Impact of cracks on vibration parameters and lifetime of hydraulic units

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Abstract. One of the key factors determining the hydraulic unit reliability and lifetime is the current technical state of the runner and, first and foremost, its lack of dangerous cracks. Dynamic load throughout the mode range affects the crack occurrence and development. Blade system crack formation process depends heavily on actual equipment operating modes. Yet, modern hydraulic unit diagnostics systems tend not to record crack formation on its early stages. As a result, routine maintenance often reveals long cracks which require significant financial and time resources to repair. Vibration-based diagnostics systems reveal cracks during operation ineffectively: calculations of the Francis (hydraulic) turbine’s runner prove that. High structural rigidity and specific eigenforms don’t permit correlating the blade dynamics to measured vibration parameters. Analytic approach based on calculating fatigue strength and fracture mechanics methods could solve the problem mentioned above.

1. Introduction

The important advantage of hydraulic units (HU) is their high maneuverability permitting to start in mere dozens of seconds and reach peak capacity or rapidly change the load. This allows using HU as a main tool in securing sustainability of energy systems as well as compensating power generation instabilities through renewable energy sources (wind, sun, tides, waves, etc.), however, also leading to frequent mode changes, prolonged medium- and low-capacity operation, high number of starts/stops and premature equipment service-life exhaustion [1-2].

This article focuses on high-pressure high-capacity hydraulic units usually equipped with Francis hydraulic turbines boasting high reliability and resistance to cavitation. In Russia Francis turbines of the highest capacity are installed at these hydroelectric power stations: Sayano-Shushenskaya HPP, Krasnoyarsk HPP, Bratsk HPP, Ust-Ilimsk HPP, Bureya HPP and Boguchany HPP.

Runner (RR) is one of the main lifetime-defining parts of the hydraulic unit. Francis turbine runner consists of 11-19 blades, firmly fixed between the lower massive band and the crown. Such blade grid ensures significant structural rigidity of the RR, however, due to geometrical features causes a high level of stress concentration in the connection area between the blade’s output edge and the band or crown [3-4].

With the hydraulic unit nearing its peak working efficiency, the maximum blade-system static stress for Francis turbine runner is usually no more than 20-30% of the blade material yield strength, while the dynamic stress not exceeding 10% of the static stress. Such operating mode can’t cause crack formation even taking heightened stress concentration into account. However, the blade flow...
pattern is far from optimal on low/medium partial power modes or when changing modes. This leads to significantly increasing dynamic stress in turbine components and accelerates crack formation [5-7].

As a result, after the HU long-term operation especially including frequent mode-factor changes cracks are often found spreading along the blade-to-crown or band-to-edge welded lines. One usually finds such cracks while checking the unit during routine maintenance [6, 8-9]. At this moment the length of these cracks usually exceeds 300 mm, their opening may be wide and even straight-through. In some cases the uncontrollable crack development leads to complete blade rupture. Blade refurbishment concerning deep and long cracks takes much time and money as well as leads to adverse effects such as high level of welded areas residual stress, redistributed blade loads due to flow part geometry distortion etc.

The problem is no alarming signs of developing defects found in time. Even when using monitoring/diagnostic equipment installed on the HU to perform regular vibration tests, the long-term (years or decades long) crack formation remains undetected. Importantly, modern hydraulic unit monitoring/diagnostic systems analyze data from multiple sensors (sometimes over 200 pcs) installed on static and rotating parts of the unit which is in fact effective for identifying series of operational faults [10]. Unfortunately, cracks developing in the runner are not in the list of the detectable faults. Main reasons for that are explained in this article from mechanical engineer’s point of view.

2. Computational models and calculation task

To prove vibration testing systems ineffective for locating emerging cracks in the hydraulic unit runner blade system a computational model of a hydraulic unit rotating part was constructed. It consists of a runner and a hydro-generator, all of them firmly connected to a shaft. To make obtained results more comprehensible the model has the following simplifications:

- RR model uses surface elements while retaining curvature and thickness in each point of the blade without taking actual blades’ welded connection area fillet radii into account;
- through cracks in the RR are modelled as ruptures along the connection line between the blade and the crown. These cracks start at the side of the output edge; the length of the rupture increases gradually with the crack’s growth;
- the generator model has a simplified view. It allows to retain main mass, dimensions and rigidity parameters along with excluding local effects determined by generator’s rotor design features;
- construction fits are assumed to provide tight fit for all the connected elements throughout the mode range;
- presumably, tightening the stud bolts ensures flange joints staying tightly closed under working loads;
- the unit’s shaft attachment to guide bearings (upper and lower ones of the generator, the turbine one) and its support by the trust bearing is modelled by anchoring the computational model in relevant cross-sections and directions while maintaining equivalent rigidity rates imitating rigidity of the support.

The first-step calculation task was determining defect-free construction’s eigenfrequencies and search for correlations between runner blades’ deformation and shaft displacement in areas of actual vibration sensor positioning on a full-scale HU.

The second-step calculations include studying blade crack growth effects on the computational model’s eigenfrequencies. Thus, these two options were considered: the growth of a single crack and the simultaneous crack growth on all blades.

In addition to the computational model described above the second step included using refined RR model which shows all the geometry features of the area where blades connect to the crown and the band. The refined model takes RR’s cyclical symmetry into account, but doesn’t take the unit’s shaft and the hydro-generator. The computational model is anchored according to the cross-section relevant to the turbine bearing.
3. Calculation results

Eigenforms and eigenfrequencies on the computational model were defined for 0-160 Hz range. This fully overlaps possible external effect frequency readings concerning the hydraulic unit working in various operating modes. Obtained results are in groups by eigenform type depending on their affecting the possibility of identifying blade system cracks using measurements of unit shaft’s vibratory displacements.

Figure 1 shows main reviewed computational model eigenform types. They present different variants of correlation between runner deformations and radial shaft displacements close to supports – places having vibration sensors installed on a full-scale hydraulic unit:

a) bending vibrations of the whole unit itself when there is a correlation between runner displacements (deformations) and radial displacement of the shaft detected by shaft-runout sensors close to bearings;

b) torsional vibrations without shaft displacements in a radial direction during the runner twist; registered vibrational parameters don’t usually include the shaft twist; accordingly, the vibration monitoring system doesn’t identify such vibrations;

c) bending vibrations of the runner without HU shaft displacement close to support components; vibratory shaft displacement is absent despite the present runner deformation; the vibration monitoring system doesn’t form a warning signal even if resonant vibrations of such type are present.

Figure 1. Main eigenform types for a defect-free construction: (a) – bending vibrations of the unit’s shaft, (b) – torsional vibrations of the runner without radial shaft displacement, (c) – bending vibrations of the runner without HU shaft deformation.

Figure 2. The effect of a single or multiple cracks on unit’s eigenfrequency.
Figure 2 shows comparative unit’s eigenfrequencies range calculation results for three cases in general: no cracks on RR blades, a crack on one RR blade, and cracks of equal length on all RR blades. Crack length is considered to be 10% of the length of the joint between the blade and the crown. The x-axis represents the eigenform number: first 12 values given fall within the range of 0-50 Hz. The y-axis sets eigenfrequency values of the computational model. Vibration-based diagnostics system is able to identify such values under operating conditions of the unit at a hydroelectric power station (for example, based on analyzing recorded vibration displacement of the turbine bearing).

Figure 3 shows selected results for the refined model: the first eigenfrequency of the blade without cracks and the blade with a through crack of different lengths $L$: $L=0.14L_0$ and $L=0.25L_0$, where $L_0$ is the total length of the line where the blade is welded to the RR crown.

Blade eigenfrequency shift ($\Delta f$, %) is calculated using relative numbers: the remainder of the non-cracked blade eigenfrequency subtracted by the eigenfrequency of the blade with a crack of $L$ length, divided by the defect-free construction’s eigenfrequency. The stress-strain state correlating to the runner eigenform was defined using the first eigenfrequency. The effect of eigenfrequency shift towards the change in the vibration parameters available for measuring on a full-scale hydraulic unit non-stop operation was investigated. There have also been similar numeric tests for higher eigenfrequencies.

![Figure 3](image)

**Figure 3.** The first form of blade eigenfrequencies with different crack length: (a) $L=0$ mm, $\Delta f=0$, (b) $L=0.14L_0$, $\Delta f=4.4\%$, (c) $L=0.25L_0$, $\Delta f=17.8\%$.

Figure 4 shows results of relative crack length $L$ (percentage of total joint between the blade and the crown) effect on eigenfrequency value ($f$). Eigenfrequencies were calculated using the refined model. The report presents data for the initial three eigenvalues of a single blade with a growing crack. Modelled through-cracking began from the output edge and was spreading along the attachment line between the blade and the crown.

![Figure 4](image)

**Figure 4.** Relation between the eigenfrequency and the blade crack length.
4. Discussion of results

The results presented in figure 1 prove identifying torsional vibrations and runner bending without shaft bending using vibration sensors installed close to the supports is impossible. The same applies to axial eigenform of the runner and any hybrid forms which possess components presented in figure 1(b) and 1(c). Therefore, in these cases blade dynamic stress level cannot be defined which leads to inability of estimating the remaining lifetime of a HU using vibration monitoring data on an operating HU.

Blade crack development leads to decreasing construction eigenfrequencies and support element reaction change. However, as conducted calculations show (see figures 2-4), high construction rigidity of a Francis turbine runner does not allow identifying these changes at an early stage. For example, changing of the first eigenfrequency of a separate blade for 5% corresponds to a through crack with a length close to 15% of total joint between the blade and the crown (see figure 4). At the same time, unit shaft eigenforms and eigenfrequencies change by less than 0.5% (see figure 2). This is practically impossible to identify under actual operating conditions. Although the development of a long, even though, crack leads to changing the blade’s eigenform (maximum deformation gradually shifts from the middle of the output edge closer to the crown in step with crack growth), this almost doesn’t affect crown or runner band form changes, not to mention the shaft or hydraulic unit supports. That’s why such changes are difficult to register using vibration monitoring data analysis.

Calculated testing has shown that separate element identifiable changes throughout the shaft eigenform monitoring system range will occur only when cracks in all the runner blades longer than 10% of total joint length are present (see figure 2). Obviously, such situation is unlikely to occur as blade cracks form unevenly and grow with different speeds. This is primarily related to the lack of RR geometry perfection and to the presence of disk eigenforms (see figure 1(c)) and the fact that significant dynamic stress distribution unevenness throughout the blades accompanies them.

5. Conclusion

Calculation results in the article show the following: high construction rigidity and specific eigenform presence do not allow to correlate blade dynamic stress levels (also responsible for emergence and developing of cracks) to measured hydraulic unit vibration monitoring/diagnostics systems. This makes detecting cracks in time on an operating hydraulic unit difficult. Solving the problem is possible through approaches which implement analytical method based on calculating fatigue strength and crack growth speed using fracture mechanics approaches [11].

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