# Influence of Bulk Viscosity on Forced Vibration of Francis Turbine Runner

## David Varner[1], Miroslav Varner[2]

**Abstract**

The paper deals with natural and forced vibration of a Francis turbine runner in vacuum and in water under acceleration load. The bulk viscosity of water does not affect the natural frequencies, but significantly reduces the amplitude of the forced vibration of the runner in water. The damping effect of the bulk viscosity of water for the natural frequencies is calculated.

### 1. Introduction

In addition to normal operation reliability of the hydro turbine generator unit (HTGU), a sufficient resistance to seismic events is mandatory. The dynamic response in the event of a seismic event must be determined for the following runner scenarios: runner in vacuum, runner in water, HTGU at stillstand, HTGU in operation and short circuit of the generator [1].

The water surrounding the runner reduces the natural frequencies of the HTGU [2], [3], [4] and [5]. Vibration damping of the HTGU includes steel damping, bearing damping, friction damping in bolted joints, acoustic impedance at the water/steel interface, viscosity and bulk viscosity of water. In addition, hydrodynamic damping takes place when the water is flowing through the runner and its labyrinth, significantly reducing the amplitude of forced vibration with the flow velocity [6].

Viscosity quantifies the internal frictional resistance between adjacent layers of fluid that are in relative motion, while bulk viscosity represent the irreversible resistance when compressing and expanding the fluid with change in the density of the fluid [7] and [8]. Due to resistances, both viscosities contribute to the conversion of the mechanical energy of the fluid into the internal energy of the vibrating system [8]. The decisive damping effect of bulk viscosity on the vibration of the HTGU can be deduced from a dissipation function [7] and dependence of bulk viscosity on frequency, especially in the 0 to 30 Hz frequency range [8], which characterize excitation during seismic events [9].

In the paper, the authors focus on the numerical simulation of the harmonic response of a HTGU with a runner in vacuum and in water. The HTGU at stillstand is excited by acceleration with constant amplitude. The damping properties of water are examined as well. ANSYS program is used to model the HTGU and compute its dynamic properties.

### 2. Model

The geometric model includes a runner with 3.6 m diameter, turbine shaft, turbine head cover, draft tube, generator shaft, simplified generator rotor, radial and thrust bearings and a water domain between the guide vanes and the draft tube section 7.5 m below the lower edge of the runner band. Fixed supports are established at the nodes on the surface of the simulated solid ground of the bearings and on the outer surface of the generator rotor. At the steel/water boundary, the condition Fluid Structure Interaction (FSI) is defined. Radial and thrust bearings are replaced by springs with corresponding stiffness.

[1] RNDr. David Varner, PhD. Kotvrdovice, Czech Republic

[2] Ing. Miroslav Varner, Litostroj Engineering a.s. Technická 3029, 61600 Brno, Czech Republic

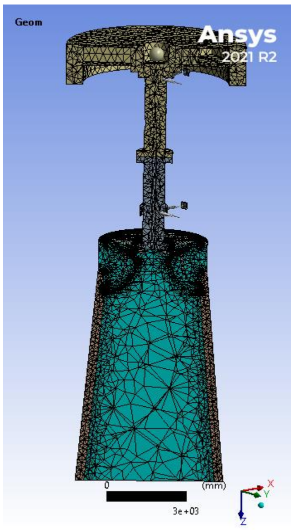


Fig. 1 Geometry and Mesh

Runner rotation, water flow through the runner and bearing damping are not modeled.

Steel properties are defined as follows: Young modulus $E = 200$ GPa, Poisson ratio $\mu = 0.3$, density $\rho = 7\,700$ kg·m$^{-3}$, Rayleigh damping with mass coefficient $\alpha = 2.38$ s$^{-1}$ and stiffness coefficient $\beta = 2.51{\cdot}10^{-4}$ s (damping $ds = 0.02$) or with mass coefficient $\alpha = 0.22$ s$^{-1}$ and stiffness coefficient $\beta = 1.00{\cdot}10^{-4}$ s (damping $ds = 0.005$). Water properties are defined as follows: density $\rho = 1\,000$ kg·m$^{-3}$, speed of sound $c = 1\,482.1$ m·s$^{-1}$, dynamic viscosity $\eta = 0.001$ Pa·s and bulk viscosity $\xi = \xi(f)$ dependent on the vibration frequency $f$ [8]. There is a similarity assumption for frequency dependency of bulk viscosity and 2$^{nd}$ viscosity on vibration frequency [6].

The finite element mesh of the HTGU with the runner in water contains 2 215 492 elements localized to 2 998 983 nodes. Natural vibration behaviors are described by 8 327 400 equations.

## 3. Natural Vibration Results

Table 1 lists natural vibration frequency for the HTGU with runner in vacuum $f_{vacuum}$ and in water $f_{water}$, coefficient of frequency reduction due to water $f_{water}/f_{vacuum}$, description of modal shapes with predominant shaft load (bend, longitudinal, torque) and number of modal diameters of runner vibration.

Table 1 Modal Behavior of the Set with Runner in Vacuum and in Water

| Mode *i* | *fvacuum*$_i$, Hz | *fwater*$_i$, Hz | *fwater*$_i$/ *fvacuum*$_i$, - | Modal Shape Description |
|---|---|---|---|---|
| 1 | 13.61 | 5.71 | 0.42 | Shaft bend |
| 2 | 13.62 | 5.72 | 0.42 | Shaft bend |
| 3 | - | 11.98 | - | Shaft longitudinal + Torque |
| 4 | 15.29 | 13.74 | 0.90 | Shaft torque |
| 5 | 43.70 | 25.95 | 0.59 | Shaft bend + Runner 1 diameter |
| 6 | 43.72 | 25.97 | 0.59 | Shaft bend + Runner 1 diameter |
| 7 | 70.29 | 26.44 | 0.38 | Shaft bend + Runner 2 diameters |
| 8 | 70.44 | 26.47 | 0.38 | Shaft bend + Runner 2 diameters |

Figure 2 shows the first, third, fourth, fifth, and seventh modal shapes of the HTGU with runner in water.

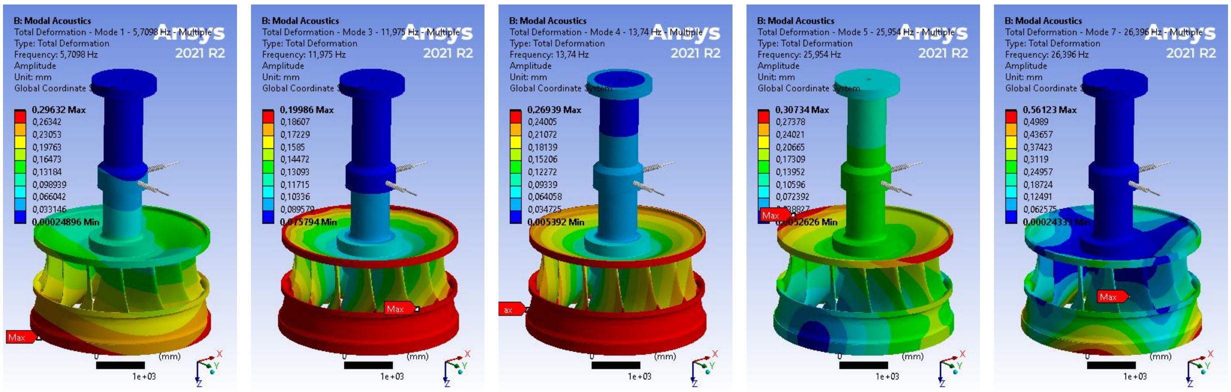


Fig. 2 1st, 3rd, 4th, 5th, 7th Modal Shapes

From an engineering point of view, the damping of the steel and the viscosity values do not affect the natural frequencies of the HTGU. The coefficients of reduction of natural frequencies $f_{water}/f_{vacuum}$ caused by water are determined by co-vibrating added masses of water. The added mass is applied in particular for vibration mode shapes characterized by high acoustic pressure e.g., in the runner labyrinth and under the turbine head cover.

Figure 3 shows computed values of total modal damping ratio *mdr* (sums of modal damping ratio of steel and modal damping ratio caused by viscosity) and modal damping ratio caused by viscosity. For all vibration modes, the vibration damping of the HTGU relative to corresponding bulk viscosity is not dependent on the damping of the steel.

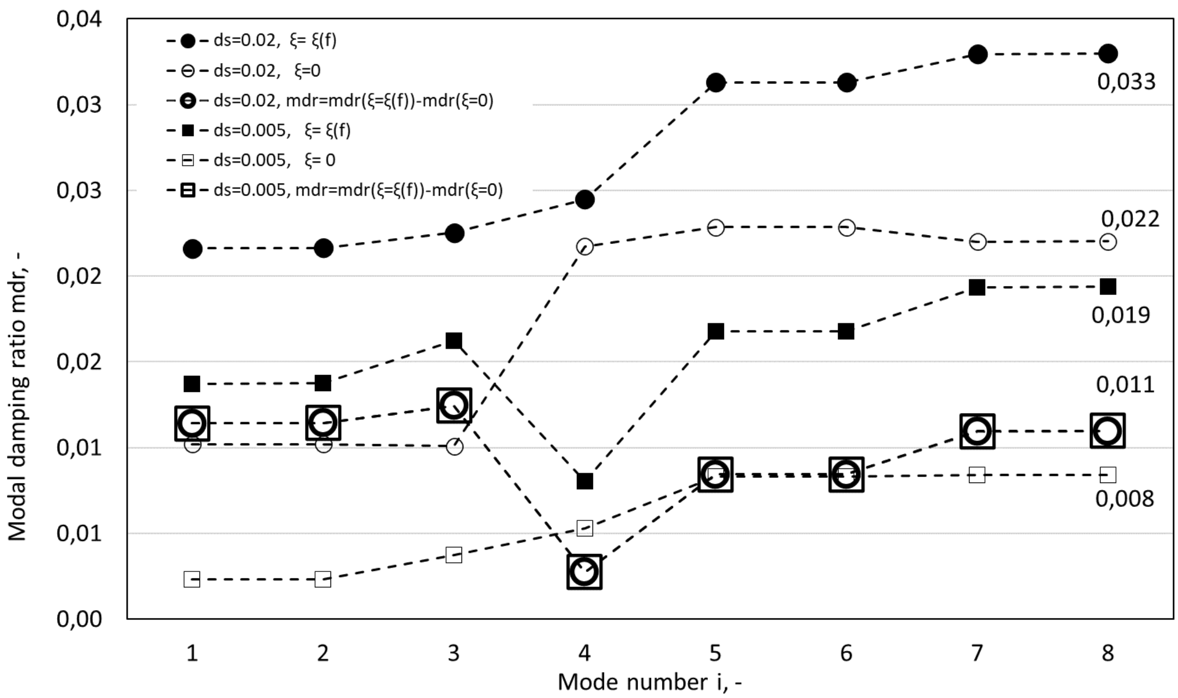


**Fig. 3 Total Modal Damping Ratio and Modal Damping Ratio due to Bulk Viscosity**

## 4. Forced Vibration with Damping

The HTGU with runner in vacuum and water is excited by acceleration with a simple harmonic course and amplitude of 1 $m \cdot s^{-2}$. Acceleration loading is applied to all modeled bodies in 0–30 Hz frequency range. Figure 4 shows maximum values of runner displacement amplitude in vacuum and in water depending on excitation frequency with steel damping *ds* = 0.02 and bulk viscosity of water $\xi = 0$ or $\xi = \xi(f)$. Figure 5 shows maximum values of runner displacement amplitude in water depending on excitation frequency with steel damping *ds* = 0.005 and bulk viscosity of water $\xi = 0$ or $\xi = \xi(f)$.

Amplitude-frequency characteristics (see Figure 4 and Figure 5) indicate vibrations with large displacement amplitude at a frequency corresponding to the first natural frequencies with no influence of the environment surrounding the runner (water/vacuum). Amplitudes of the higher-modes of vibrations are rather small and thus insignificant for a detailed analysis.

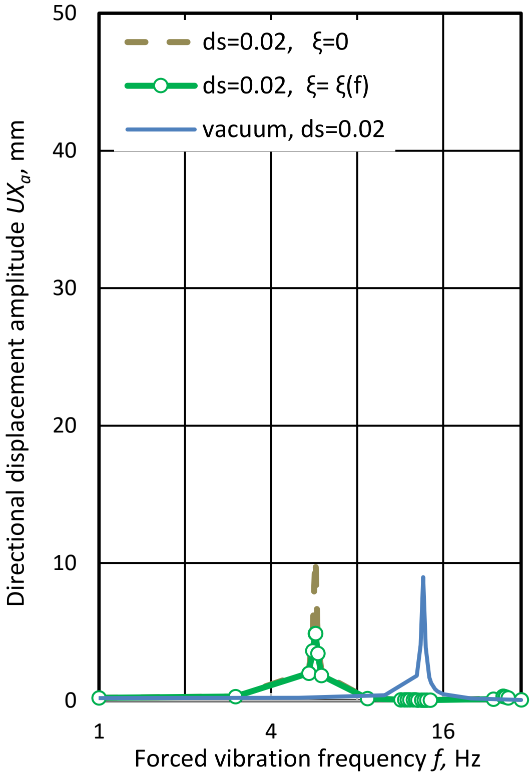


**Fig. 4 Frequency Response in Water and Vacuum, ds=0.02**

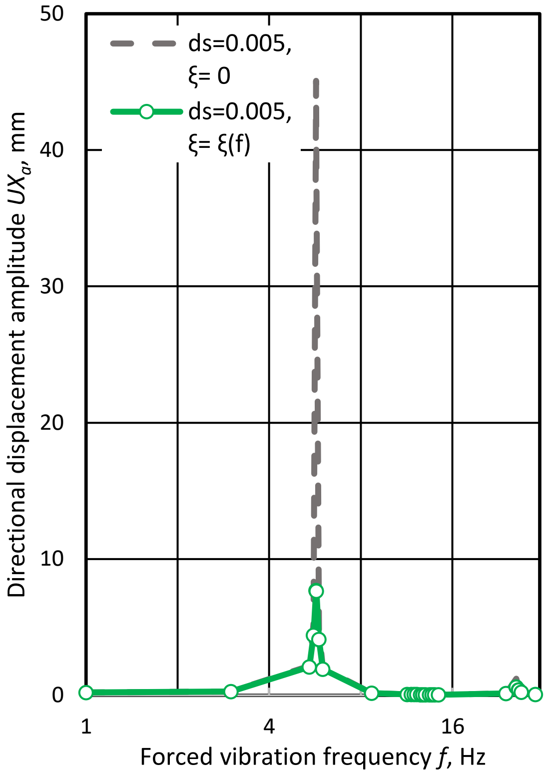


**Fig. 5 Frequency Response in Water, ds=0.005**

Table 2 lists computed results of the forced vibration in vacuum/water corresponding to the first modes of the vibration. Bulk viscosity reduces the amplitude of forced vibration to 47.3% and 17.1%

**Table 2 Modal Damping Ratios and Maximum Displacement Amplitude**

| Steel | Water | | First natural frequency $f_1$, Hz | Modal damping ratio,- | Maximum displacement, mm |
|---|---|---|---|---|---|
| Damping ratio,- | Viscosity, Pa·s | Bulk viscosity, Pa·s | | | |
| 0.02 | - | - | 13.61 | 0.0100 | 8.96 |
| 0.02 | 0 | 0 | 5.71 | 0.0102 | 10.36 |
| 0.02 | 0.001 | 0 | 5.71 | 0.0102 | 10.36 |
| 0.02 | 0.001 | $1.68 \cdot 10^6$ (average) | 5.71 | 0.0216 | 4.90 |
| 0.02 | 0.001 | $7.11 \cdot 10^5$ (lower) | 5.71 | 0.0150 | 7.03 |
| 0.005 | 0.001 | 0 | 5.71 | 0.0023 | 45.08 |
| 0.005 | 0.001 | 1.68·106 | 5.71 | 0.0137 | 7.71 |

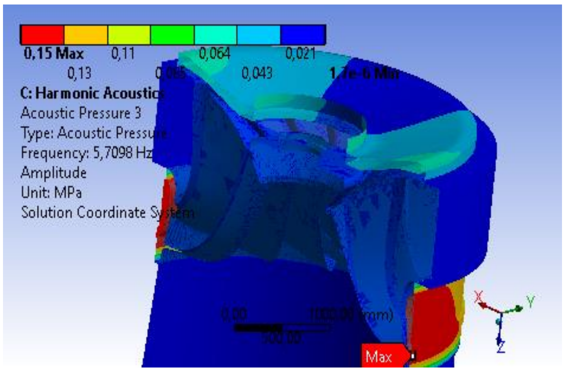


**Fig. 6 Acoustic Pressure of the 1st Mode**

for steel damping values 0.02 and 0.005, respectively. The effect of dynamic viscosity on the forced vibration of the HTGU has not been found significant. The maximum runner displacement amplitude at the forced vibration with the bulk viscosity of the water at the lower limit is 43% higher than the maximum amplitude of the forced vibration with the average value of the bulk viscosity. Figure 6 shows distribution of acoustic pressure amplitude in the water. The maximum amplitude of the acoustic pressure 0.15 MPa is in the lower labyrinth of the runner.

Displacement amplitude of damped forced vibration of the HTGU is usually calculated using displacement under static load and dynamic amplification factor [11]. The displacement under static loaded HTGU is unaffected by water. Figure 7 shows displacement amplitudes of damped forced vibration under resonance conditions calculated using formula and by the ANSYS program.

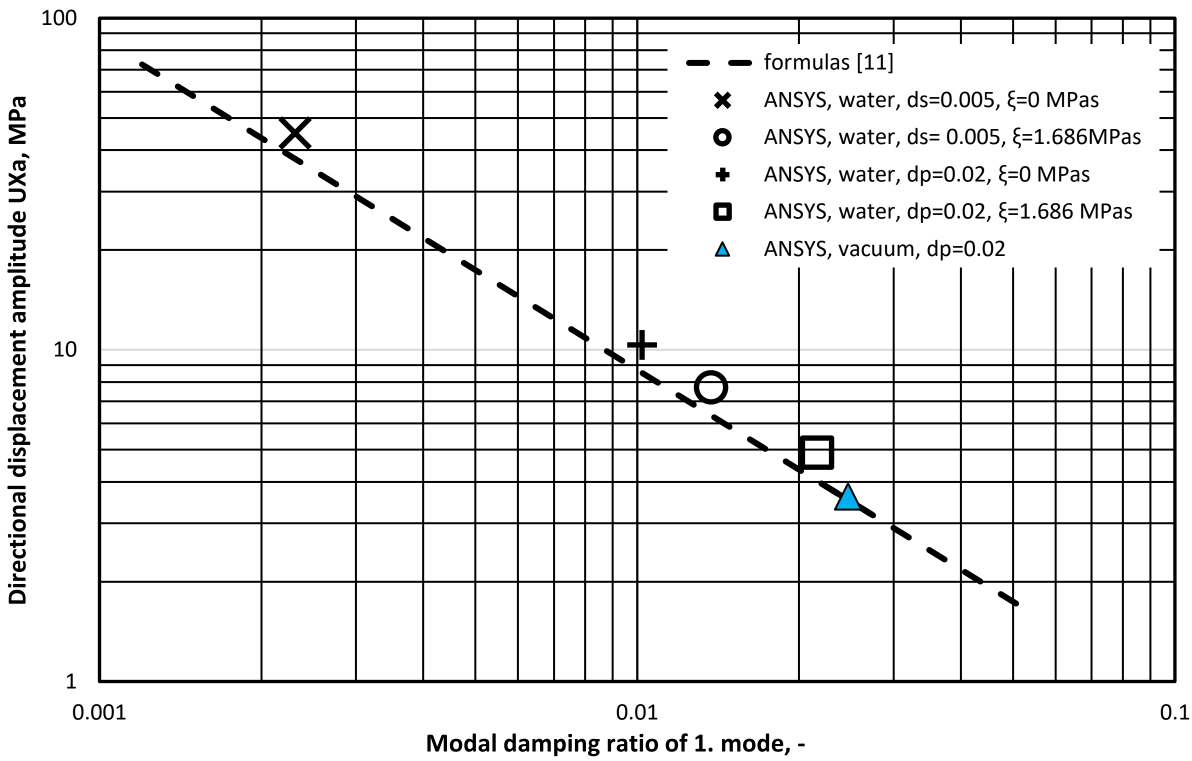


Fig. 7 Displacement Amplitude According to Formula [11] and ANSYS

ANSYS-computed displacement amplitude of damped forced vibration in vacuum corresponds to the amplitude calculated using the formula with a deviation of 2.5%. However, ANSYS-computed displacement amplitude of damped forced vibration in water is 20% higher than formula-calculated amplitude with no dependency on steel damping and bulk viscosity value.

## 5. Conclusion

Natural and forced vibrations of the runner of the Francis turbine at stillstand with acceleration loading were computed for runner in vacuum and water environments. The natural frequencies of the runner in water are lower than the natural frequencies in vacuum. The modal damping caused by bulk viscosity of water is independent of the steel damping and reaches a value of 0.01.

The displacement amplitude of damped forced vibration is significantly affected by the bulk viscosity of the water. The damping effects of viscosity are negligible in a given range of forced vibration frequencies. Due to the bulk viscosity of water, the displacement amplitude is reduced at least to 47% at seismic event.

Regardless of the steel damping and bulk viscosity of the water, the displacement amplitude of the damped forced vibration in resonant condition is 20% greater than the value calculated by formulas from the static displacement and the dynamic amplification factor.

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