

## The rotor configuration and the stability range in the seal of a tidal turbine

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**Abstract:** In a tidal turbine of electric power production, a fundamental component of its safe operation is the sealing seal, so that water does not penetrate the turbine machinery and thus cause a catastrophic breakdown. The current turbines that are started to be installed on the coast reach power generation of 2MW and its sealing seals, in addition to having sealing seals of elastomers, have a pressurized circuit above the pressure of the water at its point of installation, as a guarantee of this essential sealing. It is studied in this experiment as the configuration of the supports of the transmission shaft of the rotor/seal affects the results of dynamic phenomena known as "oil whirl" and "oil whip". Rotor start-up date are given in the form of cascade spectra indicating thresholds of stability and whirl/whip forward sine-sweep perturbation of the rotors, while maintaining a constant rotational speed show the existence of two distinct resonances. The first resonance occurs at the forward perturbation frequency close to  $\frac{1}{2}$  rotation speed ("oil whirl" resonance) and the second resonance occurs at forward perturbation frequency close to the natural bending frequency determined by the rotor mass and stiffness ("oil whip" resonance). The configuration and position of the support bearings of the transmission shaft reveal different frequencies to which are produced these phenomena of "oil whirl" and "oil whip", which should be taken into account in the design of this turbine to ensure the sealing of its sealing seal.

**Keywords:** Tidal Turbines; Rotor dynamics; dynamic stiffness; stability threshold; oil whirl; oil whip.

### 1. Introduction

A failure to sealing seal of a tidal turbine produces a total collapse of this turbine, so in this type of machines, in addition to sealing joints on the shaft, such as synthetic polymers, is installed a closed circuit oil lubrication pressurized, (Figure 1). Its remote surveillance, by means of pressure switches, is essential to guarantee in the cavity a higher oil pressures than that of the maritime environment.

Even before the pressure of the oil circuit alarm of a situation of lack of watertightness, it is also necessary to monitor other phenomena that can precede this situation. It must be installed in the radial bearing immediately to the sealing seal, two induction transducers at  $90^{\circ}$  and one of phase reference, in order to detect the value of the stability threshold preceding the phenomena of "whirl" and "whip".

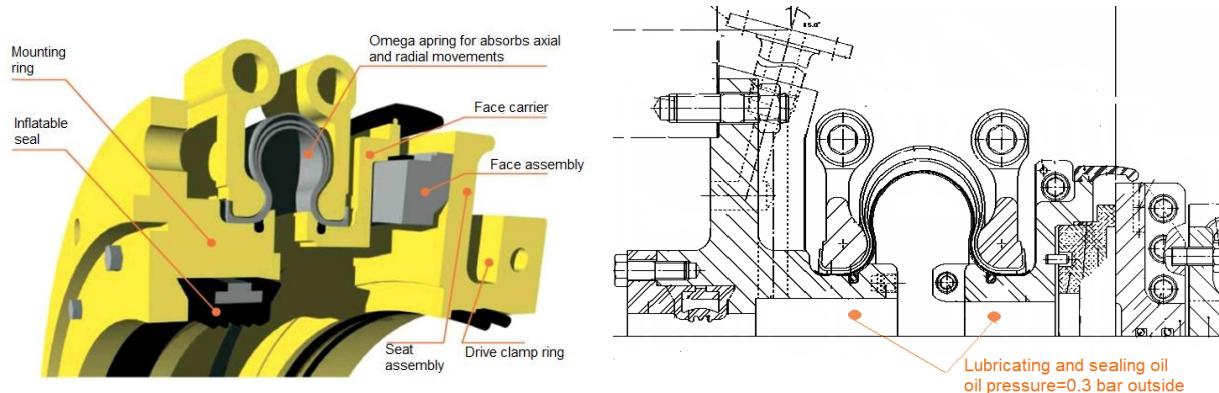


Figure 1. Sealing Seal

There are certain radial entity vibrations, which are not related to the imbalance (1X response) or other periodic forces, such as misalignment (1X and 2X response) or a curvature of the shaft (1X and 2X response),

which correspond to frequencies subsynchronous between which the friction between static and rotating parts has already been mentioned [1]. They are self-excited vibrations and are produced by the dynamic flow of fluids, including lubricating oils, vapour and gas, which transfer rotational energy to the lateral vibrations, and therefore are possible to happen in the pressurized circuit of the sealing seal.

The physical phenomenon happens because when a shaft rotates in a closed fluid environment like that of the seal, this shaft drags the fluid in the movement of rotation. The movement of the fluid is three-dimensional but the transmission of the circumferential component will be the most significant and generates a rotational force that will be detected in the radial vibration of the shaft.

It is assumed that there is no phenomenon of imbalance or others. The system of a tidal turbine is due to model of the Figure 2, that identifies the mechanical seal close to the rotor and a fixed support that would be that of the gearbox, i.e. the so-called "slow shaft" of this turbine. The movement of this system for its first mode of vibration, is: [2]

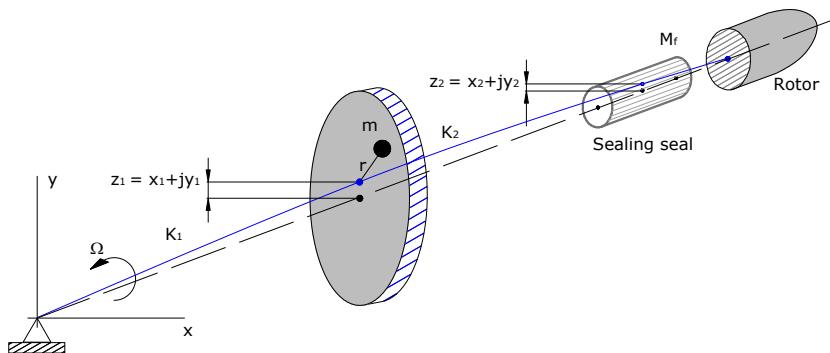


Figure 2. Rotor model of a tidal turbine

$$\text{Rotor: } M_1 \ddot{z}_1 + D_S \dot{z}_1 + (K_1 + K_2)z_1 - K_2 z_2 = mr\Omega^2 e^{j\Omega t} \quad [1]$$

$$\begin{aligned} \text{Sealing seal: } & M_2 \ddot{z}_2 + M_f (\ddot{z}_2 - 2j\lambda\Omega\dot{z}_2 - \lambda^2\Omega^2 z_2) + [D + \psi_2(|z_2|)](\dot{z}_2 - j\lambda\Omega z_2) + \\ & [K_0 + K_f + \psi_1(|z_2|)]z_2 + K_2(z_2 - z_1) = 0 \end{aligned} \quad [2]$$

where:

$K_0$ , the linear stiffness of the film for null eccentricity,

$\Omega$ , the shaft rotation speed,

$\lambda$  is the application coefficient at the fluid velocity in the seal,

$\lambda\Omega$  the average velocity of the flow of the pressurized fluid,

$K_1$  and  $K_2$  are the partial stiffness of each section of the rotor to the lateral movement,

$M_f$  the mass,  $K_f$  and  $D$ , the stiffness and damping of the pressurized fluid film.

$z_1(t)$  and  $z_2(t)$ , represent radial displacement of shaft-rotor and seal,

$M_1$  and  $M_2$ , rotor masses,

$D_S$  is the rotor damping,

$\psi_1$  and  $\psi_2$  are non-linear functions of the radial stiffness of the fluid oil wedge as a function of the eccentricity of the shaft.

$m, r$ : disturbance mass and situation radius.

The natural frequencies of the phenomena called "oil whirl", in which according to equation [3] will be a fraction of the value of the rotation speed  $\Omega$ , and the one of the "oil whip" equation [4], it is roughly identified with the first critical frequency of the rotor [3]:

$$\omega_1 = \lambda\Omega \quad [3]$$

$$\omega_{2,3} \approx \pm \sqrt{(K_1 + K_2 + K_f)/M_1} \quad [4]$$

$\Omega_{ST}$  indicates the value of the rotation speed for the *stability threshold*, in which the "oil whirl" phenomenon begins: [4]

$$\Omega_{ST} = \frac{1}{\lambda} \sqrt{\frac{K_1}{M_1} + \frac{K_2 K_0}{M_1 (K_2 + K_0)}} = [5]$$

The rotor will have an unstable movement for  $\Omega > \Omega_{ST}$ . The first term in the equation [5],  $K_1/M_1$ , is meanly dominant. The second term contains the influence of two stiffness:  $K_2$  y  $(K_0 - M_2 K_1/M_1)$ . Generally, and more in the case of the tidal turbine analyzed, the  $K_0$  stiffness of the mechanical seal has a very small value, in addition to its mass  $M_2$  compared to the total rotor  $M_1$ , so the expression  $(K_0 - M_2 K_1/M_1)$  is almost null and much smaller than  $K_2$ , so that the equation [5] it can be reduced, finally and roughly, to:

$$\Omega_{ST} \approx (1/\lambda) \sqrt{K_1/M_1} [6]$$

This stability threshold is determined by the partial stiffness of the rotor  $K_1$ , its mass, and the average velocity coefficient " $\lambda$ " of the fluid.

The solutions seen of the model analyzed of the system rotor/seal have three important values: the first " $s_1$ ", has an imaginary part very close to " $\lambda\Omega$ ", and corresponds with the frequency of the "oil whirl". The second and third eigenvalue " $s_{2,3}$ ", have an imaginary part of a value very close to  $(\pm\sqrt{(K_1 + K_2 + K_f)/M_1})$ , which is related to the first natural frequency of the rotor, in its first mode of vibration, and belongs to the frequency of the "oil whip". The radial  $K_B$  stiffness pressurized oil wedge depends on the eccentricity and increases with this (Figure 3): [3]

$$K_B \equiv K_0 + \psi(A) = K_2 \frac{M\omega^2 - K_1}{K_1 + K_2 - M\omega^2} [7]$$

where the intercession of both graphs, marked as " $A_{whirl}$ ", is the solution of the equation [7]. The amplitude "A" of the vibration corresponding to the phenomenon of "whirl" is calculated for the value of the frequency of those vibrations self-excited:  $\Omega = \lambda\Omega$ , and the equation [7] results:

$$K_B = K_0 + \psi(A_{whirl}) = K_2 \frac{M(\lambda\Omega)^2 - K_1}{K_1 + K_2 - M(\lambda\Omega)^2} [8]$$

It shows in Figure 3 different cases of the value of the rotation speed  $\Omega$ , with respect to the *stability threshold*  $\Omega_{ST}$ :

- For  $\Omega < \Omega_{ST}$ :

$$\Omega < \Omega_{ST} \equiv \frac{1}{\lambda} \sqrt{\frac{K_1}{M} + \frac{K_0 K_2}{M(K_0 + K_2)}} = \text{rotation speed for the stability threshold},$$

The right part of the equation [8] and [7] is smaller than  $\{K_0 + \psi(A)\}$ , the two functions do not intersect (Figure 3), and this range corresponds to the stable performance zone of the rotor before reaching that the *stability threshold*.

- For  $\Omega > \Omega_{ST}$ , when  $\Omega \approx \omega_{whirl} = \lambda\Omega$  (equation [3]), both functions of the equation [8] intersect (Figure 3), and indicate in this figure the level of the amplitude corresponding to the "whirl" ( $A_{whirl}$ ).
- For values of  $\Omega \approx \omega_{whip} = \sqrt{(K_1 + K_2 + K_f)/M}$  (equation [4]), the right part of the equation [8] increments towards infinity, which is when the shaft is already completely close to the surface of the seal. This last value corresponds to the transition of the frequencies of the phenomenon of "whirl" to the "whip", which are the values of the equations [3] and [4] (Figure 3).

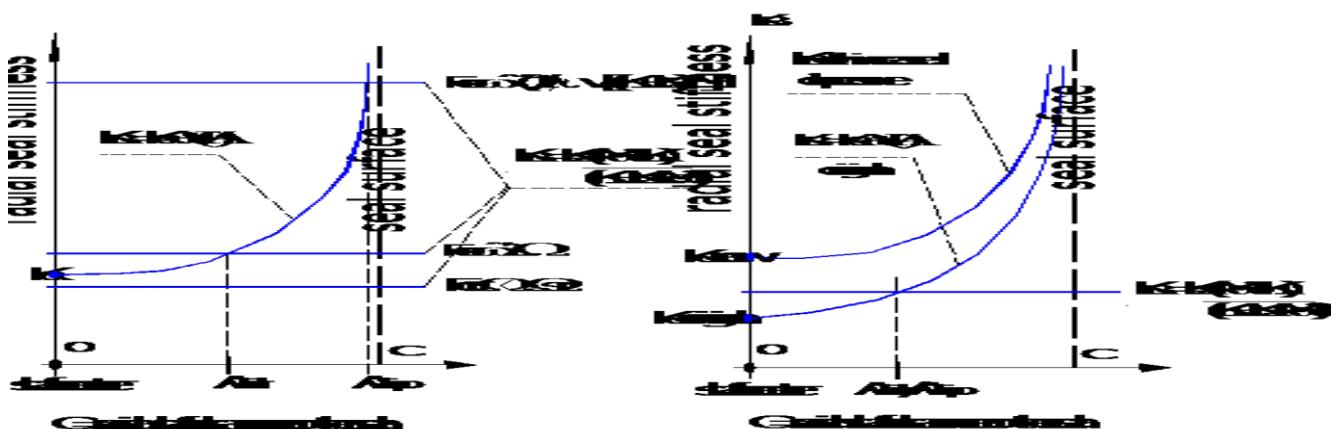


Figure 3. Amplitude range of the whirl and whip

In the equation [7], it is also observed that when the frequency "Ω" is equal to the frequency value of the "whip" (equation [4]), the term of the right of this equation [7] tends to infinity. Then, when the radial stiffness of the  $K_B$  fluid tends to infinity, it is when the shaft approaches the surface of the static part of the mechanical seal, and the amplitude of these self-excited vibrations is limited by the mechanical seal clearance ("c"), and therefore  $A_{\text{whip}} \approx c$ .

In the classic study of rotor dynamics, of transverse isotropic characteristics, the dynamic stiffness of the fluid in seals or bearings will have two components: a direct one mainly referred to a combination of the spring effect, damping and mass in radial direction; and another quadrature component, which contains the term of damping, which are [5]

$$\text{Direct: } K_D = (F_{\text{pert}-z}/|z|) * \cos \alpha = \underbrace{K_B - (\omega - \lambda\Omega)^2 M_{fl}}_{\text{fluid parameters}} + \underbrace{K - M\omega^2}_{\text{rotor}} \quad [9]$$

$$\text{Quadrature: } K_Q = -(F_{\text{pert}-z}/|z|) * \sin \alpha = \underbrace{D(\omega - \lambda\Omega)}_{\text{fluid}} + \underbrace{D_s\omega}_{\text{rotor}} \quad [10]$$

being " $\alpha$ ", in these equations, the phase angle between the force and the response.

## 2. Experimental Setup

In a test rotor such as Figure 4, by applying a force known as a given mass imbalance, at different rotation speeds the components of the dynamic stiffness can be calculated [6].

- Direct Dynamic Stiffness:  

$$K_D = K_{DS} * \cos \theta_k = 92,28 \text{ (N/m)} * \cos 217^\circ = -89,00 \text{ N/m}$$
- Quadrature Dynamic Stiffness:  

$$K_Q = K_{DS} * \sin \theta_k = 92,28 \text{ (N/m)} * \sin 217^\circ = -24,35 \text{ N/m}$$

In the case of this study, in which this rotor is configured according to a horizontal shaft tidal turbine, this situation of rotational fluid instability can be given in the mechanical seal that makes seal sealing and in the outer ring of the multiplier gears. Its configuration responds to the outline of the Figure 4 that is a rotor with a rigid bearing, which would correspond to the gear train and another that simulates the mechanical seal lubricated with pressurized oil. In addition to a system of alteration of the dynamic forces that consists of an accessible disc to cause identified mass imbalances.

The rotor was driven by a 1/10 HP electric motor with a variable speed controller up to 5.000 rpm. A spring frame is installed to compensate for the gravity load on the rotor and center the shaft on the seal. The pressurized oil fluid is supplied through four radial points at a pump outlet pressure of 10 to 20 kPa. The oil temperature in a holding tank was 27 °C.

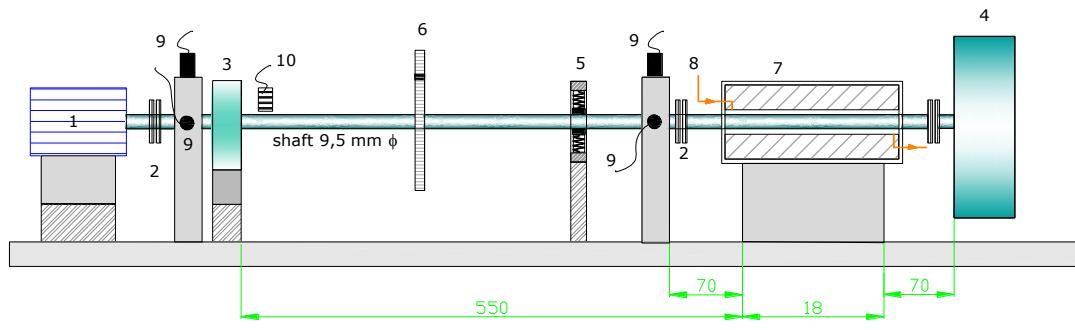


Figure 4. Test rotor

The equipments and instrumentations available are digital vector filter to separate the responses of different frequencies, spectrum analyzer amplitude/frequency, oscilloscope for observation of the shaft displacement, computer with software of GE/Bently for the acquisition and processing of vibration data, graphically and analytically.

The test speeds are made from 500 to 5000 rpm, with intervals of 500 rpm. The masses of imbalance controlled are  $m_e = 0.75, 2.35$  and  $7$  gr., placed to a radius of  $0.03$  m.; the oil density is  $794$  kg/m $^3$ , its operating temperature  $26.7$  °C, its viscosity  $0.05$  kg/m.s. (50 cpoises); oil pressure =  $10$  to  $20$  kPa; shaft weight  $0.58$  kg;  $1.81$  rotor weight; radial clearance in seal  $190$  µm; rigid bearing: length  $0.018$  m. and diameter  $0.0254$  m.

In this test we study the position of a second rigid bearing in two situations, one next to the seal and another in the middle of the shaft, in order to analyze the results of the frequency at which the stability threshold occurs and the phenomena of "oil whirl" and "oil whip".

The stability threshold for this model is  $\omega_R \approx (1/\lambda)\sqrt{K_1/M_d}$  and is identified as the first natural resonance frequency of the system. The test has been carried out with the different bearing situations, which influence the stiffness of the shaft  $K_1$  and  $K_2$ , so that the stability threshold will vary also in these cases.

### 3. Seal stability according to your design.

The amplitude/frequency response for different rotation speed values is shown below. For  $\lambda = 0.435$  values, the following frequencies of the phenomena are identified:

$$\omega_R = 1000 \text{ rpm, whirl frequency} = \lambda\omega_R = 435 \text{ cpm}$$

$$\omega_R = 2000 \text{ rpm, whirl frequency} = \lambda\omega_R = 870 \text{ cpm}$$

In this test, the action forced by the imbalance of masses is  $= 58.7 \times 10^{-6}$  kg.m. There are already other tests made of the stiffness of this test rotor for different situations, which have been obtained [6]:

- For the situation indicated in Figure 5 with a rigid support "3" which would correspond to the support of the gear multiplier train, and only another more flexible support that corresponds to the seal:

$l_1 = 0.550$  m,  $K_1 = 2627$  N/m and  $K_f + K_3 = 29281$  N/m  $\{K_1 \ll (K_f + K_3)\}$  in this configuration and  $M_d$  (rotor mass) =  $1.81$  kg.,  $K_f$  = stiffness of the pressurized oil wedge of the seal, so the second resonant frequency is:

$$\omega_{whip} = \sqrt{(K_1 + K_f + K_3)/M_d} = \sqrt{2627 + 29281 \text{ N/m}/1,81 \text{ kg}} = 1267.95 \text{ rpm}$$

A cascade spectrum diagram, for this configuration, is represented in Figure 6 [7], in which the stability threshold is produced at the rotation speed of:

$$\Omega_{ST} = (1/\lambda)\sqrt{K_1/M_d} = (1/0.435)\sqrt{2627 \text{ N/m}/1,81 \text{ kg}} = 87,57 \text{ rad/s} = 836,23 \text{ rpm}$$

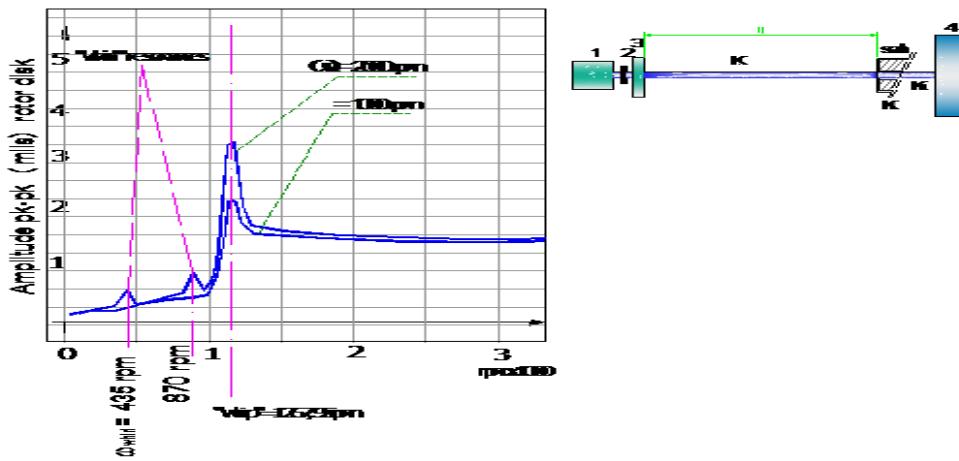


Figure 5. Case i) vibration response (Bodé diagram).

One can observe, in this analysis of spectrum, as if the stability threshold of the inertial rotation of the fluid of 836.23 rpm coincides very close with the frequency at which the phenomenon of the "whirl" begins. In addition, coincides also with that same value in the amplitude of first resonance of the Figure 5, and it is maintained until the appearance of the frequency corresponding to the "whip", of approximately 1300 cpm, in coherence with the analytical values obtained.

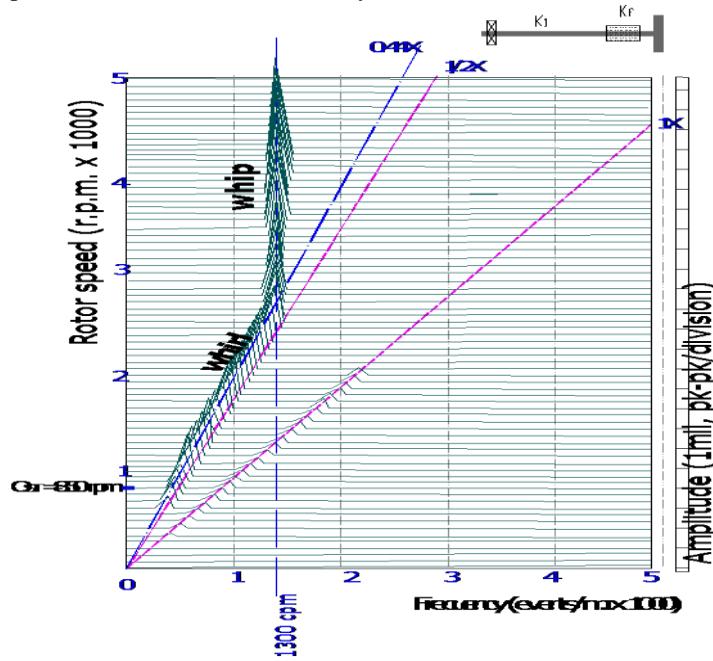


Figure 6. Cascade spectrum

- ii. In another test, and in order to increase the stability threshold, a new rigid bearing is installed in the proximity of the seal (Figure 7a), since the stiffness of a shaft between rigid supports is greater than between rigid and flexible, as corresponds to the pressurized seal with oil:

$$K_1 = 20672 \text{ N/m}; K_2 = 74951 \text{ N/m}, (K_2 \text{ far superior to } K_1, \text{ given the short length of } l_2 \ll l_1), K_f + K_3 = 29281 \text{ N/m}$$

$$\omega_{whip} = \sqrt{\frac{K_1 + K_2 + K_f + K_3}{M_d}} = \sqrt{\frac{20672 + 74951 + 29281 \text{ N/m}}{1,81 \text{ Kg}}} = 2508.69 \text{ rpm}$$

In addition, a stability threshold, at the speed of rotation, of:

$$\omega_{ST} = \left(1/\lambda\right)\sqrt{K_1/M_d} = \left(1/0.435\right)\sqrt{20672\text{ N/m}/1,81\text{ kg}} = 2346.07 \text{ rpm}$$

- iii. The third test is made by placing the rigid bearing in the middle of the shaft (Figure 7 b), and it is obtained:

$K_1 = 64552 \text{ N/m}$ ;  $K_2 = 54360 \text{ N/m}$ ; ( $K_1 \gg K_2$ , because the stiffness between two rigid supports is greater than between a rigid and a flexible one, the seal),  $K_f + K_3 = 29281 \text{ N/m}$

$$\omega_{whip} = \sqrt{\frac{K_1 + K_2 + K_f + K_3}{M_d}} = \sqrt{\frac{64552 + 54360 + 29281 \text{ N/m}}{1,81\text{ kg}}} = 2732.52 \text{ rpm}$$

and a stability threshold, at the speed of rotation, of:

$$\omega_{ST} = \left(1/\lambda\right)\sqrt{K_1/M_d} = \left(1/0.435\right)\sqrt{64552 \text{ N/m}/1,81\text{ kg}} = 3767.95 \text{ rpm}$$

These situations of cases ii) and iii) correspond to the amplitude/frequency responses of Figure 7, in which it is observed as the phenomenon of the "whip" occurs at speeds of rotation greater than in the first case i). In this case, no more rigid bearings were installed than that corresponding to the support of the gear train, and to the cascading spectrum diagrams of the Figure 8, in which it can also be seen how the stability threshold value is increased.

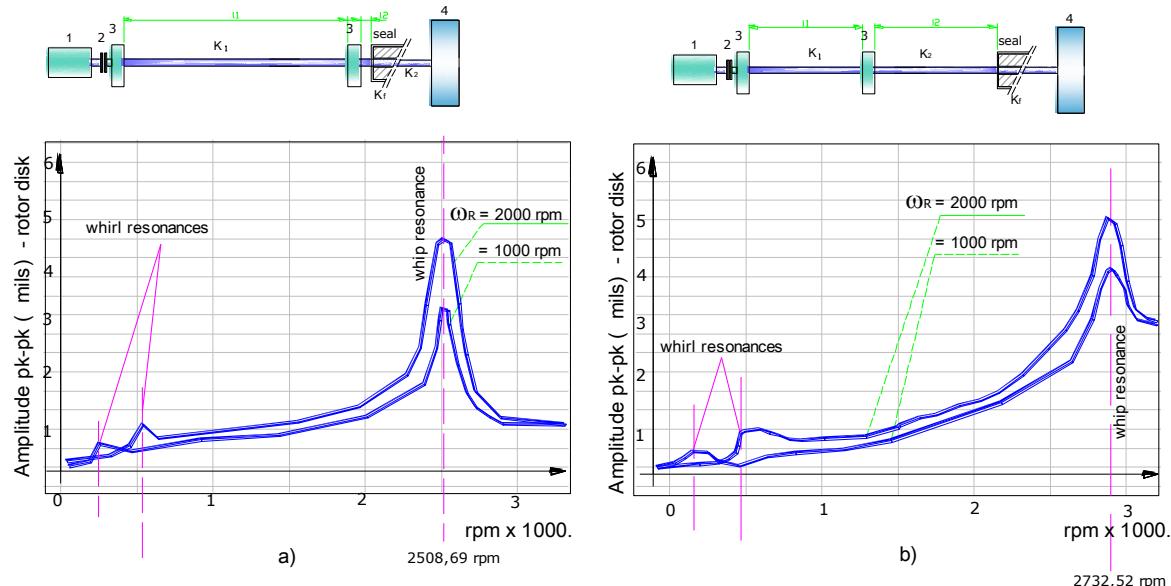


Figure 7. Amplitude/Frequency response.

In these diagrams of cascading spectrum of Figure 8, with a rigid bearing "3" close to the seal, and in the other case, with the rigid bearing placed in the middle of the shaft, it is observed:

- That the phenomenon of the "whirl" appears in the value of the velocity that corresponds to the stability threshold of this case ii), 2300 rpm, and is maintained until the appearance of the frequency corresponding to the "whip", of 2500 cpm, values that are identified with the analytical ones obtained.

In addition, for the position of the rigid bearing in the middle of the shaft, caso iii):

- The whip phenomenon occurs more frequently, near the 2800 cpm, and the stability threshold is now reached at rotor speeds close to 3800 rpm.

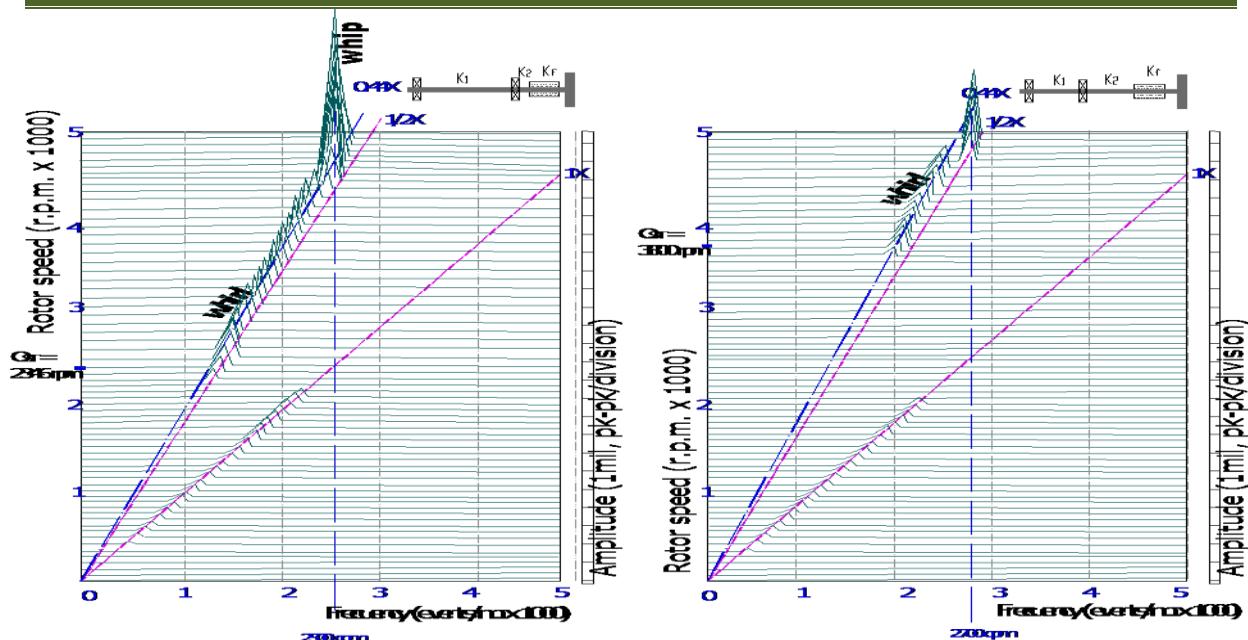


Figure 8. Cascade spectrum.

### 3.1. Control of the stability of the seal by pressurization.

In a stable system, the direct and quadrature components of the dynamic stiffness are not zero at the same frequency. The "stability margin" is defined as the interval between the two null values [8], which is where both  $Z_D$  as  $Z_Q$  have positive values. In the case analyzed is assumed from the inflection point of the parabola  $Z_D$  to the null value of the quadrature component = 4000-2000 = 2000 rpm (Figure 9).

If both, Direct Dynamic Stiffness and Quadrature Dynamic Stiffness become zero at the same frequency, the denominator of equation [11] is made zero. There is nothing then that restrains the amplitude of vibration and the response tends to infinite. This is what happens at the *Threshold Stability*, when the rotor speed starts into fluid instability and it begins the so-called "whirling" phenomenon to later reach the "whip".

On the other hand, the direct component of the dynamic stiffness, for the same mechanical seal, depends on the pressure of the fluid, so that, at greater pressure more stiffness, as indicated in Figure 9, while the quadrature component is shown independent of the pressurization of the fluid [9]. The stability threshold is shown in the Figure 9, as the common area between the two graphical representations of  $Z_D$  and  $Z_Q$ , which is where they present positive values, and therefore a dynamic stiffness of higher value thus provides responses of lower vibration amplitude:

$$\text{Response amplitude} = \frac{\text{Disturbing force}}{\text{Dynamic stiffness}} = \vec{R} = \frac{(mr_u\Omega^2)\angle\delta}{[K-M\Omega^2]+j[D(1-\lambda)\Omega]} \quad [11]$$

The oil pressure on the seal rises with the velocity square (fluid dynamics theory), and the stiffness of the pressurized fluid ( $K_B$ ) increases linearly with that pressure, so the direct component of the dynamic stiffness is also increased with the square speed (Figure 3). Once fluid pressure is controllable, the dynamic stiffness is also controllable, and as can be seen in Figure 9. Higher fluid pressures achieve wider stability zones for rotors, so that the simple monitoring of the value of the fluid pressure will already indicate the state of the dynamic stiffness and attenuation of the lubricating oil film, besides being able to obtain better stability margins.

The increase of the eccentricity of the shaft, not only reduces the coefficient  $\lambda$ , but also causes higher values of the stiffness and damping of the fluid, which provides greater stability to the rotor, a situation that is also produced by an increase in the pressure of that fluid or of alterations in its temperature [10].

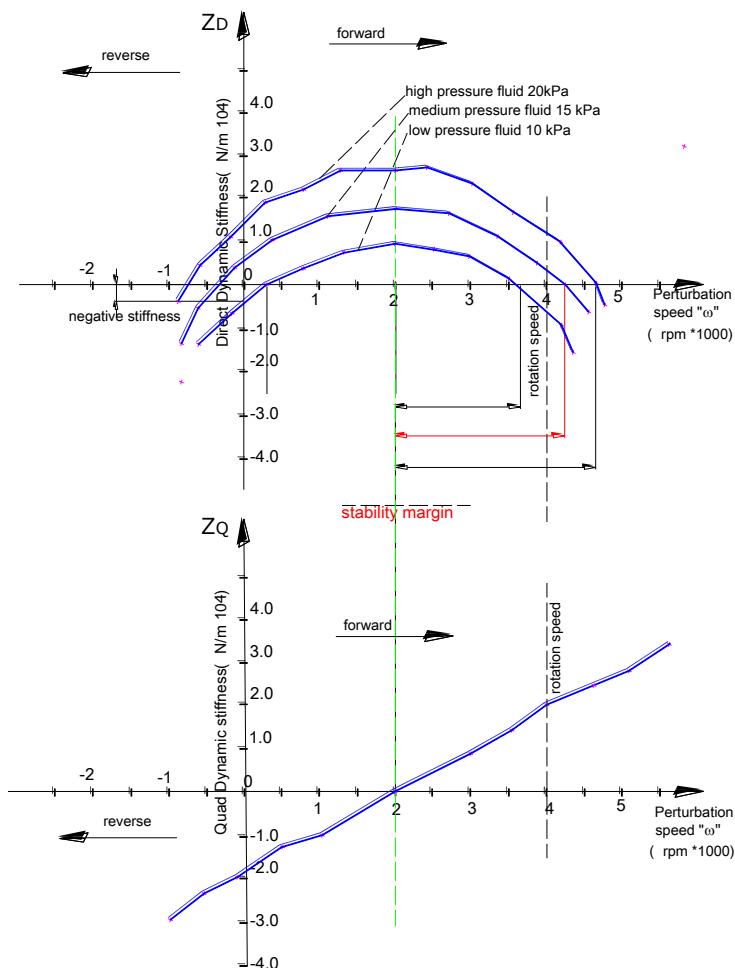


Figure 9. Dynamic stiffness and range of stability.

#### 4. Conclusions

In these tests, it is observed that the stability margin decreases as the rotor speed approaches the *Threshold Stability* that is where the phenomenon of whirl begins in its natural frequency.

A stable operation of the rotor will always occur at operating speeds in the range that is defined by the *Margin of Stability*.

It is therefore essential to know these parameters of a system, in order to achieve the stability of the rotation to the possible instabilities of a shaft with lubricating fluid and watertight as is the case of the tidal turbine.

The rotor configuration has an influence on the system response, as it modifies the rotor parameters. By installing a rigid bearing close to the seal, an increase in the seal's damping coefficient is produced. This fact can be related to an apparent change of the seal clearance due to modification of the rotor modal line slope.

By installing the rigid bearing in the middle of the shaft span, the stiffness of the rotor is further increased and the rotational speed in which the "whirl" and "whip" phenomena are produced increases.

The bearing radial damping has a minor influence on the oil whirl resonance, which is controlled by the average swirling ratio " $\lambda$ ".

The whirl amplitude increases rapidly with the increasing rotative speed. When the rotative speed approaches to  $1/\lambda \sqrt{(K_1 + K_2)/M_d}$ , two systems natural frequencies coincide and a synergistic effect is observed.

It is therefore essential to perform design tests and configuration of the supports to ensure that both the first "slow transmission shaft" ( $\approx$  30 to 50 rpm), as well as the coupling shaft to the generator, do not present these phenomena of "whirl" and "whip" in their range of speeds rotation.

As for fluid pressure, it does not intervene in the damping of the rotor, according to calculations of the component of the quadrature dynamic stiffness, but if the direct component so the range of the stability margin also increases.

It is therefore essential to perform these tests on each of the tidal turbines, on a real scale, in order to know these parameters of the value of the stability threshold and its margin.

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