Geometric parameters of influence on rotor stator interaction in radial hydraulic turbines

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Abstract. Numerical methods for the prediction of the hydraulic forcing from rotor stator interaction (RSI) in hydraulic turbines as well as the corresponding methods to calculate mechanical dynamic stresses are mature and well-established in the industry. The geometric factors influencing RSI are generally known, but little information is available on the quantification of these factors with respect to RSI forcing. We present relative quantifications of four geometric factors on RSI forcing expressed as dynamic blade torque. The factors are the blade leading edge lean, the radial distance between guide vane trailing and runner leading edges, the phase between adjacent runner channels as a function of number of guide vanes and blades, and finally the number of guide vanes independently of their effect on the phase. RSI pressure pulsations on the stator in the vane-less space between guide vanes and runner show a different tendency with the variation of the blade number compared to the dynamic blade torque. This indicates that dynamic pressure measurements in the vane-less space are not a good indicator for dynamic forcing on the runner blades.

1. Introduction

Rotor stator interaction (RSI) has been known in gas turbines for a long time, e.g. [1]. In hydraulic machinery, cracking on pump turbines due to RSI have been reported since the 1990’s [2]. A thorough derivation of the theoretical RSI pressure fields validated with experimental results is presented in that publication as well as in a more modern version by the same author [3]. First publications about cracks in Francis runners resulting from RSI can be found since the early 2000’s, e.g. [4]. Many other studies on RSI in hydraulic turbines, mostly high head Francis and pump turbines, have been published since, e.g. [5]-[9]. Most of these studies deal with the dynamic pressure patterns in the vane less space (VS) resulting from the combination of number of blades and guide vanes, the natural frequency prediction of the structure in water by finite element analysis (FEA), as well as forced response analysis of turbine components under RSI based on computational fluid dynamics (CFD) input. As these publications indicate, prediction methodologies for FEA in water as well as CFD analyses to provide unsteady pressure loads are mature in the hydraulic turbine industry. Originally, finding a dependable damping value for forced response analyses (FRA) was a challenge. Later, values from experimental correlations [6] were used and then prediction methods based on fluid structure interaction (FSI) were published, e.g. [10][12]. Acoustic effects can play a role, particularly in pump turbines [8]. Some parameters influencing the intensity of RSI are mentioned in the literature.

Dring et.al. [1] present experimental data for two axial distances of their investigation on a gas turbine stage, showing a clear pressure pulsation decrease with increasing blade row distance on both
rotor and stator. Arndt et al. [13] provide dynamic pressure measurements in the vaned diffuser of a centrifugal pump for two radial distances between vanes and rotor blades. They also measure a clear reduction in pressure pulsations with radial distance. Fan et al. [14] investigate the influence of the blade lean on RSI pressure pulsations on a pump turbine. Their comparison of a large positive with a large negative lean angle indicates that a negative lean angle results in lower RSI pressure pulsations at a low discharge operating condition. This is the result of a large low-pressure area due to blade loading near the blade leading edge, which travels past the pressure sensors with runner rotation. Agnalt et al. [15] present dynamic pressure measurements in the vane-less space of a high-head Francis turbine and a pump turbine. They identify two causes of pressure pulsations on pressure sensors: the throttling of guide vane passages and the passing of the rotating pressure field attached to the runner leading edge. In the draft of a soon to be published IEC standard on pressure pulsations, several influence factors on RSI are named without reference, namely radial distance between guide vanes and blades, lean angle of the blade leading edge and combinations of numbers of blades and guide vanes. An explanation of why these factors influence RSI is only partially given and only in a qualitative way. Tanaka’s seminal paper [3] discusses many factors influencing RSI in high-head pump turbines. From the hydraulic forcing side, he mentions the phase due to the combination of number of guide vanes and blades and the radial distance between guide vane trailing edges and blade leading edges, as well as multiple factors regarding the mechanical response of the machines.

Our study concentrates on the hydraulic excitation of RSI in Francis turbines, and provides further explanations for the effect of some geometric parameters of influence. A relative quantification of some parameters based on the dynamic blade torque as well as a comparison between locally measured dynamic pressure and dynamic blade torque is given. Most of our work is based on well-validated CFD analyses (see [5] for validation).

2. General considerations
In the literature three causes of RSI are mentioned: the potential interaction due to the pressure fields from runner and guide vanes [1][14][15], the runner interaction with viscous wakes from the guide vanes [1], and the flow blockage caused by the runner blades passing the guide vane channels [15].

There are two general locations from which RSI can be observed: from the runner in the rotating frame of reference, and from the stator in the stationary frame of reference. RSI can be quantified by means of local unsteady pressures or by means of integral values. Experimentally, typically local pressure variations (unsteady pressures) are measured [1][13][14][15], most commonly in the vane-less space or in the guide vane channels. Sometimes local pressures on the runner blades are measured [1][5]. Integral quantities that can be measured are, for example, guide vane torque or dynamic stresses. If RSI is measured on prototype runners, it is typically done by means of dynamic strains [4][6][9][11]. While local dynamic pressures are easier to measure, integral quantities are generally more useful regarding the effect of RSI on mechanical components.

In numerical simulations, the dynamic blade torque $T_{Bl_{dyn}}$, i.e. the time variation of the torque on individual blades, is a convenient and useful measure. It gives a measure of the integral dynamic loading of the blades and is therefore somewhat related to dynamic stresses. In parameter studies, integral values such as the dynamic blade torque and local values such as dynamic pressures in the vane-less space can show opposing tendencies.

3. RSI effect on the runner
The runner has been the main concern in studies on RSI, since it is the component where typically cracks occur if an RSI issue is present. In this section we discuss the basis for RSI as experienced by the runner and present four geometric parameters of influence on RSI forcing.

3.1. Guide vane outflow
The guide vane outflow is the cause of RSI in Francis runners, because it creates the non-uniformity of the flow which the runner is subjected to. Figure 1 shows typical pressure, velocity and flow angle
distributions around guide vanes. The outflow from the guide vanes resembles a typical nozzle/jet flow with the pressure and velocity gradients occurring in the flow direction. Highest gradients are found in the narrowest flow cross section A-B. This type of nozzle flow has the tendency to produce the same conditions at points A and B. At the same time the circumferentially repeating pattern of the guide vanes enforces periodicity and therefore the same flow condition at A and C. As a result, along a streamline from B to C the nozzle flow tends to produce a pressure decrease while the periodicity requires the pressure to remain constant. In the actual flow the pressure first drops and then increases again along this line, thus producing the flow non-uniformity. Curvature and thickness of the guide vanes are contributing factors. According to the Euler (angular momentum) equation for turbines it is the circumferential component \( c_{1,u} \) of the guide vane outflow that is responsible for the blade torque

\[
T_{bl} = \bar{m}_{bl} (r_1 c_{1,u} - r_2 c_{2,u}) \approx \bar{m}_{bl} r_1 c_{1,u}.
\]

Therefore, the non-uniformity of \( c_{1,u} \) results in a blade torque fluctuation as a blade passes through this flow. In addition, RSI results in a mass flow variation in each channel. We can divide the blade torque into its average and a fluctuating component where the time variation comes from both the mass flow and the angle variations:

\[
T_{bl}(t) = T_{bl,ave} + T_{bl,dyn}(t) \approx \bar{m}_{bl}(t) r_1 (c_{1,u,ave} + c_{1,u,dyn}(t))
\]

Analytical estimates for \( m_{bl}(t) \) and \( c_{1,u,dyn}(t) \) are not yet available. Instead, CFD is an excellent and efficient tool to calculate the dynamic blade torque. We generally use the relative dynamic blade torque, \( T_{bl,dyn}/T_s \) where \( T_s \) is the shaft torque, as a measure for quantifying RSI forcing on runner blades. The blade average torque \( T_{bl,ave} \) is also a useful normalization in some instances. The influence of the guide vane count and the guide vane opening angle are discussed below. The above-mentioned flow non-uniformity is a potential flow effect. It declines fast with radial distance from the guide vanes as figure 1 shows. The viscous wake also contributes to RSI but typically in a minor way in Francis turbines. It is more prominent in thermal turbomachinery.

**Figure 1.** Flow topology relevant to RSI between runner and guide vanes of a Francis turbine: pressure, flow velocity and flow angle

### 3.2. Blade leading edge lean

A straight, vertical blade leading edge is the worst case with respect to the dynamic forcing of the blade due to the non-uniform guide vane outflow. All points on such an edge are in phase, i.e. receive the forcing at the same time. If the leading edge has a lean in the circumferential direction as indicated in figure 2, there is a phase difference \( \varphi \) between any two points on that line, i.e. the points receive the maximum forcing at different points in time. As shown in the figure, the forcing along the leading edge of the blade can be considered as a harmonically varying line force \( f(t,z) \)

\[
f(t,z) = a \cos(\omega t - \varphi(z))
\]

The amplitude \( a \) can be a function of \( z \) but we consider it to be constant in a first approximation. The force acting on the leading edge of the blade is then

\[
F(t) = \int_0^{B_0} f(t,z) \, dz
\]

In order to calculate the integral over the height of the leading edge \( B_0 \), we need to know the function \( \varphi(z) \). Let’s consider a straight-line leading edge at an arbitrary lean. Then
\[ \varphi(z) = \varphi_0 + \Delta \varphi / B_0 \cdot z \]  

(5)

And with the number of guide vanes \( n_{GV} \) the forcing factor is:

\[ f_{\varphi} = 2 / \Delta \varphi \sin \Delta \varphi / 2 = 2 \sin \left( \frac{n_{GV} \Delta \vartheta}{2} \right) / (n_{GV} \Delta \vartheta) \]

(6)

For clarification, the angle \( \varphi \) describes the RSI phase relationship due to the lean. It covers \( 2\pi \) for the circumferential distance between two guide vanes. The phase difference between the blade leading edge points on crown and band is \( \Delta \varphi \). The angular coordinate \( \vartheta \) in the cylindrical coordinate system can be used to express the phase difference by \( \Delta \varphi = n_{GV} \Delta \vartheta \).

Figure 3(a) shows a comparison between the relative dynamic blade torque calculated with CFD for multiple lean angles, and the curve obtained by scaling a single reference CFD point with equation (6). The agreement is very good for most of the curve. Only at large values of positive lean there are effects that are not fully captured by the theory. The equation has a singularity at \( \Delta \varphi = 0 \). In practice this can be avoided by using very small values unequal to zero. For cases where the inlet leading edge of the blade is not a linear curve, the integral from equation (4) has to be solved for that particular curve. Figure 3(b) illustrates the theoretical effect of large leading-edge inclinations on the amplification factor. After a minimum with zero amplification is reached, the amplification increases again as \( \Delta \varphi \) increases. This shows that, if other design constraints permit, the leading-edge lean can be an effective way of reducing RSI forcing on the runner blades. In very high head turbines and pump turbines \( B_0 \) is often too small for a significant reduction of RSI forcing by means of inlet leading edge lean.

![Figure 2. Nomenclature for geometric RSI forcing factors](image)

![Figure 3. Effect of lean angle on dynamic blade torque: (a) comparison CFD and values from reference CFD point scaled to other lean phase angles (b) Theoretical lean factor over large (hypothetical) range of lean phase angles](image)

3.3. Radial distance runner-guide vane

As can be seen from the flow topology at the guide vane outlet in figure 1, pressure and flow angle non-uniformity decrease rapidly towards smaller radii. This results in an exponential decrease of the
dynamic blade torque with radial gap $\Delta r$ between guide vane trailing edge and runner leading edge. Rather than using the simple ratio of the guide vane trailing edge radius $r_{GV,TE}$ and the average blade leading edge radius $r_{Rn,LE}$, it is useful to subtract one to define a normalized gap ratio:

$$r_{\text{norm}} = \frac{r_{GV,TE}}{r_{Rn,LE}} - 1 \quad (7)$$

In this form we need a single constant for a radial distance influence factor because it is reasonable to assume that such an influence factor becomes 1 at $r_{\text{norm}} = 0$, i.e. no reduction in forcing at zero gap. Then the exponential radial gap influence factor becomes

$$f_{dr} = \exp(-20 \cdot r_{\text{norm}}) \quad (8)$$

In that equation, the coefficient is obtained from a curve fit of CFD data. Figure 4 shows a comparison of the CFD dynamic blade torque for two Francis turbines where the radial gap is varied. The graph in figure 4 illustrates that equation (8) is a good fit for the data. This is purely empirical. It may be possible to derive this exponential law analytically.

3.4. Blade number and phase

Figure 5(a) shows the phase distribution on a constant span surface of a Francis turbine. The phase difference between adjacent channels as well as the near-constant values within a runner channel are clearly visible. The graph in figure 5(b) gives a more detailed picture of the phase distribution with respect to blade loading. The phase value is plotted on lines where blades intersect a span-constant surface for three adjacent blades (see figure 6(b)). From the middle of the channel to the trailing edge, the actual phase corresponds nearly perfectly with the theoretical phase (equation (9)): the value on the pressure side of one blade is the same as that on the suction side of the next blade. From the leading edge to about the middle of the channel, the phase shows a transition where it deviates from the theoretical phase.

Figure 4. Dynamic blade torque forcing factor $f_{dr}$ due to radial distance between guide vane trailing and runner leading edge as function of normalized radial gap.

Figure 5. Phase angle of RSI pressure pulsation (a) on a span-constant surface (b) on lines from intersection of span-constant surface and blade surface for 3 adjacent blades.
In order to shed some light on the effect of the phase angle on the dynamic forcing quantified by the dynamic blade torque, we varied the number of blades for a constant number of 16 guide vanes on a hypothetical design. The results are shown in Figure 6(a). Minima are present at \(n_{GV}/n_{BL} = m\) with \(m = 1,2,...\). The dynamic blade torque increases to local maxima \(n_{GV}/n_{BL} = (2m + 1)/2\) with \(m = 0,1,2,...\) as stated by Tanaka [3]. He introduces the following forcing factor due to the phase difference between two adjacent runner channels \(\phi\):

\[
\phi_1 = 2|\sin \Delta \phi / 2|
\]  

Our definition of phase difference

\[
\Delta \phi = 2\pi (n_{GV}/n_{BL} - 1)
\]

is \(2\pi\) phase shifted compared to Tanaka’s, which does not affect the forcing factor. The data in Figure 6(a) show that the phase effect is largest for \(0.5 < n_{GV}/n_{BL} < 1.5\), i.e. when blade and guide vane numbers are close. A relative factor of almost 6 is possible. For small blade numbers compared to guide vane numbers the effect becomes small. This is different from Tanaka’s [3] findings on pump turbines: he measured significant variations in dynamic stresses even for small blade numbers.

Equation (9) predicts zero blade forcing for zero phase, i.e. when the numbers of guide vanes and blades are the same or integer multiples of each other. As we see in Figure 6(a), not surprisingly, the forcing does not drop to zero at \(n_{GV}/n_{BL} = 1\). One reason for this is that the assumption of equal forcing amplitudes on pressure and suction sides of a blade, which was made for equation (9), is not correct. Figure 6(b) shows a clear pressure amplitude difference along much of the blade length. The dynamic forcing on the blade then follows from

\[
F_{BI}(t) = F_{ps}(t) - F_{ss}(t) = a_{ps} \cos \omega t - a_{ss} \cos(\omega t + \Delta \phi)
\]

With \(x = a_{ss}/a_{ps}\)

\[
F_{BI}(t) = a_{ss} \sqrt{x^2 - 2x \cos \Delta \phi + 1} \cos(\omega t + \Delta \phi / 2)
\]

Our forcing factor due to the phase becomes

\[
\phi_2 = \sqrt{x^2 - 2x \cos \Delta \phi + 1}
\]

The average amplitude ratio from our CFD results is \(x \approx 0.65\). The forcing factor from equation (13) becomes the same as from equation (9) for \(x = 1\).

Dividing the normalized relative dynamic blade torque from Figure 6(a) by the phase forcing factor \(\phi_2\), we obtain the graphs in Figure 7. Expecting a constant value, the phase factor greatly reduces the bias on most data points, indicating that the main effect of guide vane to blade ratio is due to the phase effect on the forcing. However, there are two singular points at 360° and 1080° which deviate from the values of the other points by a large factor. These are points where the theoretical forcing function
has a minimum. In other words, for points of minimal forcing factor \( f_{\phi 2} \) other than at \( n_{GV}/n_{Bl} = 1 \), the forcing factor significantly overestimates the reduction of dynamic blade torque.

The zoomed phase range of figure 7(b) represents guide vane and blade number combinations that are most relevant for the practical design of Francis turbines. The variability is largest around zero phase. Away from zero phase the variability is quite small. Two plateaus of different levels can be discerned below and above zero (\(~1.1~\) and \(~0.75\)). Therefore, there is still a general bias with respect to blade numbers lower or higher than guide vane numbers. From these data, blade numbers higher than guide vane numbers tend to generate higher dynamic blade torque, once the phase effect is removed.

![Graph](image)

**Figure 7.** Normalized relative dynamic blade torque divided by phase forcing factor \( f_{\phi 2} \) as a function of phase angle, and normalized phase forcing factor \( f_{\phi 2} \) (a) full calculated phase range (b) phase range zoomed to most relevant range for Francis turbines (0.5 < \( n_{GV}/n_{Bl} < 1.6 \)).

### 3.5. Guide vane number

It is intuitive, that the number of guide vanes independent of all other factors influence RSI forcing. If we assume the limit case of an infinite number of infinitely thin guide vanes, there would be no flow non-uniformity at the guide vane outlet and therefore no RSI forcing. In order to extract the influence of the guide vane number on RSI forcing we performed a parameter study where we kept the same runner geometry and only varied the number of guide vanes. As we have seen in section 3.4 varying the number of guide vanes will affect the phase. In addition, when changing the number of guide vanes, the individual guide vanes must be scaled in order to maintain the relative position of the contact points. By scaling the guide vanes for a constant guide vane circle diameter, we change the radial distance between guide vane trailing edge and runner leading edge. Therefore we divide the dynamic blade torque from our parameter study with the radial gap forcing factor from equation (8) and the phase forcing factor from equation (13).

\[
T_{Bl,dyn,corr} = T_{Bl,dyn,CFD}/(f_{dr}f_{\phi 2})
\]  

We assume that the remaining variation of dynamic blade torque is a function of guide vane number only. To represent the number of guide vanes, we relate them to the arbitrarily chosen guide vane number of 24 and subtract one to obtain the relative normalized guide vane number

\[
n_{n,GV} = (n_{GV}/n_{GV,ref}) - 1 = (n_{GV}/24) - 1
\]

In figure 8 the corrected dynamic blade torque divided by the value at 24 guide vanes \( T_{Bl,dyn,corr}/T_{Bl,dyn,ref} \) is plotted as function of the normalized guide vane number \( n_{n,GV} \). The red triangles represent the CFD results and show an exponential tendency. The dashed black line is the slightly simplified curve fit of the CFD data with the following exponential function

\[
f_{nGV} = \exp\left(-4.5n_{n,GV}\right)
\]

The graph shows that this gives a good fit. Its \( R^2 \) value is about 0.98.
3.6. Others

Other geometric influence factors surely exist. One such factor is the guide vane opening. As indicated in section 3.1 it will influence the flow non-uniformity at the outlet of the guide vanes. By tendency it should increase with smaller guide vane openings. At the same time, decreasing the guide vane opening results in a smaller radial gap, which, as we have seen in section 3.3 has a strong influence on the RSI forcing. Both effects seem to mostly cancel each other out. In our experience, within the typical operating range of Francis turbines, there is little influence of guide vane opening on dynamic blade torque. Dynamic stresses tend to slightly increase from full load to 50% load due to decreasing hydrodynamic damping [9].

Guide vane overlap is thought to have an effect as well. The nozzle-like flow acceleration and pressure reduction described in section 3.1 will be more gradual if there is more guide vane overlap. As discussed in section 2, we know from the literature [14][15] that guide vane channel blockage and blade leading edge pressure fields influence RSI. Therefore, blade leading edge thickness and blade inflow angles must influence RSI. However, we have no knowledge of any quantification of the effect these parameters have on RSI other than by CFD or measurements on a case by case basis.

3.7. Combined effect of geometric influence factors

The four RSI forcing coefficients presented in this paper can be combined to extract the bias that each geometric parameter of influence introduces into the dynamic blade torque. To normalize it, we divide it by the total shaft torque $T_s$. Our corrected relative dynamic blade torque then becomes

$$T_{\text{Bl.dyn.cor}}, = \frac{T_{\text{Bl.dyn,CFD}}}{T_s \cdot (f_p f_{dr} f_{\phi} f_{\text{nGV}})}$$

This dimensionless dynamic blade torque can be plotted against a suitable machine type related parameter. Based on such a correlation, an estimate of the dynamic blade torque can be obtained. The correlation based on Andritz’ data is confidential but indicates that the above-mentioned influence factors eliminate a considerable part of bias in the dynamic blade torque. We should mention, that for an accurate assessment of RSI, unsteady CFD analysis is indispensable. With today’s computer resources such calculations can be performed within a few hours.

4. RSI effect on the distributor

As mentioned in section 2, in many investigations, local dynamic pressure measurements in the vane-less space (VS) are used to assess RSI intensity. To investigate the pressure pulsations in the vane-less space, we express them as local dynamic head over head $dH/H$. We use the maximum value on a circle around the vane-less space on the head cover. The graph in figure 9(a) shows the evolution of the VS pressure amplitude as a function of the blade number. The blue line (TS) represents half the

![Figure 8. Scaled relative dynamic blade torque $T_{\text{Bl.dyn.cor}}/T_{\text{Bl.dyn.ref}}$ as function of shifted relative number of guide vanes $n_{n,GV}$. CFD results and empirical curve based on these CFD results.](image)
pressure variation based on the time signal, and the yellow line (FT) the pressure amplitude of the fundamental RSI frequency. For large blade numbers \( n_{GV}/n_{BL} < 1.4 \) the two lines overlap. At smaller blade numbers the two curves start to deviate strongly. This happens when in Tanaka’s [3] conditional equation for the existence of RSI

\[
ND = m \cdot n_{BL} - n \cdot n_{GV}
\]

the integer multiplier \( m \) becomes \( >1 \) for the smallest nodal diameter ND. Then it becomes important to include the harmonics of the fundamental RSI frequency.

On the blue curve in figure 9(a), we can identify a singularity at \( n_{GV}/n_{BL} = 1 \) and a hint of one at \( n_{GV}/n_{BL} = 2 \). In general, the pressure pulsation in the vane-less space increases with decreasing blade number. This can be explained with the larger pressure loading per blade as the blade number decreases. The resulting larger pressure variation around the blade represents a non-uniformity with an upstream effect. As the blade rotates, an observer in the stationary VS sees the periodic passing of a spacial pressure non-uniformity (see also [14][15]), which at approximately equal power output is greater for a runner with fewer blades. In a range of the guide vane to blade ratio \( n_{GV}/n_{BL} \) just above 1 the general tendency of an increase in VS pressure amplitudes with deceasing blade number is interrupted with a local minimum.

Comparing the VS pressure pulsation with the dynamic blade torque in figure 9(a) (blue vs red curves), shows that they have opposing tendencies over much of the \( n_{GV}/n_{BL} \) range. This is important to note, because it means that the VS pressure pulsation is not a good indicator for the dynamic forcing of the runner blades.

![Graph](image)

**Figure 9.** Comparison of maximum pressure pulsation on a fixed circle around the vane-less space on the head cover with dynamic blade torque as function of (a) blade number (b) radial gap between guide vanes and blades.

Figure 9(b) shows a comparison of the VS pressure pulsations with the dynamic blade torque as a function of the normalized radial gap. Clearly, both are well correlated. Both decrease with an increasing radial gap between guide vane trailing edges and blade leading edges.

5. **Summary and conclusion**

Our study presents details on four geometric factors influencing the dynamic forcing on turbine runner blades from RSI. The forcing is quantified by means of the dynamic blade torque. The four factors are:
- Blade leading edge lean
- Radial distance between guide vane trailing edge and runner leading edge
- Phase between two adjacent runner channels as a result of the combination of number of guide vanes and blades
- Number of guide vanes independent of their influence on the phase
From the runner perspective, the number of blades is found to only have a small effect on the dynamic blade forcing once it has been made independent of the phase. The effect of RSI on the distributor is different from the effect on the runner. The maximum pressure amplitude in the vane-less space – as often measured in hydraulic laboratories to assess RSI intensity – has an opposing tendency with blade number compared to the dynamic blade torque over much of the \( \frac{n_{GV}}{n_{BL}} \) range. This is important, since it indicates that the local vane-less space pressure amplitude is not a good indicator of RSI intensity on the runner blades.

The correlations for the RSI influence parameters give a good estimate of how the blade forcing evolves with these parameters. However, secondary factors are not captured by these correlations, and blade forcing is not the only important aspect of RSI to consider. Ultimately, full CFD and FRA analyses are required and are standard in the assessment of modern hydraulic turbine designs.

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