket mass. This effect leads to the measured natural frequency reduction as shown in Figure 12. A compensating effect in between water mass and increased stiffness because of rotation, as described by [4] was not detected.

At two rotational speeds Figure 12 also contains the strain amplitudes at the 16th harmonic of Figure 8 and 9. The dynamic amplitudes of Figure 8 can be seen on the left side of the strain peaks at optimum load. Figure 9 is represented in resonance directly at the highest amplitude of optimum load.

4.3. Influence of Mode shapes on Resonance Frequencies

It is well known that modes with low number of node lines can be detected easily in an experimental modal analysis. Modes with higher number of node lines are always difficult to
measure. Often these difficulties result from small inaccuracies during manufacturing. This effect is also called mistuning. Figure 7 shows that the natural frequencies with a higher number of node lines could not be detected during the experimental modal analysis. So the missing natural frequencies have been replaced by the calculated natural frequencies during the preparation for the measurement plan.

The following explanation shows the influence of a k=4 mode and a k=10 mode excitation on the strain amplitudes measured at the runner. At a k=4 mode the buckets are excited by four periods of a sine function for one revolution of the runner. As a result, the neighboring buckets have only a small shift in the phase angle. A runner with 20 buckets excited by this k=4 mode shows a phase angle of 72° between neighboring buckets. In that case small uncertainties in the bucket mass or stiffness have a minor influence on the global mode shape.

At a k=10 mode shape the single buckets are excited out of phase. The phase angles of the neighboring buckets are 180°. In this situation differences in bucket mass or stiffness can be seen more clearly. Because the buckets are not in a compound this effect can mismatch the steady shape of a k=10 mode. All buckets have different natural frequencies and will get in resonance at different excitation frequencies.

Figure 13. Frequency spectrum at k=4 excitation for different rotational speeds and buckets

Figure 14. Frequency spectrum at k=10 excitation for different rotational speeds and buckets
Figure 13 shows an example of a $k=4$ excitation. In this analysis the response of two buckets for two different excitation frequencies is shown. It can be seen that the dynamic amplitudes at the 16th harmonic increase synchronously with the increasing frequency. In contrast Figure 14 shows a $k=10$ mode excitation at the 15th harmonic. The response for the same buckets as before are analyzed. It can be detected that there is a measurable shift in the frequency for the two buckets where the maximum amplitude occurs. While bucket 1 is in resonance already at 933 rpm bucket 5 shows resonance later at 937 rpm, when bucket 1 is already out of resonance. The results of the measurement verify that an increasing number of node lines comes along with an increased influence of the local mass variations and stiffness on the mode shapes. For the tested runner these mass variations are caused by the manufacturing uncertainties.

5. Conclusion
For a reliable layout of hydropower machines the relevant physical effects that influence the structural integrity have to be considered during the design. For Pelton runners the bucket dynamics and its change during the operation is one effect that may lead to catastrophic failure. Prototype measurements are an important source for evaluating the theories and tools for the layout. However, its possibilities concerning excitation are limited.

Because of this, strain measurements on a rotating Pelton runner in a model turbine setup were carried out. With this, the excitation was varied in frequency, energy level, shape and discharge to allow a detailed analysis of various influencing parameters. In addition, numerical and experimental modal analyses were used to calibrate the numerical models and to determine the resonance regions for the different mode shapes.

The influence of the water mass is one of the effects shown in this paper. A clear shift in the resonance frequencies was observed in the measurements. In addition it was found out how the individual bucket masses influence the resonance frequencies of the buckets depending on the mode shapes.

The above described measurements are another important step to include all relevant physical phenomena into the design process. Further analyses of the measurement data will serve to gain a more detailed understanding of the dynamic behaviour and to estimate the distance to critical operating conditions, which will improve the design stage and ensure the quality of the investment of our customers.

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