

Fluid Instability in Rotating Machines Controlled by Dynamic Stiffness

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Abstract: In this work, a theoretical study of dynamic stiffness and its calculation in rotors that rotate in a fluid environment between them and the stator, such as oil-lubricated bearings, hydraulic pumps, steam or gas turbines, etc., is carried out, as well as its graphical representation with its effect on the stability margin. A practical case is also studied in a laboratory rotor-kit in which, by means of a frequency-controlled disturbance, this rotor is subjected to different rotational speeds to observe the phenomena of oil whirl and oil whip, as well as the stability margin in the approach and during these phenomena. These controls of the vibration obtained and of the values of the stability margin and the components of the dynamic stiffness will announce the induction of fluid instability in the vibration response of the machine.

Keywords: Rotor dynamics, vibration, dynamic stiffness, whirl and whip.

1. Introduction

This work focuses on the study of a fluid that surrounds a rotor or shaft rotating in the gap with the stator, which can happen in liquids or gases of a rotating machine such as fluid bearings, impulsion pumps, seal seals in turbines or other machines.

The fluid, in that hole, has a velocity of zero on the surface of the stator and the same velocity as the shaft or rotor on the surface of the stator, having an average velocity in that gap, which can be cited as $\lambda\Omega$ where λ is the coefficient that of that average velocity and Ω the rotational velocity of the rotor or shaft, λ is always somewhat less than 0,5.[1]

A study is made on a laboratory rotor, according to Figure 1, to which a radial load is caused, by means of a spring to move the center of the shaft inside the seal, so that the fluid acts according to tangential forces and spring according to Figure 2 and Figure 3.

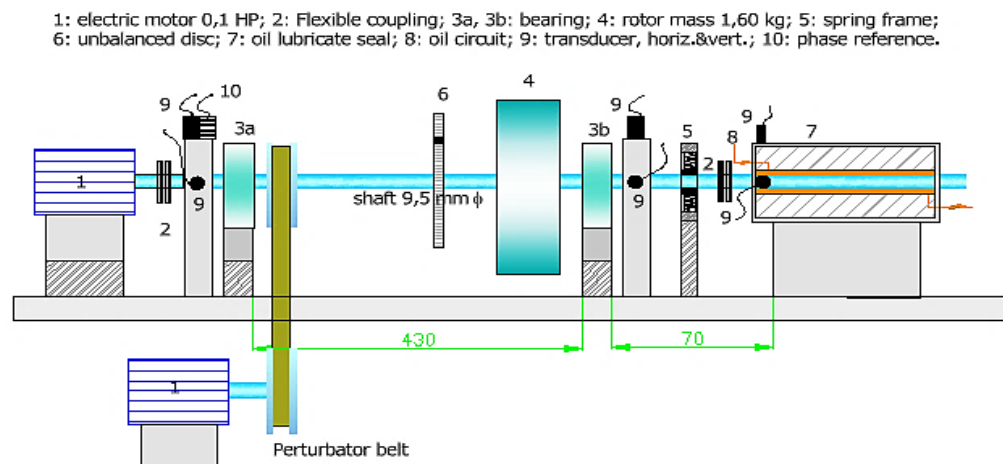


Figure 1. Rotor kit.

The tangential force induces the shaft to position itself in a tangential direction, but the spring effect of the oil wedge drives the shaft to its original position, and it is the complex combination of these forces that definitively positions that shaft and causes its stability. Instability is caused by the relationship between the spring effect produced by the stiffness of the mass of the oil wedge fluid, the damping of that fluid, and tangential forces[2].

The dynamic stiffness of a rotor is given by:

$$\vec{K}_{DS} = \vec{K}_D + \vec{K}_Q = (K - M\omega^2) + jD\omega - j\lambda\Omega D \quad (1)$$

Where: K = spring stiffness, $(-M\Omega^2)$ = mass stiffness, $(+jD\omega)$ = damping stiffness and $(-jD\lambda\Omega)$ = tangential stiffness. M = rotor mass and D = seal damping, ω = frequency of precession, λ = fluid circumferential average velocity and Ω = rotation speed. K_D and K_Q are the direct and quadrature component of dynamic stiffness, which is the denominator of equation (2) where $\vec{F}$ is the perturbing force of the system and $\vec{R}$ the vibration response to the disturbance (amplitude and phase):

$$\vec{R} = \frac{\vec{F}}{[(K - M\omega^2) + jD\omega - j\lambda\Omega D]} \quad (2)$$

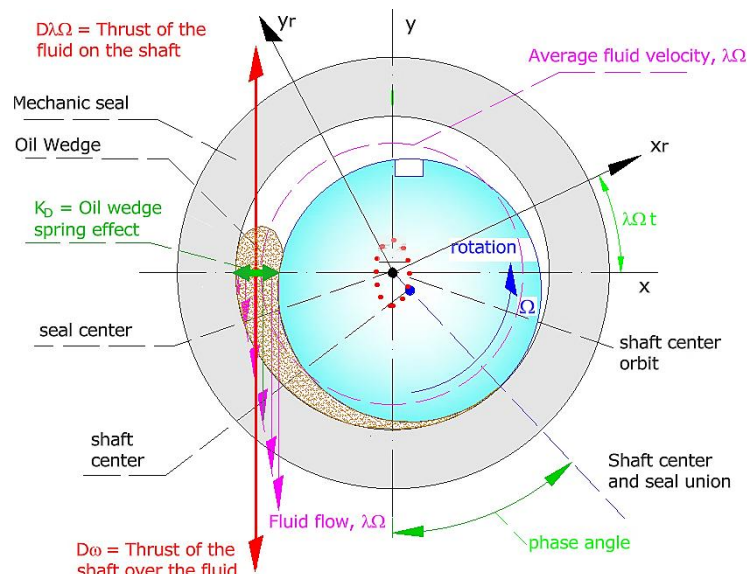


Figure 2. Oil wedge spring effect.

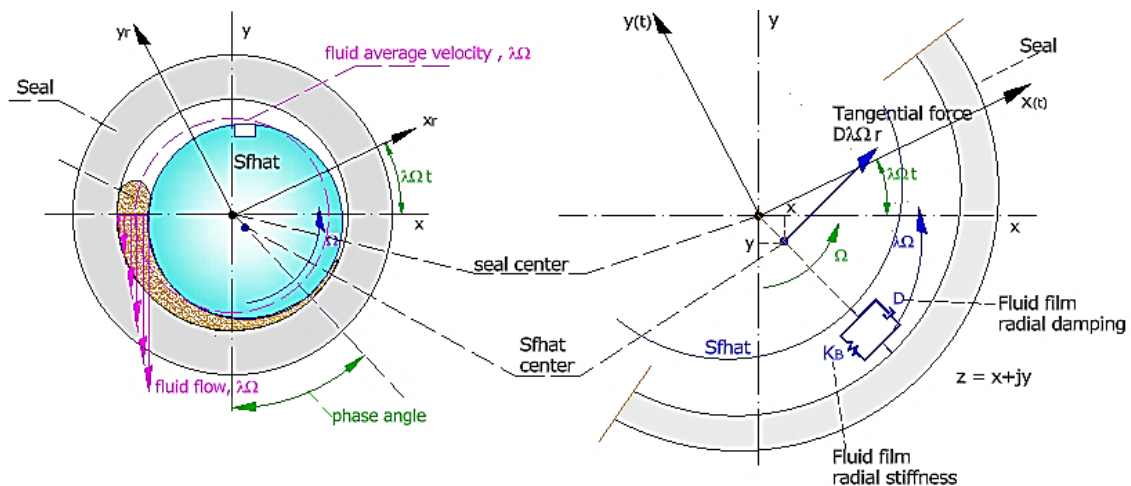


Figure 3. Oil wedge spring effect.

2. Measure of Synchronous Dynamic Stiffness.

There are five steps involved in determining Dynamic Stiffness, [3]:

- 1) Measure reference data of startup or shutdown.
- 2) Add a known unbalance mass in a known location and get data.
- 3) Measure response due to calibration weight alone.
- 4) Measure applied force from this unbalance mass.
- 5) Measure Synchronous Dynamic Stiffness.

I. Obtain data from startup and shutdown.

Select at point, at 1000 rpm that corresponds to 2,8 mils $\angle 65^\circ$ (R_1), according to Figure 3, and convert from polar to rectangular is: $-2,70 + j 0,72$ mil pp.

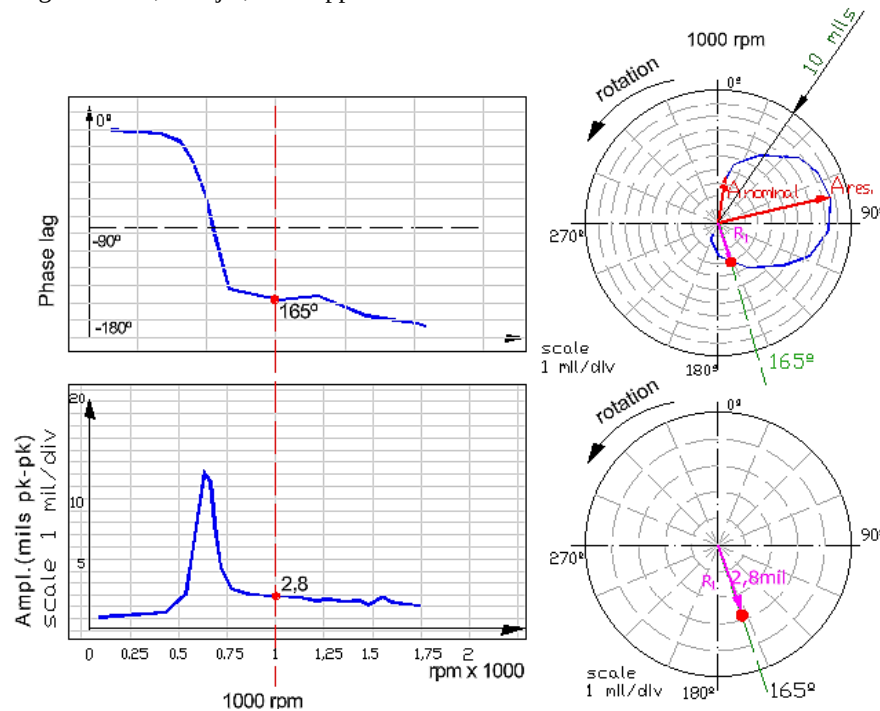


Figure 4. Data from startup. Bode and polar plot.

II. Attach a known unbalance weight in a known location and get new data.

Added 0,13 gr. in a position of 90° and at a radial distance of 35 mm, you get the response at the same previous speed of 1000 rpm, which indicates a vibration amplitude of 3,5 mils in the 201° direction, according to polar diagram a) of Fig. 1.

This new response is the sum of the original reference response of the rotor, in its original initial undisturbed state (R_1) plus the given imbalance weight (R_2) that has been added.

The response is now $R_1 + R_2 = 3.5 \text{ mils} \angle 201^\circ$ (Fig. 1, polar diagram b)), that in rectangular form it is: $-3,27 - j1,25$ (Table 1). The response that is interested to obtain is that which is due exclusively to the weight of the imbalance added R_3 , and is carried out according to the graphic representation in polar of the Fig. 2:

	polar	rectangular
$R_1 + R_2 =$	$3,5 \angle 201^\circ$	$-3,27 - j 1,25 \text{ mil pp}$
$- R_1 =$	$-(2,8 \angle 165^\circ)$	$- (-2,70 + j 0,72 \text{ mil pp})$
$R_3 =$	$2,61 \angle 269^\circ$	$-0,57 - j 1,97 \text{ mils pp}$

Table 1. Polar and rectangular coordinates

$R_3 = 2,05 \text{ mil pp} \angle 253,86^\circ$ is the change in the rotor system response due to the weight that was added. In the Fig. 2, R_3 is translating to the origin, on a polar plot, to easily see amplitude and phase.

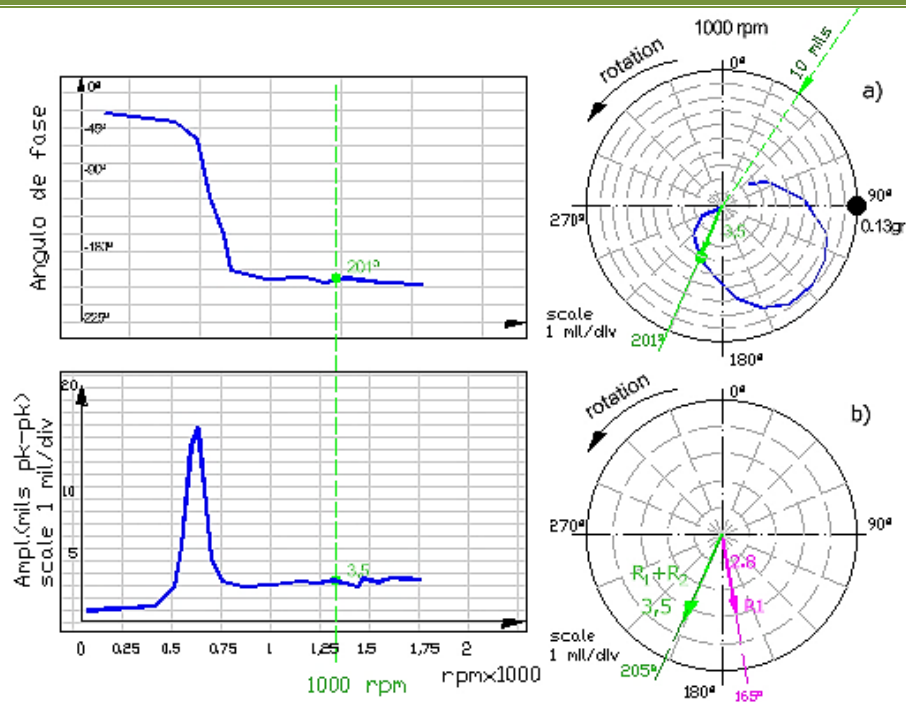


Fig. 1 Data obtained with the known unbalanced weight.

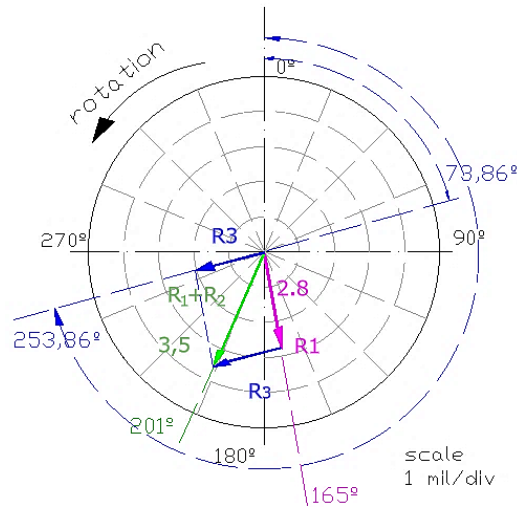


Fig. 2. Response vector due to calibration only.

III. Calculate applied force from the calibration weight.

$$\begin{aligned} \vec{F} &= m r_u \Omega^2 e^{i\delta} = (0,13 * 10^{-3} kg)(35 * 10^{-3} m) \\ &\left[1000 \text{ rpm} * \left(\frac{2\pi \text{ rad}}{1 \text{ rev}} \right) \left(\frac{1 \text{ min}}{60 \text{ sec}} \right) \right]^2 \angle 90^\circ \\ &= 49,84 * 10^{-3} N \angle 90^\circ \end{aligned} \quad (3)$$

IV. Calculate Synchronous Dynamic Stiffness.

The Dynamic Stiffness is a zero-to-peak (N/m), first convert the vibration amplitude of the response vector ($\vec{R}_3$) from peak-to-peak to zero-to-peak (i.e. divide by 2):

$$\vec{R}_3 = 2,05 / 2 \angle 253,86^\circ = 1,025 \text{ mils} \angle 253,86^\circ = 0,03 * 10^{-3} \text{ m} \angle 253,86^\circ$$

This response R_3 , represented in a polar diagram as in Fig. 2, easily indicates its amplitude and phase. It is transferred to the vector field and matches the result of $(\vec{R}_1 + \vec{R}_2) - \vec{R}_1 = \vec{R}_3$.

$\vec{R}_3$. can be expressed as $\vec{R}_3 = Ae^{j\alpha}$, where “A” is the amplitude of the response and “ α ” the phase angle. The disturbing force of the controlled imbalance is: $\vec{F} = Fe^{j\delta}$:

$$\begin{aligned}\vec{K}_{DS} = \frac{\vec{F}}{\vec{R}} &= \frac{Fe^{j\delta}}{Ae^{j\alpha}} = \frac{F}{A}e^{j(\delta-\alpha)} = \frac{F}{A}\angle(\delta-\alpha) = \frac{F}{R_3}\angle(\delta-\alpha)^\circ = \\ &= \frac{49,84 * 10^{-3} \text{ N}}{0,03 * 10^{-3} \text{ m}}\angle(253,86 - 90) = 1661,33 \text{ N/m} \angle 163,86^\circ\end{aligned}\quad (4)$$

▪ Direct Dynamic Stiffness:

$$K_D = K_{DS} * \cos\theta_k = 1661,33 \text{ (N/m)} * \cos 163,86^\circ = -1595,84 \text{ N/m}$$

▪ Quadrature Dynamic Stiffness:

$$K_Q = K_{DS} * \sin\theta_k = 1661,33 \text{ (N/m)} * \sin 163,86^\circ = 461,82 \text{ N/m}$$

This previous test is then carried out with the same determined weight of unbalance 0,13 gr. $\angle 90^\circ$, but at different speeds of rotation, and the response data is taken to obtain these components of the dynamic stiffness as a function of speed of rotation, according to Figure 5[4].

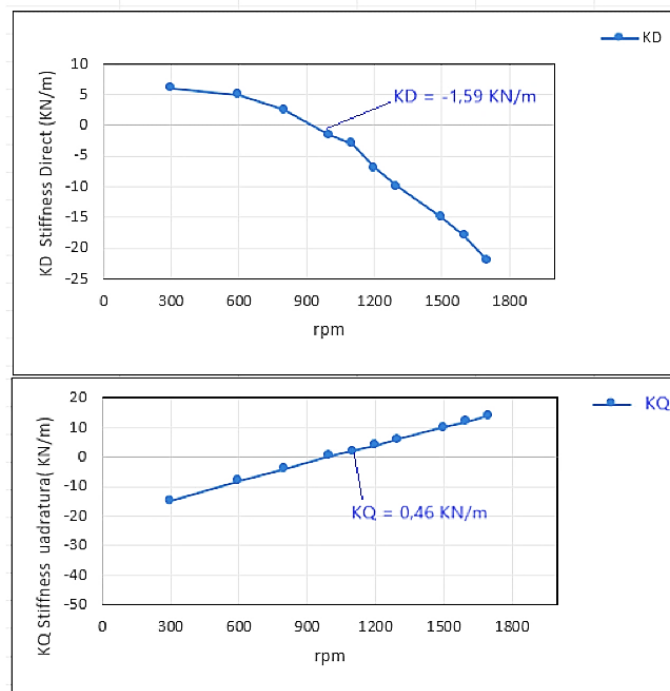


Figure 5. Dynamic stiffness over entire speed range.

3. Case study. Results.

The dynamic stiffness of a rotor, such as the one in Figure 1, is going to be studied when it is in a situation of instability at the rotation speeds that the phenomenon of the whirl or whip appears. The mass disc is located 6 cm from the mechanical seal, and the bearing spacing on the shaft is made 43 cm. The electric motor was 0.1 HP with a variable speed controller and the pressure of pump outlet is 14 kPa. The oil whirl appeared at 2200 rpm and the switch to the oil whip happened at just under 5000 rpm, according to Figure 6[5][6], in which a stability margin of almost 2000 rpm is observed. [1]

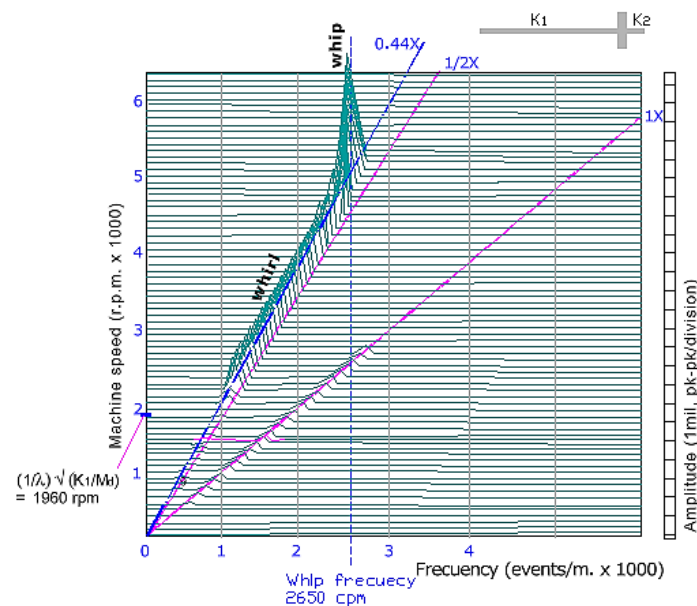


Figure 6. Full spectrum cascade of rotor test.

In a stable rotor, the dynamic stiffness is not zero: $jD\omega$ is unequal to $j\lambda D\Omega$ at the same frequency, and the difference between the zero of both components of the dynamic stiffness is defined as the margin of stability, as shown in the Figure 7, which is $960 - 700 = 260$ cpm, to 1000 rpm in stable operation.

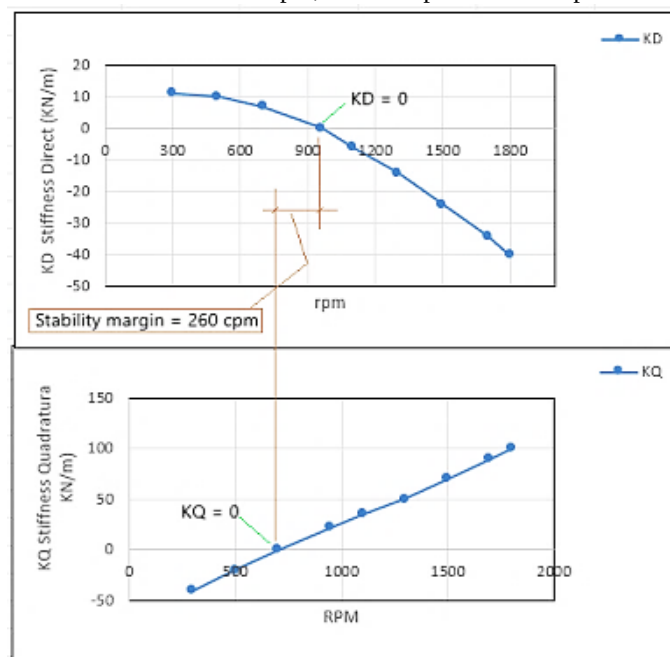


Figure 7. Dynamic Stiffness to 1000 rpm, in stable operation. Stability margin = 260 cpm.

To study the dynamic stiffness of the rotor when operating under the oil whirl and whip, it is necessary to produce a non-synchronous perturbation out. This was done by another 0,1 HP motor and a drive belt close to the first bearing in 1:1 speed ratio as indicated by Figure 1, which caused a constant rotational force of 0,12 kg at a variable rotational frequency. The spring support disc, 5 in the Figure 1, was adjusted so that the shaft was located in the center of the mechanical seal.

The rotor-kit was put to work in the stable zone, at 1000 rpm, and the disturbance force was applied by increasing its transmission speed from 250 to 2500 rpm. The acquired vibration signal is taken by the installed XY induction transducers and sent to a computer that has Bently Nevada's Adre software for windows

installed[7], at intervals of 50 cpm, with the appropriate filters for the disturbance frequency. The dynamic stiffness was calculated for each speed, according to the equation:

$$\vec{K}_{DS} = \frac{\vec{F}}{\vec{R}} \quad (5)$$

where $\vec{F}$ is the disturbance force and $\vec{R}$ is the filtered vibration response of the rotor-kit.[8]

The Figure 8 shows the dynamic stiffness at 1500 rpm, with the two direct and quadrature components, which are zero at 1110 rpm and 950 rpm respectively. The margin of stability has decreased to the value of $1110 - 950 = 200$ cpm, as it approaches the appearance of the whirl.

When the direct and quadrature stiffness are zeroed at the same frequency, the margin of stability becomes zero, because the denominator of the equation (2) is also zeroed and there is no restriction on the amplitude of the vibration of the response. The rotor enters the operation of the fluid instability and at that point the rotor begins the operation of the whirling, and the natural frequency of the system is modified.

The Figure 9 shows dynamic stiffness at 1900 rpm, with the two direct and quadrature components, which are zero at 1250 rpm and 1160 rpm respectively, so the margin of stability has decreased to the value of $1250 - 1160 = 90$ rpm. This means that when the rotor is running at 1900 rpm, there is therefore a stability margin decreased to 90 cpm and the machine still not subject to fluid-induced instability. The margin of stability is reduced, and the rotor approaches the limit of its fluid-induced stability.

The Figure 10 shows how the direct component of dynamic stiffness becomes zero at the same frequency as the quadrature component, so the margin of stability becomes zero when the rotor operates at 2200 rpm, since the oil whirl starts at 2200 rpm (Figure 6). The fluid, in this case, already operating in the oil whirl operand, is still in good condition and the gap between seal and shaft continues to be lubricated throughout its space.

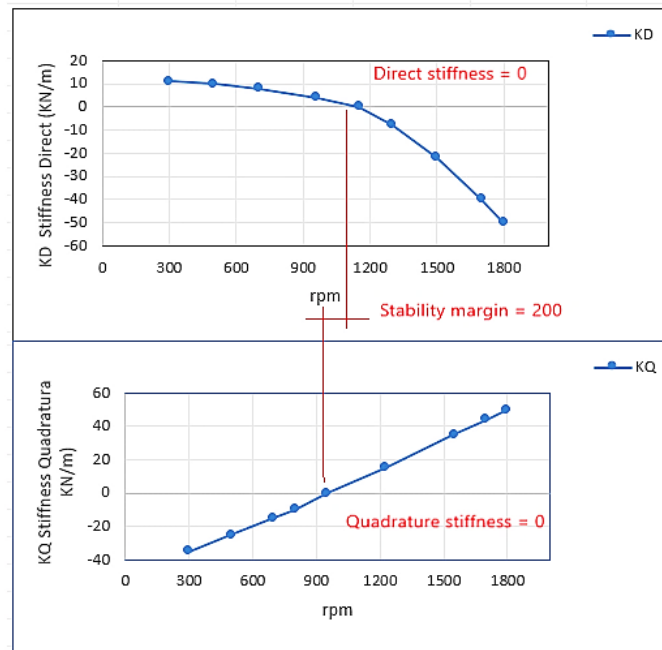


Figure 8. Dynamic stiffness to 1500 rpm. The stability margin decreases to 200 cpm.

The Figure 11 corresponds to a rotational speed of 5000 rpm at which the oil whip already appears. The temperature of the lubrication oil was increased from 27°C, which was at the beginning of the test, to 44°C, which indicates that the fluid film is being crushed and cut during this oil whip phenomenon, according to a physical effect like that in the Figure 12[9], and the seal stiffness is very high. In this Figure 11 it can also be seen that two values of zero are produced, for direct and quadrature components of dynamic stiffness, at the whip frequency which is 2650 cpm (Figure 6). Direct and quadrature stiffness are zero at the same frequency and the rotor behaviour show proves of another vibration mode [10].

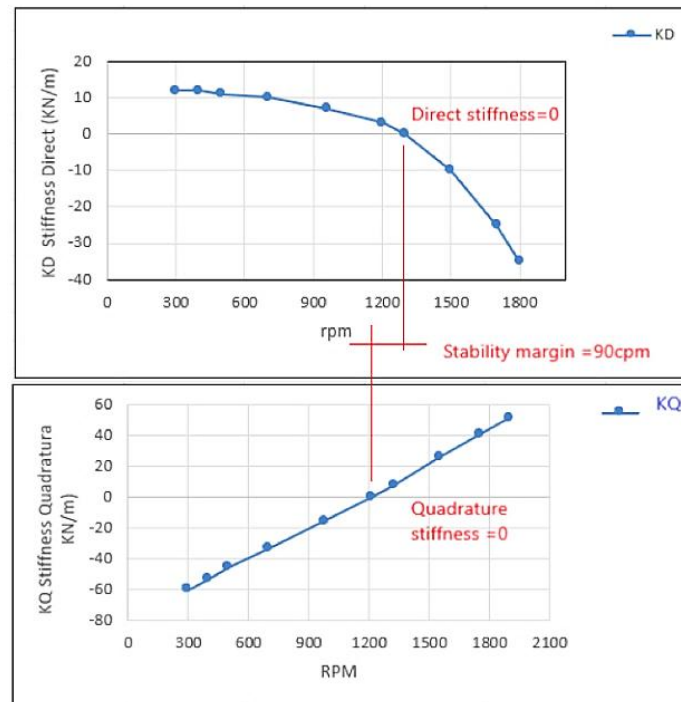


Figure 9. Dynamics stiffness with rotor at 1900 rpm. The stability margin decreases to 90 cpm.

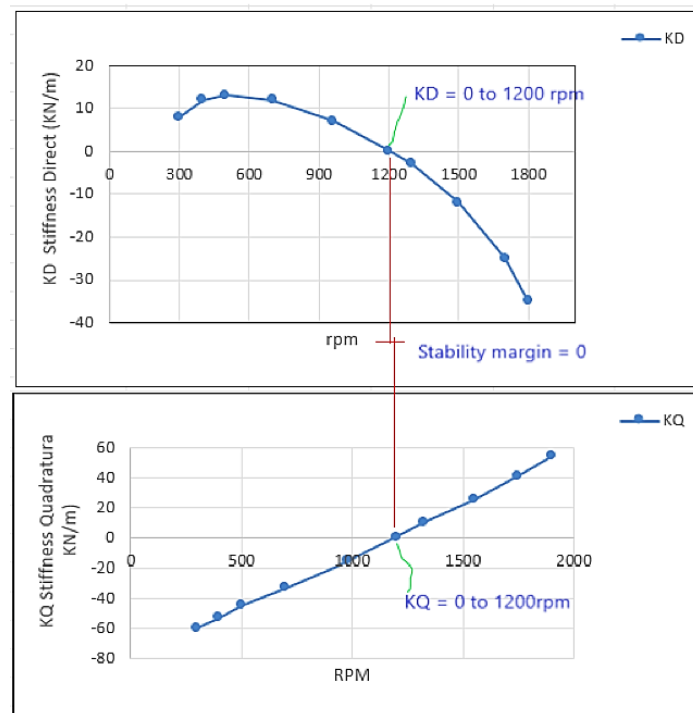


Figure 10. Dynamic stiffness with rotor operating at 2200 rpm, in oil whirl. Direct and quadrature stiffness are zero at the same frequency.

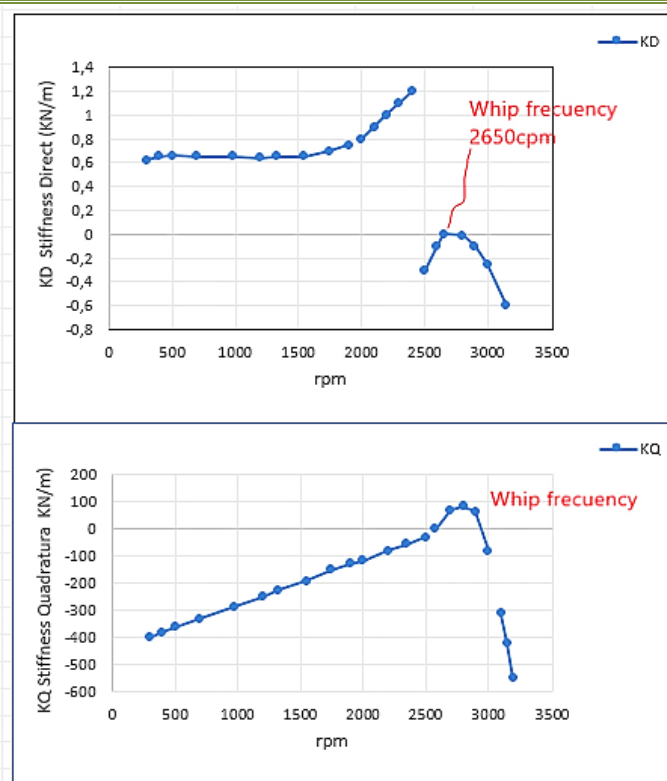


Figure 11. Dynamic stiffness at 5000 rpm, in oil whip. Oil whip frequency =2650 cpm.

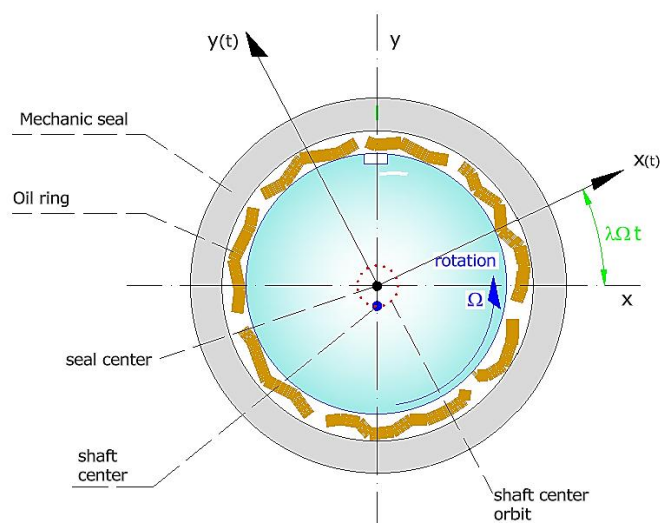


Figure 12. Oil whip.

4. Conclusions

It is observed in these tests that the Margin of Stability decreases as the rotor speed approaches the Threshold of Stability, which is where the whirl and whip phenomenon begins at its natural frequency. These results indicate that in the whirl and whip the rotor works at the stability threshold, the stability margin is zero, and the direct and quadrature dynamic stiffness becomes zero at the whirl or whip frequency respectively. For whip data, the graphical results of dynamic stiffness cannot be easily obtained, the rotor vibration would increase in amplitude to a great extent, so it was plotted based on the behaviour. Stable rotor operation will always occur at operating speeds in the range defined by the stability range.

The components of the dynamic stiffness are zeroed at the same frequency as the frequency of the whip and the amplitude of the vibration grows significantly.

Stable rotor operation will always occur at operating speeds in the range defined by the stability range. Therefore, it is essential to know these parameters of a system, to achieve rotational stability in the face of possible instabilities of a shaft with surrounding fluid, such as oil lubrication bearings, hydraulic pumps, mechanical seals, turbines, etc.

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