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Chapter

Periodic Instabilities in a Specific Low-Speed Pump Working as a Turbine

Hernan Bolaños and Francisco Botero

Abstract

The hydrodynamic instabilities in a turbomachine can be divided into two main groups: periodic (or quasi-periodic) and nonperiodic. And the total instability, calculated from a statistical parameter with linear characteristics, such as variance, can be defined as the sum of periodic and nonperiodic instabilities. Based on the above, the main objective of the study was to estimate the periodic instabilities in a pump operating as a turbine. For this purpose, pressure fluctuation signals from sensors installed on the turbomachine volute and spaced 135° apart were used. The signals were analyzed in the time and frequency domain to identify, initially, the periodic instabilities and their relationship with the spectral components and, subsequently, to estimate the magnitude of these instabilities as the variance of the filtered series in the spectral band related to the periodic instability. In addition, the study aims to establish the contribution of periodic instabilities to total instability.

Keywords: pump as turbine, hydrodynamic instabilities, periodic instabilities, variance, phase analysis, Fourier analysis

1. Introduction

A reversible pump can either supply energy to the fluid or it can obtain energy from the fluid, depending on the direction of rotation of the impeller and the direction of flow. When the pump draws energy from the fluid, it is known as pump as turbine (PAT). The interest in this type of turbomachine is because power generation is less expensive in equipment than a conventional turbine for low power ranges [1–8]. Even though the use of PATs may be the best option for harnessing small hydro resources [9], especially in rural and remote areas with power supply problems [7, 8], the study of this type of turbomachines has not been extensive. This could be verified by a quick search on the Web of Science database, revised on November 1, 2022. There, 259,547 references are reported on pumps, 132,230 references on turbines, and only 281 references on PAT’s. Of these 281 references, a general classification was made according to some keywords of interest in the field of turbomachines, which can be seen in Table 1.
Table 1 shows that the most studied issue related to PAT performance is efficiency, which is obvious given the importance of this aspect in power generation, and the least studied are cavitation and flow instabilities. Cavitation is an important source of instabilities, so it was included in the same query as instabilities. In this group, the following topics are addressed: geometry, efficiency, cavitation [4, 7–21], flow structures [8, 22–25], pressure fluctuations [16, 21, 26, 27], vortex rope [27], energy losses when switching from pump to turbine mode [28], and application of entropy production theory for energy losses [29]. None of them addresses the measurement of instabilities.

The study of instabilities is important because they cause several problems in the performance of turbomachines, such as efficiency losses, noise, and vibrations [12, 14, 16, 30–33] and can even threaten their structural integrity [16, 33]. Given this scenario, it is deemed important to identify the hydrodynamic phenomena that can affect the performance of a PAT and to estimate its level of instability. Brennen’s classification [33] is used to establish a framework for flow instabilities in turbomachines. According to this author, hydrodynamic instabilities causing vibrations can be classified into three classes: global flow oscillations, local flow oscillations, and radial and rotodynamic forces. These three classes bring together at least 12 flow instabilities. The total instability at an operating point of a turbomachine may be the combination of several types of instabilities.

Instability analysis can be done using theoretical models; however, when the flows are very complex, it is necessary to use techniques that include experimental information [34]. One of those techniques is the frequency domain analysis of signals from sensors installed in the turbomachine. This is also known as Fourier analysis. An advantage of Fourier analysis is that the signals of the variables of interest obtained in the time domain can be converted into individual frequency components and vice versa. In addition, variables of interest (such as velocity, acceleration, and pressure, ), can be expressed as a sum of their mean and a complex component incorporating the amplitude and phase of the fluctuation [34]. In this paper, the words fluctuation, perturbation, and instability are synonymous.

Although the total instability at an operating point can be regarded as a sum of instabilities of different origins under the linearity assumption [34], not all these sources of instability have periodic characteristics. This means that, using Fourier analysis, only periodic or quasi-periodic instabilities can be clearly identified. Therefore, the main objective of this research is to estimate the instability due to periodic phenomena in a low-specific-speed pump working as a turbine. Concerning periodic instabilities, it is relevant to point out that they can be divided into two groups. The first group corresponds to those that depend on the rotation frequency of the

<table>
<thead>
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<th>Query</th>
<th>Number of references</th>
</tr>
</thead>
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<tr>
<td>“Pump as turbine”</td>
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<tr>
<td>“Pump as turbine” AND performance</td>
<td>203</td>
</tr>
<tr>
<td>“Pump as turbine” AND performance AND efficiency</td>
<td>151</td>
</tr>
<tr>
<td>“Pump as turbine” AND performance AND design</td>
<td>114</td>
</tr>
<tr>
<td>“Pump as turbine” AND performance AND (cavitation OR “flow instabilities”)</td>
<td>24</td>
</tr>
</tbody>
</table>

Table 1. References on the web of science database, revised November 1, 2022.
turbomachine, and the second to those that do not. In this research, only those of the first group were considered.

2. Materials and methods

2.1 Test rig

The centrifugal pump under study, hereafter referred to as PAT, is an ITT Goulds of 1491.4 W (2 HP), with the specifications shown in Table 2. The PAT was instrumented with pressure transducers in the high- and low-pressure orifices; two pressure fluctuation sensors on the volute; a torque sensor and an encoder on the shaft; a flow meter in the high-pressure pipe; and a frequency drive. The PAT was added to the test rig, which consisted of a closed pipe loop, a supply tank, and a recirculation pump, which was used to simulate the hydraulic head conditions for the PAT.

The pressure fluctuation sensors were placed on the volute in the same vertical plane, where sensor 1 (DYT1) and sensor 2 (DYT2) are 135° apart, as shown in Figure 1. The circular arrow inside the circle represents the direction of flow in the pump working as a turbine.

All sensor signals were acquired simultaneously by a National Instruments® CompaqRio® 9045 data acquisition unit. The signals from the pressure fluctuation, torque, and encoder sensors were acquired at a rate of 102.4 k samples/s. Signals from the pressure transducers and flow meter were acquired at 50 k samples/s.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>External diameter impeller</td>
<td>0.14764 m</td>
</tr>
<tr>
<td>Reference diameter impeller</td>
<td>0.08104 m</td>
</tr>
<tr>
<td>Number of blades (backswept)</td>
<td>6</td>
</tr>
<tr>
<td>Inlet diameter (low-pressure nozzle)</td>
<td>0.0635 m</td>
</tr>
<tr>
<td>Outlet diameter (high-pressure nozzle)</td>
<td>0.0508 m</td>
</tr>
</tbody>
</table>

Table 2. Geometric characteristics of the PAT.

Figure 1. Diagram of the location of the pressure fluctuation sensors on the volute.
2.2 $Q_{ED}$-$n_{ED}$ characteristic

The PAT characteristic curve was made based on dimensionless factors of velocity ($n_{ED}$) and discharge ($Q_{ED}$), according to IEC 60193 [35], as defined below:

$$n_{ED} = \frac{nD}{E^{0.5}}$$

(1)

$$Q_{ED} = \frac{Q}{D^2E^{0.5}}$$

(2)

where $n$ is the rotation frequency (s$^{-1}$), $D$ is the reference diameter of the impeller (m), $E$ is the specific energy (N.m/kg), and $Q$ is the discharge (m$^3$/s). The characteristic curve was calculated with 60 operating points (OP), which had flow rates between 0.0024 and 0.0061 m$^3$/s, rotation frequencies between 5.04 and 33.85 s$^{-1}$, and a hydraulic head around 8 m. The characteristic curve corresponds to quadrants 3 and 4 defined by IEC 60193 [35], with the operating modes given in Table 3.

2.3 Spectral analysis

Spectral analysis is performed to detect periodic or quasi-periodic components in a signal; therefore, it is important to differentiate these components from narrowband random contributions [36]. In practice, a simple way to identify periodic or quasi-periodic components is by means of a power density spectrum (PSD) or also known as an auto spectrum. In the PSD, the periodic or quasi-periodic component is identified as a sharp and clearly distinguishable peak. Other spectral representations derived from the PSD, such as the power spectrum (PS), linear spectrum (LS), and linear density spectrum (LSD) [37], can also be used to identify this type of component. In this work, PS was used to identify periodic components and PSD was used to measure the instability associated with hydrodynamic phenomena.

For the spectral analysis, we used a time series of 2,621,440 records (25.6 s) divided into 2.5 parts and overlapped by 50%. In this way, four segments were obtained, and the spectrum was estimated for each segment and then averaged. In addition, the Hanning window was used to fix possible discontinuities in the data series. The frequency resolution was 0.09765625 Hz.

2.3.1 Fourier analysis

In turbomachines, Fourier analysis assumes that the fluctuations are periodic and linear [34]. With this assumption, instability can be considered as the sum of instabilities from different types or sources. In other words,

<table>
<thead>
<tr>
<th>Quadrant</th>
<th>Name</th>
<th>Operation mode</th>
<th>Remark</th>
</tr>
</thead>
<tbody>
<tr>
<td>3</td>
<td>Turbine</td>
<td>Turbine</td>
<td>Discharge, velocity, and torque positives</td>
</tr>
<tr>
<td></td>
<td>Runaway</td>
<td></td>
<td>Discharge and velocity positives. Torque = 0</td>
</tr>
<tr>
<td></td>
<td>Turbine brake</td>
<td></td>
<td>Discharge and velocity positives. Torque negative</td>
</tr>
<tr>
<td>4</td>
<td>Reverse pump</td>
<td>Reverse rotation</td>
<td>Velocity positive. Discharge and torque negatives</td>
</tr>
</tbody>
</table>

Table 3. Operation modes in quadrants 3 and 4.
\[ \tilde{x} = \tilde{x}_1 + \tilde{x}_2 + \ldots + \tilde{x}_n = \sum_{i=1}^{n} \tilde{x}_i \]  

(3)

where \( \tilde{x} \) is the total instability of the variable \( x \) and \( \tilde{x}_1, \tilde{x}_2, \ldots, \tilde{x}_n \) are the instabilities by type or source. Hence, when passing from time domain to frequency domain, the spectral representation of the instability of the interest variable contains the spectral components of the instabilities by type or source. Thus,

\[ \tilde{x} \xrightarrow{F} \tilde{X} \]  

(4)

\[ \tilde{X} = \tilde{X}_1 + \tilde{X}_2 + \ldots + \tilde{X}_n = \sum_{i=1}^{n} \tilde{X}_i \]  

(5)

where Eq. (4) denotes the shift from time domain to frequency domain by means of the Fourier transform. \( \tilde{X} \) is the frequency domain representation of the total instability of variable \( x \) and \( \tilde{X}_1, \tilde{X}_2, \ldots, \tilde{X}_n \) are the frequency domain contributions of instabilities by type or source. The Fourier transform estimation was performed by means of the Matlab\textsuperscript{®} fft function.

2.3.2 Dimensionless representation of frequency and pressure fluctuations

Frequency and pressure fluctuations were represented dimensionless. The frequency of the spectra was defined in terms of the frequency coefficient (\( f_n \)), according to IEC 60193 [35]:

\[ f_n = \frac{f}{n} \]  

(6)

where \( f \) is the frequency of the spectral component (s\(^{-1}\)) and \( n \) is the rotation frequency of the rotor or impeller (s\(^{-1}\)). The pressure fluctuation signals were represented by pressure fluctuation factor (\( \tilde{P}_E \)), which is defined by IEC 60193 [35] as:

\[ P_E = \frac{\tilde{p}}{\rho E} \]  

(7)

where \( \tilde{p} \) is the pressure fluctuation (N/m\(^2\)), \( \rho \) is the density (kg/m\(^3\)), and \( E \) is the specific energy (N.m/kg).

2.3.3 Phase analysis

The phase shift angle between corresponding spectral components was computed using the coefficients of the Fourier analysis. Phase analysis is useful to determine whether hydrodynamic phenomena are moving within the volute or whether they manifest as pressure pulses that simultaneously affect all parts of the system. In the first case, the phase difference is expected to be approximately equal to the angular separation between the sensors, and in the second case, approximately equal to zero. A negative phase lag means that the first signal is ahead of the second, and a positive phase lag is the opposite.
2.4 Measurement of flow instabilities due to low-frequency phenomena

The spectral analysis yields the spectral components or bands associated with hydrodynamic phenomena. To determine the flow instabilities, the signals are filtered into frequency bands containing the components of interest, and their instability in the time domain and then in the frequency domain is estimated on the filtered signals. The signals were filtered in the interest bands with digital FIR passband filters, designed with the Matlab\textsuperscript{®} `designfilt` function. Filtering was performed on the signals from the pressure fluctuation sensors to determine the contribution of low-frequency phenomena to the total instability at each OP.

According to Hasmatuchi et al. [38], one measure of instability is the standard deviation. However, this statistic has the disadvantage that it does not have linear characteristics. That is, the standard deviation of the sum of the filtered signals is not equal to the sum of the standard deviations of each of the filtered signals. To overcome this limitation, the variance (Var) can be used as an indicator of the instability, since this statistic is linear in nature, and the following can be verified:

\[
\text{Var} \left( \sum_{i=1}^{n} \tilde{x}_i \right) = \text{Var}(\tilde{x}_1) + \text{Var}(\tilde{x}_2) + \ldots + \text{Var}(\tilde{x}_n) \quad (8)
\]

where \(\tilde{x}_1, \tilde{x}_2, \ldots, \tilde{x}_n\) are the filtered signals in the frequency bands that were identified in the spectral analysis.

Considering that the area under the power density spectrum (PSD) is equal to the mean squared value of the signal [39], the variance of a zero-mean series can be used to measure the instability. This relationship between the area under the PSD and the variance of a zero-mean series allows for comparing the estimated instabilities in the frequency domain and in the time domain.

For estimation in the frequency domain, it would be enough to determine the area under the PSD in a frequency band associated with a component of interest, and this would be its approximate measure of instability. And as seen in Eq. (5), the linear nature of Fourier analysis allows the summing of the spectral power associated with the different phenomena to estimate the total instability.

To compare the estimated instabilities in the time and frequency domains, it is necessary to use the same units of measurement of the signals in the different domains. For this purpose, the technique proposed by Heinzel et al. [37] was used to convert the power spectral units into engineering units. In this research, the power spectrum is used since its representation in engineering units coincides with the units of variance.

3. Results

3.1 Q_{\text{ED}-n_{\text{ED}}} characteristic curve

Figure 2 shows the PAT operating modes in quadrants 3 and 4 of the Q_{\text{ED}-n_{\text{ED}}} characteristic. The OPs corresponding to each operating mode are distinguished by colors. According to Greitzer [40], a hydraulic system can exhibit two forms of instability. The first, static instability, occurs when a small change in the flow rate at one OP can cause an increase in pressure forces and deviate the system to another OP farther away. The second, dynamic instability, occurs when a disturbance oscillates continuously increasing in amplitude. These forms of instability are associated with
hydraulic transients and clues can be found by analyzing the $Q_{ED}-n_{ED}$ and $T_{ED}-n_{ED}$ characteristic curves [41–43]. The occurrence of static instability is a necessary but not sufficient condition for the occurrence of dynamic instability [43].

A practical method to identify static instability in the $Q_{ED}-n_{ED}$ curve is by the slope of a segment tangent to the runaway. If the slope is negative, the system is considered stable; if the segment is vertical, it is critical; and if the slope is positive, the system is unstable [41, 43]. The characteristic curve of an unstable system acquires the form “s,” which means that for the same value of $n_{ED}$, several values of $Q_{ED}$ can occur. In some cases, the $Q_{ED}$ can reach values in all operating modes (turbine, turbine brake, and reverse pump). According to this criterion, the system under study is stable (see Figure 2) and, therefore, the system does not exhibit transient phenomena. This suggests that the instabilities occurring in the system are stationary.

3.2 Spectral analysis

3.2.1 Identification of periodic or quasi-periodic components

This analysis was based on the waterfall spectra of the pressure fluctuation signals from DYT1 and DYT2 sensors (see Figures 3 and 4). In Figures 3 and 4, the spacing between the spectra (OP axis) is defined by the Euclidean distance between the OPs in the $Q_{ED}-n_{ED}$ plane. Comparing Figures 3 and 4, it is observed that there is a frequency correspondence in the components with higher spectral power, suggesting the existence of periodic or semi-periodic hydrodynamic phenomena detected by the two sensors. From this inspection, three spectral groups of interest were defined. Group 1 (G1) is composed of OPs from 32 to 48 in the band $0.76 \leq f_n \leq 0.86$, group 2 (G2) by OPs from 38 to 48 in the band $1.61 \leq f_n \leq 1.71$, and group 3 (G3) by all OPs in the band $5.95 \leq f_n \leq 6.05$. Figures 3 and 4 also emphasize the groups of interest with shading.

G1 can be divided into two subgroups, with the following characteristics:

- Subgroup 1: It includes OPs 32–41 and describes a phenomenon that appears at OP 32, reaches its maximum instability at OP 38, and then decays up to OP 41. Its frequency coefficients vary between 0.7995 and 0.8378.
Subgroup 2: It is constituted by OPs 42–48. It describes a phenomenon that initiates at OP 42, reaches its maximum instability at PO 44, and decays until it reaches OP 48. Its frequency coefficients vary between 0.7798 and 0.8381.

G2 contains OPs 38–48, where the way the spectral power is distributed suggests the same hydrodynamic phenomenon. The phenomenon begins at OP 38, increases in power up to OP 41, and from there decreases to OP 48. It occurs in the lower part of the turbine-brake mode, and its frequency coefficients range between 1.6438 and 1.6834.

G3 comprises all operating points and corresponds to the blade passage with $f_n = 6$. It is distributed throughout all operating modes in quadrants 3 and 4 and is most noticeable in turbine-brake and reverse pump modes. Near the runaway, the spectral power increases gradually until it reaches the last operating point in the reverse pump.
mode. In the high part of the turbine operating mode (OPs 3–9), some spectral powers are observed that outstand with respect to those of the other OPs in this part.

Another finding from comparing Figures 2 and 3 is the spectral power of the components of interest in the DYT2 sensor signal spectra is higher than that of the DYT1 sensor. On average, the spectral powers in DYT2 are 2.6 times those of DYT1.

### 3.2.2 Characterization of periodic or quasi-periodic components

The characterization was performed for the groups of interest under the hypothesis that the components are the representation of hydrodynamic phenomena detected by the two sensors. For this purpose, power spectra were used, and three criteria were applied for the selection of periodic components. First, components with sharp, clearly distinguishable, and isolated peaks [36], that is, without contiguous spectral components of similar spectral power. Second, corresponding components in the spectra of both sensors in terms of frequency factor, and third, coherence between them greater than or equal to 90%.

The characterization of G1 and G2 was performed by phase analysis. In G1, OPs 32, 33, 34, 36, 37, 38, 39, 44, 45, and 46 satisfy the periodicity criteria. These subsynchronous components have frequency coefficients between 0.8018 and 0.8378 and signal coherence between 90.3 and 99.7%. In OP 34, 37, 38, 39, and 45 the signal from DYT1 is ahead of DYT2, and they show phase shifts between $-150.6^\circ$ and $-135.2^\circ$, approaching $135^\circ$ when considering the separation of the sensors in the opposite direction of flow from DYT1. For OPs 32, 33, 36, 36, 44, and 46, the phase shifts are between 206.3 and 236.9°, which are approximately 225°, the physical separation of the sensors in the opposite direction of flow but starting at DYT2. These components were subjected to an additional test, the wave number determination. This test serves to identify phenomena that sometimes decompose in rotating cells, such as rotating stall [44, 45]. For these phenomena, the wave number is equal to the number of rotating cells. The wave number was estimated as follows:

$$K = \frac{f_{n_p}}{f_n}$$

(9)

where $K$ is the wave number, $f_{n_p}$ is the passage frequency, and $f_n$ is the frequency identified in the PS. The passage frequency is the frequency perceived by the sensors and is estimated from the correlation between signals in a band containing the component of interest. Details of the method can be found in Refs. [44, 45]. For the periodic components of G1, it was found the passage frequency is approximately equal to the PS frequency, so the wave number is equal to one. The evidence collected suggests the existence of a one-cell subsynchronous hydrodynamic phenomenon ($K = 1$), moving around the volute in the opposite direction of flow. Full characterization of the phenomenon requires further studies that are beyond the scope of this investigation.

In G2, OP 39–47 fulfill the periodicity criteria. These components present frequency coefficients between 1.6438 and 1.6742, and coherence between 92.00 and 99.51%. OPs 39, 41, 43, 44, 46, and 47 present offsets between 108.1 and 161.8°, and OPs 40, 42, and 45 between $-228.3$ and $-204.8^\circ$. Based on the same arguments used in the characterization of the periodic components of G1, it can be concluded the components of this group correspond to a one-cell phenomenon ($K = 1$) moving in the direction of the flow.
The G3 components correspond to instabilities due to rotor-stator interaction (RSI) blade excitation. In this case, the excitation is probably caused by the asymmetry of the volute [46] since the PAT has no guide vanes. The effect of the rotor blades on the volute produces a periodic disturbance or force whose frequency, expressed in terms of frequency coefficient, is given by:

\[ f_{n,k} = Z_b k, \quad (k = 1, 2, 3, \ldots) \]  

where \( f_{n,k} \) is the frequency coefficient of the disturbance due to the blades, \( Z_b \) is the number of blades, and \( k \) is harmonic. In our case, \( f_{n,1} = 6 \), is the blade passage.

Notice in Figures 3 and 4 that OPs 20 (runaway) and 21 (next to runaway) do not show significant perturbations compared to the interest group components. On an appropriate scale, it can be seen that OPs 20 and 21 have a component that matches the periodicity criteria, with frequency coefficients of 0.776 and 0.758, respectively, coherence of 99.64% and 98.21%, and phase shifts of 0.1 and 3.3°, suggesting an in-phase phenomenon. This phenomenon is probably a surge, which is characterized by pressure and flow fluctuations that affect all parts of the hydraulic system simultaneously [33]. Figures 5 and 6 show the PS of these points, where the highlighted spectral components are clearly distinguishable.
3.3 Measurement of flow instabilities due to low-frequency phenomena

The instability measure of the prominent spectral components (G1, G2, and G3) at each OP was estimated based on the filtered signals at 2 Hz bandwidth. The bandwidth was equally distributed with respect to the spectral component of interest. In other words, the center of the filtering band corresponds to this component. Notice a distinction in the terms “interest band” and “filtering band.” The former refers to the band defined by frequency coefficients \( f_n \) in which the spectral components of interest are found, and the latter refers to the frequency band in which the filtering was done around the prominent component identified in the interest band.

The filtered signals were used to estimate the instability due to periodic or quasi-periodic phenomena in the time domain and the frequency domain. In the time domain, we considered the filtered signals and computed the variance. In the frequency domain, we computed the PSD of the filtered signal and then estimated the area under the spectrum. Figures 7 and 8 depict the three-dimensional representation of the instability estimates for the DYT1 and DYT2 sensor signals, respectively, based on the \( Q_{\text{ED}}-n_{\text{ED}} \) characteristic curve and classified according to interest bands.

Comparing the estimates of instabilities in the time domain with the estimates in the frequency domain (Figure 7a vs. b and Figure 8a vs. b), it is observed they are not equal, but quite similar. Regarding the sum of variances, we obtain a mean error of 4.9% and a standard deviation of the errors of 13.5% for DYT1, and a mean error of 4.0% and a standard deviation of the errors of 11.7% for DYT2.

Looking at Figures 7a vs. 8a (time domain instabilities estimation) and Figures 7b vs. 8b (frequency domain instabilities estimation), it is suggested that the distribution of the pressure pulses is not homogeneous in the volute, since the magnitude of the instabilities in DYT1 is smaller than their corresponding ones in DYT2. If the most unstable operating zone is considered (OPs 32–48), the magnitudes of the instabilities in DYT2 are larger than their corresponding ones in DYT1 by 1.74–3.32 times.

In terms of the contribution of periodic or quasi-periodic phenomena to the total instability, DYT2 is analyzed, since the magnitude of instabilities perceived by this sensor are higher than those perceived by DYT1. In this case, the sum of the instabilities estimated with the filtered signals is compared with those of the original unfiltered signals, which were used to estimate the total instability. Figure 9 shows a 3D representation of the sum of the periodic or quasi-periodic instabilities with respect to the total instability. Total instability estimations in the time and frequency domain yield a mean error of 0.3% with a standard deviation of the errors of 4.8%.

Given the contribution of periodic or semi-periodic instabilities to total instability (see Figure 9), three zones of operation can be defined. Table 4 describes these zones.

4. Conclusions

The following conclusions can be drawn from the evidence collected:

At least four types of periodic or quasi-periodic phenomena exist in the PAT under study, which are described below:
In OPs 20 (runaway) and 21 (next to runaway), an in-phase phenomenon is detected, perhaps a surge with frequency coefficients of 0.776 and 0.758, respectively. Regarding the total instability, the disturbance magnitude of this phenomenon reaches 13.2 and 12.9% in DYT1 and 15.9 and 15% in DYT2, respectively.

Figure 7.
Estimation of periodic or quasi-periodic instabilities for DYT1. Above, estimation in the time domain. Below, estimation in the frequency domain.

- In OPs 20 (runaway) and 21 (next to runaway), an in-phase phenomenon is detected, perhaps a surge with frequency coefficients of 0.776 and 0.758, respectively. Regarding the total instability, the disturbance magnitude of this phenomenon reaches 13.2 and 12.9% in DYT1 and 15.9 and 15% in DYT2, respectively.
At all operating points, blade excitation is present due to rotor-stator interaction, with $f_n = 6$. The magnitude of this disturbance gradually increases from the middle zone of the turbine-brake mode (OP 23) to the last OP in the reverse pump zone (OP 60). In the upper part of the turbine mode, there is a rapid increase and decay of this disturbance at OP 2–6. Between OP 18 and 22 the magnitude of the disturbance is practically negligible.

Figure 8. Estimation of periodic or quasi-periodic instabilities for DYT2. Above, estimation in the time domain. Below, estimation in the frequency domain.
In G1, the evidence suggests the existence of a subsynchronous phenomenon that moves inside the volute in the opposite direction to the flow direction, with phase shift angles close to the physical separation of the sensors. In G2, a supersynchronous phenomenon is present and something like G1 occurs, only that the movement of the phenomenon is in the same direction of the flow.

In terms of the contribution of periodic instabilities to total instability, three zones can be identified. In the first one, constituted by OPs 1–31 (turbine, runaway, and

![Figure 9. Contribution of periodic or quasi-periodic instabilities in the total instability for DYT2. Above, estimation in the time domain. Below, estimation in the frequency domain.](image)
upper part of turbine-brake), periodic instabilities represent approximately 1–42% of the total instability. In the second zone, between OP 32 and 48 (middle and lower turbine-brake mode), there are at least three hydrodynamic phenomena whose contribution to the total instability varies between 1.2 and 42.2%, if the variance is considered, and between 1.1 and 39.2%, if the area under the PSD is considered.

The third zone covers OP 49–60 (last OPs in the turbine brake and all OPs of the reverse pump) where blade excitation predominates and presents contributions to the total instability ranging from 7% to 28%.

Conflict of interest

The authors declare no conflict of interest.

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