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Numerical assessment of parameters influencing the modal response of a Kaplan turbine model

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Abstract. The present study intends to investigate numerically the impact of: i) the fluid added mass, ii) the turbine rotational speed and iii) the variation of the runner blade bearing stiffness due to the turbine water head on the modal response of a reduced scale Kaplan turbine. The impact of each variable has been quantified by means of a series of numerical modal analyses of the turbine in vacuum and in water, and at rest and rotating. It has been found that the natural frequencies of the Kaplan turbine model are sensitive to all the variables investigated in the present paper, the added mass being the factor which has the greatest impact.

Key Words: Kaplan turbine runner, modal analysis, rotor dynamics, fluid structure interaction, added mass, structural response.

1. Introduction

The introduction of renewable energy sources such as solar and wind in the electric market causes hydraulic turbines to work longer periods of time at non-optimal operation points than ever before.

In addition, the aim of increasing the power density of runners, the higher number of start and stop cycles and the use of lighter materials to build up new hydraulic turbines has raised concern about the dynamic behaviour of these structures. It is therefore considered indispensable to determine the natural frequencies of turbines accurately during their design stage.

The dynamic parameters which fundamentally determine the natural frequency of a structure are its stiffness and its mass. Assuming the fluid restoring force is negligible, added mass would be the only fluid phenomenon affecting the natural frequencies of a submerged structure.

Traditional techniques used to estimate the natural frequencies and mode shapes in air and water, which are based on empirical coefficients and the expertise of design engineers, are no longer reliable because the added mass effect depends on several parameters which include the shape of the structure, its relative position from boundaries, nearby structures, amplitude, direction of vibration and the local flow condition [1]. Most of the current investigations about the added mass rely on numerical methods.

This paper aims to quantify numerically the impact of the added mass, the turbine rotational speed and the variation of the runner blade bearing stiffness due to the turbine water head, on the natural frequencies of a Kaplan turbine model.



2. Description of the reduced scale Kaplan turbine

The present test case is a Kaplan turbine model which is located at the Vattenfall Research and Development Facility in Älvkarleby, Sweden. The model turbine is a 1:3 reduced scale of an existing prototype in a nearby power station and operates in a closed-loop configuration.

The turbine model is composed of 6 runner blades, 20 equally distributed guide vanes and 18 unequally distributed stay vanes with a runner diameter of 0.5 m. The nominal operating parameters of the turbine model are: i) a head of 7 m, ii) a discharge of 0.71 m³/s and iii) a rotational speed of 696.3 rpm.

In Fig. 1, a general view of the Kaplan model with the bearing details is shown together with the hydraulic circuit. The runner blade bearing, highlighted in red on the left of Figure 1, is a double ball bearing system which allows the adjustment of the pitch angle. The turbine bearing, highlighted in green, is a roller bearing which only restricts the radial displacements. Finally, the inertial wheel bearing, highlighted in blue, is a ball bearing which restricts all displacements, axial and radial.

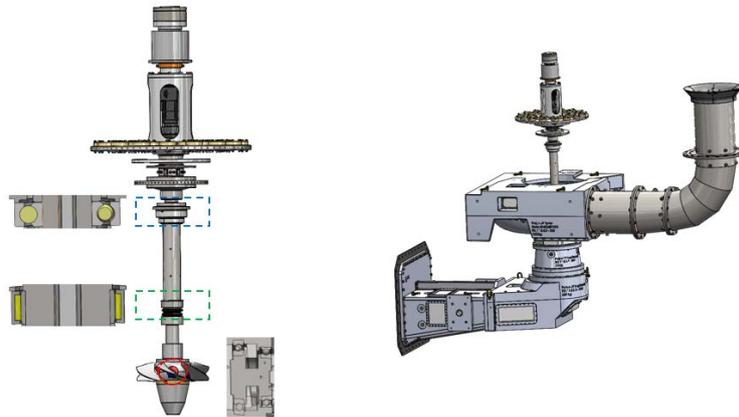


Figure 1. Kaplan turbine with the bearing details (left) and general view of the model (right).

3. Numerical model

The fluid added mass is usually considered an inviscid phenomenon and modelled by a constant coefficient for small motions [2]. In the field of the hydraulic turbines, the use of structural elastic elements coupled with potential flow elements have been successfully used to capture the added mass effect. It has been demonstrated that, for either blades or turbines, the natural frequencies measured experimentally and calculated numerically by considering the fluid as a potential flow are in good agreement [3]. In [4, 5, 6], the natural frequencies of submerged disk-like structures were also calculated with high accuracy using potential flow elements coupled with structural ones.

The pressure wave equation together with the structural equation have been used to carry out the coupled acoustic-structural modal analysis of the Kaplan turbine test stand based on Equation 1:

$$\left. \begin{aligned} [M_s]\ddot{u} + [K_s]u &= [R]p \\ [M_f]\ddot{p} + [K_f]p &= -\rho[R]^T\ddot{u} \end{aligned} \right\} \quad (1)$$

where $[M_s]$ and $[M_f]$ are the structural and fluid mass matrices, respectively, $[K_s]$ and $[K_f]$ are the structural and fluid stiffness matrices, respectively, $[R]$ is the coupling matrix, p is the pressure, ρ is the fluid density and u is the structural displacement.

3.1. Finite element model

Two different numerical models have been built, the first one includes the runner, the shaft and the surrounding water which is contained between the penstock, spiral casing, distributor, runner water passage and draft tube, see left of Fig. 2. The second one only includes the structural components: the shaft, up to the connection with the generator, and the runner, see right of Fig. 2.

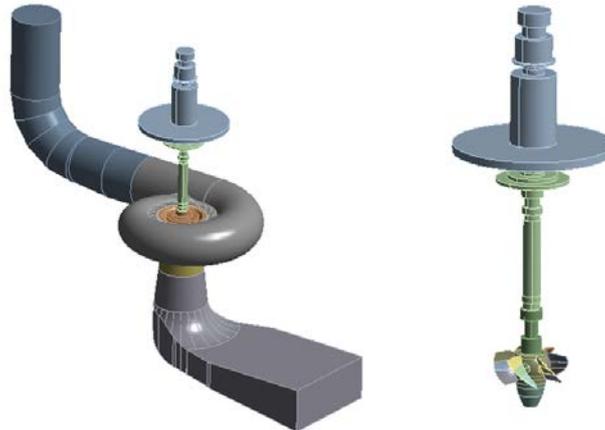


Figure 2. Model of the Kaplan turbine test stand with the rotor and the surrounding fluid domain (left) and only with the rotor in vacuum (right).

3.2. Boundary conditions

3.2.1. Structural boundary conditions

The turbine bearing has been modelled by a connection that allows axial displacements but restricts the radial ones whereas the inertia wheel bearing is modelled by a connection that restricts the displacements in all directions.

The runner blade bearings have been modelled by using a bushing contact whose stiffness value can be modified.

3.2.2. Acoustic boundary conditions

The nodes of the fluid domain in contact with the runner are defined as fluid structure interaction interface, which means that these surfaces satisfy the non-penetrative condition of the fluid in the structure as well as the kinematic and dynamic continuity.

The nodes of the fluid domain in contact with static walls are fixed, thus neither displacements nor absorptions of the acoustic wave energy are permitted.

Finally, the penstock inlet and draft tube outlet have been defined with a prescribed relative pressure of 0 Pa.

4. Results

4.1. Added Mass

The added mass effects have been investigated with two numerical modal analyses of the turbine at rest. One analysis with the turbine in vacuum and another one with the turbine in water, see Fig. 2.

The reductions of natural frequencies have been quantified using the frequency reduction ratio (FRR) coefficient defined by Equation 2:

$$FRR(\%) = \frac{W_v - W_w}{W_v} \quad (2)$$

where W_v is the natural frequency of the structure in vacuum and W_w in water.

It has been considered that the maximum periodic excitation frequency to be taken into account is the guide vane passing frequency which is approximately 250 Hz. Therefore, the frequency range in the present study has been set between 0 and 300 Hz. It is important to notice that within this range there are no runner's natural frequencies. That is why the first family of natural frequencies of the runner has been studied separately.

In Table 1 the natural frequencies of the first 5 modes are indicated in vacuum and water, as well as the corresponding FRRs. It can be observed that the added mass is almost negligible for most of the modes. The invariance between the natural frequencies of the turbine in vacuum and in water can be explained by the fact that the fluid mass affected by the displacement of the rotor is almost negligible compared to the total modal mass. In particular, only M3 is affected by the added mass with a FRR of about 12.5 %.

Table 1. Natural frequencies of the turbine rotor in vacuum and in water with the corresponding FRRs.

	M1	M2	M3	M4	M5
Turbine in vacuum (Hz)	28	55	97	271	281
Turbine in water (Hz)	28	55	85	271	281
FRR (%)	0	0	12.5	0	0

In Fig. 3, the mode shapes of the turbine for the first five (5) modes with natural frequencies below 300 Hz are plotted.

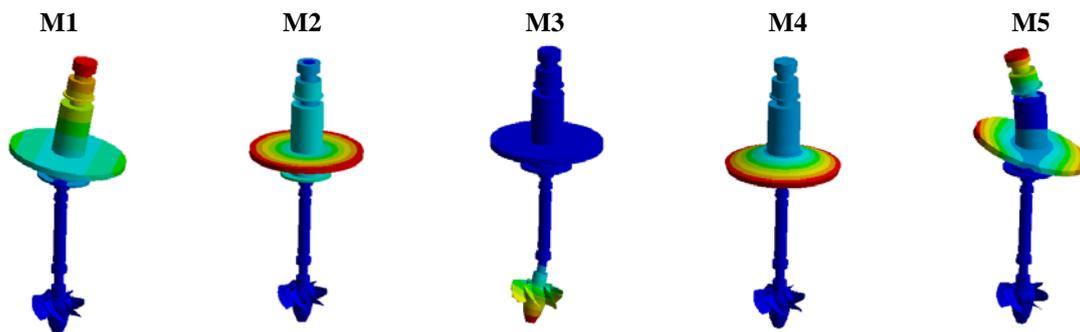


Figure 3. First five (5) mode shapes of the Kaplan turbine rotor.

The natural frequencies of the runner itself have been classified in families in which blades share the same mode shape topology following [7]. In Table 2, the natural frequencies of the first four (4) modes of the bending family are listed.

Table 2. Natural frequencies of the runner modes bending family in vacuum and in water with the corresponding FRRs.

	B1	B2	B3	B4
Runner in vacuum (Hz)	923	924	960	1030
Runner in water (Hz)	466	473	499	585
FRR (%)	49.5	49.0	48.0	43.0

For the natural frequencies with mode shapes mainly affecting the runner, the impact of the added mass is much more important than the ones which have displacement fields affecting both runner and shaft as it can be seen in Table 2 and 1, respectively.

In Fig. 4, the mode shapes of the runner modes bending family are plotted.

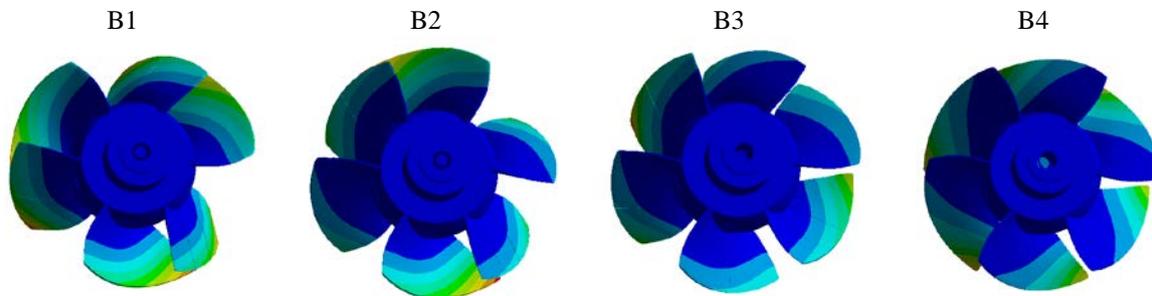


Figure 4. Mode shapes of the banding family.

4.2. Turbine rotational speed

In this section, the impact of the turbine rotational speed on the natural frequencies of the Kaplan model in vacuum is presented. In this case, 10 modal analyses have been carried out when increasing the rotational speed from 0 to 1500 rpm which covers most of the operational range of the test stand. Then, the evolution of the natural frequencies as a function of the rotational speed has been studied.

In Fig. 5, the natural frequencies of M1 and M5, which are most sensitive to the rotational speed, have been plotted as a function of the rotational speed. It can be concluded that the more bent the rotating shaft is, the higher the impact of the rotational speed on the natural frequencies.

In Table 3, the maximum variation of the natural frequencies for each mode shape is indicated. It is interesting to observe that for M1 the split of the natural frequency is symmetrical whereas this symmetry is lost in M5.

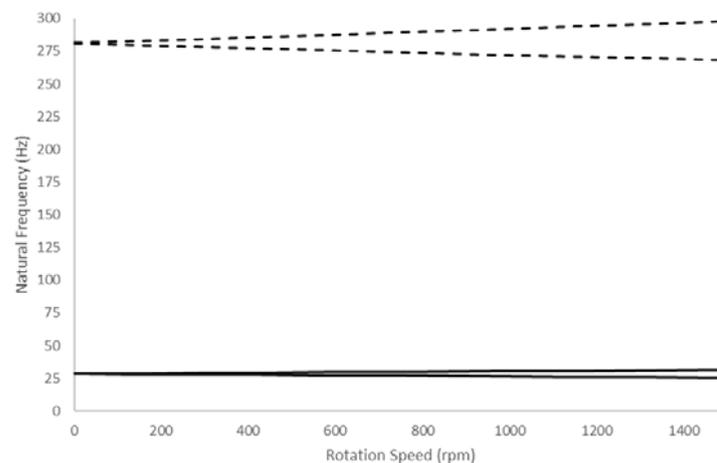


Figure 5. Black lines (M1) and dashed lines (M5).

Table 3. Maximum variation of the natural frequencies of M1 and M5 due to the rotational speed.

	0 rpm	1500 rpm	Maximum Variation (%)
M1	28	25	10.7
		31	10.7
M5	281	268	4.6
		298	6.0

4.3. Runner blade bearing

In this section, the sensitivity of the turbine natural frequencies in water to the variation of the runner blade bearing stiffness is presented. The water head modifies the blade bearing stiffness which alters the modal response of the Kaplan turbine model. Therefore, three modal analyses with blade bearing stiffnesses corresponding to heads of 17.6, 7 and 2.3 mwc have been carried out with 17.6 and 2.3 mwc being the highest and lowest possible head configurations for the turbine model and 7 mwc being the nominal head.

Only the mode shapes primarily affecting the runner are influenced by this variable. The study has therefore only been focused on the natural frequencies of the runner modes bending family. In Table 4, it can be observed that the maximum variation of the natural frequencies decreases for higher modes.

Table 4. Natural frequencies of the bending family runner modes for different water heads.

	17.6 mwc	7 mwc	2.3 mwc	Maximum variation (%)
B1	473	466	454	4
B2	479	473	460	4
B3	503	499	489	3
B4	588	585	581	1

5. Summary of the results

The impact on the natural frequencies of the added mass, the rotational speed and the runner blade bearing stiffness are summarized in Table 5 for comparison by presenting the maximum variation of the natural frequencies. It can be observed that the variable which causes the highest variation of the natural frequencies of the Kaplan turbine model is the added mass. And in this case, the most affected mode shapes are the ones with displacements of the runner.

Table 5. Impact of different phenomena on the change of natural frequencies of a Kaplan turbine model.

	Maximum Variation (%)
Added Mass	49.5
Rotational Speed	10.7
Runner Blade Bearings	4.0

6. Conclusion

In the present paper, the impact of the added mass, the turbine rotational speed and the runner blade bearing stiffness on the modal response of the reduced scale Kaplan turbine has been investigated.

With regard to the added mass, it has been observed that for the first mode shapes of the whole turbine rotor, which primarily show shaft displacements, the impact of the surrounding water can be considered negligible. This is consistent with the definition of added mass because the amount of water mass entrained by the first mode shapes is negligible compared to the modal structural mass. However, the natural frequencies with significant displacements at the runner are strongly reduced because of the added mass reaching maximum reductions of about 50%.

Regarding the turbine rotational speed, the maximum variation of a natural frequency between the turbine at rest and with a rotating speed of 1500 rpm is of about 11 %.

Concerning the modification of the runner blade bearing stiffness due to the change of the turbine head, it has been found a maximum variation of 4%.

In conclusion, the largest variations of the runner natural frequencies will be induced by the added mass when the turbine is submerged in water.

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