Stability and Vibrations of One RCP's Lineshaft

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1 INTRODUCTION

To assess the mechanical behaviour of primary Reactor Coolant Pump set in operating conditions, there is a method wich consists in watching over vibrations, or better, the evolution of vibrations. Means and methods of calculation, instrumentation for the surveillance of machines in real time and the processesses of interpretation of measurement have improved nearly simultaneously. It is possible now to predetermine by calculation the vibratory behaviour of a lineshaft by getting some parameters specific to operation conditions varied.
The object of this document is to present the methods to determine:

- wether the lineshaft vibrations are stable,
- the vibrations while taking into account at any moment the characteristics of the bearings in transient conditions.

The results are compared to measurements.

2 EVOLUTION OF CALCULATION METHODS

During the design phase of a rotating machine, one always tries to avoid the rotating rated speed to be near a particular lineshaft frequency.
Assumptions are then made on the unbalances so as to study the lineshaft vibratory regime.
Two periods can be considered in the evolution of methods for these two calculations :
In the first period the particular vibration frequencies have been calculated without taking dampings in the bearings into account.
The lineshaft vibrations are calculated on steady state. It is assumed that the shaft is vibrating at the frequency corresponding to the rotating speed, and that the trajectories of the different points of the lineshaft are stable.
In the second period, the dampings in the bearings are taken into account to calculate the natural frequencies. Complex mode shapes are determined at different rotating speeds. Such a mode shape is then characterized by a frequency and a damping. The lineshaft vibrations will be stable or not depending on this damping value.
The vibrations can be calculated in transient operating conditions, that is to say the reactions in the bearings are calculated by considering the real position of the shaft in the bearings.

3 NATURAL FREQUENCIES AND VIBRATIONS IN LINEAR OPERATING CONDITIONS

3.1 Methods of calculation

There are different methods to determine the natural frequencies of a lineshaft. It is possible for instance to use the finite element method. We have here chosen to use the transfer matrix method, where Oz is the shaft centerline, Ox is a vibration direction perpendicular to the shaft-centerline. The lineshaft is a succession of «slices».

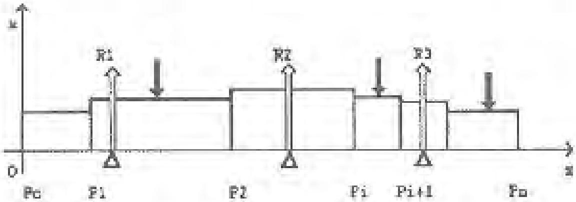


Fig. 1

Each slice has characteristics independent of z : external diameter and internal diameter, external load by length unit, external inertia, etc . In the case of a simple vibrating shaft in the xOz plane the calculation principle is made clearer hereafter :

If V is vector (x, dx/dz, M, T) where M and T are the moments of flexion and transverse forces, x the displacement at abscisse point z. One prooves that $V_i = A_i V_i$ where V_i is the vector V on point P, beginning of the «slice» number i, $V_i + 1$ the vector V at the end of this slice and A_i a matrix that only depends on the slice characteristics and on the vibration frequency f. At the beginning of the first slice, two components are null : M and T; two components are unknown : xo and (dxo/dz).

The calculation allows to express the successive vectors V_i in function of these two unknowns. Especially at the end of the lineshaft of figure 1, one expresses that M_n and T_n are null :

$$\begin{bmatrix} M_n \\ T_n \end{bmatrix} = \begin{bmatrix} a & b \\ c & d \end{bmatrix} \begin{bmatrix} x_o \\ dx_o/dz \end{bmatrix} = \begin{bmatrix} o \\ o \end{bmatrix}$$

There is resonance when the determinant of matrix 2 x 2 is null.
Then it is theoretically possible to find out an infinity number of natural frequencies.

The method can be adapted when considering vibration in 2 axis, gyro-effects, elasticity and bearing dampings coefficients.

The complex natural value which annuls the determinant has then the form $(\omega + \delta)$ where $\omega = 2\pi f$ and δ is the damping coefficient.
Each component of vector V has the following form :

$$f(z) = \text{Real}\ \underline{f(z)}\ e^{j(\omega + j\delta)t}.$$

It is also feasible to determine the vibrations due to the unbalances resulting from a given rotating speed by using the transfer matrixes.
As well as for determining the complex natural modes, the bearings coefficients are supposed to be steady. It is only true when the displacement in the bearings are small. This assumption is no longer acceptable to calculate vibrations if one of the natural frequencies indicates that the lineshaft is unstable.

3.2 Results of calculation.

These calculations are performed for the vertical motor pump set in figure 2. It is characterized by three bearings :

. a bearing with tilting pads in the upperside, between the flywheel and the rotor of the motor,
. a bearing with tilting pads under the motor,
. a bearing located in water e. g. in the pump.

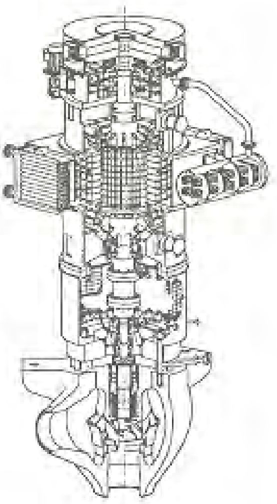


Fig. 2

3.2.1 Stability

The pumps equipped with a journal bearing may be unstable. The instability has a frequency value near the half of a rotating speed. When the bearing with tilting pads is preloaded there is no more instability.

	Without preloading		With preloading	
N°	Speed	Dampings	Speed	Dampings
1	672	13.1	992	2.99
2	729	- 3	1663	31.3
3	905	24.9	1934	18.2
4	1203	14.2		

The trajectories of neutral fiber points have an elliptical shape. These computations indicate that it would be difficult to reduce vibratory level of the set when the preload of the motor bearing is not sufficient : the instable vibration frequency having a value roughly equal to the half of the rotating speed and therefore making the balancing impossible.

Figure 3 shows the 4 natural modes corresponding to a lower motor bearing without preload. The vibration of the impeller (indicated in the right part of the diagrams) is very high for each natural mode. The replacement of the journal bearing of the pump by an hydrostatic bearing located on the impeller insures the stabilization of the lineshaft and reduces the vibration level of the impeller.

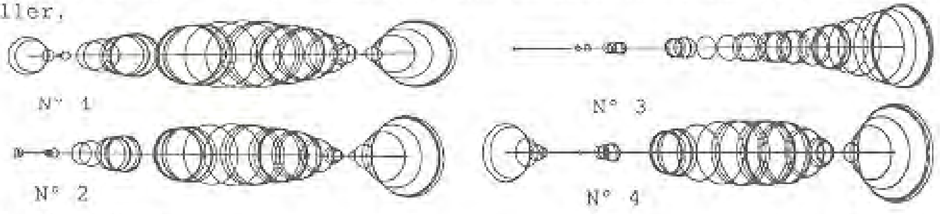


Fig. 3

3.2.2 Vibrations

The vibrations in steady operating conditions can only be calculated if lineshaft is stable. Assumptions must be made on the position and values of the unbalances, on the bearing loads as well as on the centering defects due to the set configuration. The vibration curve at the rotating speed is shown in Figure 4, when the lower bearing of the motor is preloaded.

Fig. 4

The vertical position of the shaft line and the low static loading seem to be the origin of non linear dynamical behaviour. These phenomenons no longer allow use of stiffness and damping coefficients classically calculated by linearizing fluid film equations near a stable static equilibrium. They require specific modelization of fluid film.

4 NUMERICAL SIMULATION IN NON LINEAR CONDITIONS

4.1 Non linear method presentation

A specific non linear model has been integrated in the EDF rotordynamic simulation code CADYRO. CADYRO is a finite-element code allowing representation of main components of the machine (shaftline, casing, bearings). A modal synthesis method is used in order to reduce the computational effort due to the detailed representation of the machine.

For non-linear modelization of bearing fluid films the REYNOLDS equation is solved with the shaftline and casing mechanical equations. For that purpose CADYRO uses an explicit EULER transient integration scheme. At each time step the film fluid pressure is integrated, depending on the rotor situation, speed and rotating speed. The bearing fluid films models are extracted from specific bearing calculation tools considering laminar flow and taking into account fluid viscosity, fluid temperature, pads inertia and technological parameters as preloading or oil feeding grooves.

4.2 Analysis of the motor pump rotordynamic

This pump is rigidly coupled to an electrical motor (fig. 2). The shaftline is guided by two oil supplied tilting pads bearings for the motor and by an hydrodynamic circular bearing for the pump. The monitoring instrumentation of shaftline is situated at coupling location.

The unstable behaviour, anticipated by the linear simulations, leads to some troubles during balancing operations : difficulty in reducing global vibrations under a certain level, presence of a subsynchronous component (half speed frequency) and a tendency to increase this component after balancing.

A shaftline finite element model has been developped. The various static and dynamic loadings (mechanical and hydraulic unbalances), roughly estimated, have been simulated. The non linear method has been used to represent the non harmonic behaviour of the shaftline.

4.2.1 Subsynchronous component origin

In order to estimate the efficiency of the bearings on the shaftline behaviour, we apply an initial short time duration loading to it. Then we can observe the stabilized trajectories. Figure 5 summarizes the obtained results. We can notice that pump rotor trajectory does not converge to the center of the bearing : the motion is kept on its initial amplitude by an unstable whirl. A frequency analysis shows us a single predominent subsynchronous component at 12.5 hertz near half the rotation speed. This phenomenon is quite typical of an unstable behaviour of the shaftline in its bearing.

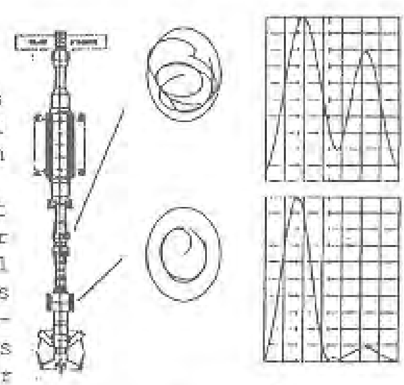


Fig 5 : Stability test

A second simulation, taking into account the hydraulic loadings, represents the normal working condition of the pump. The frequency analysis still shows the unstable subsynchronous component of the shaftline trajectory in the pump bearing at 12.5 hertz, but here the presence of hydraulic balancing imposes a synchronous component at 25 hertz (figure 6). At the coupling, location of the monitoring instrumentation, no unstable component, can be noticed. This result shows that unstable behaviour of the rotor in the pump bearing can be hidden from the operating monitoring.

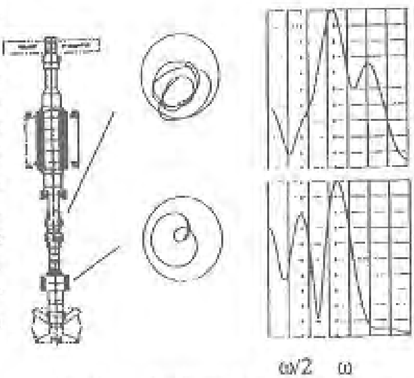


Fig 6 : Hydraulic loadings

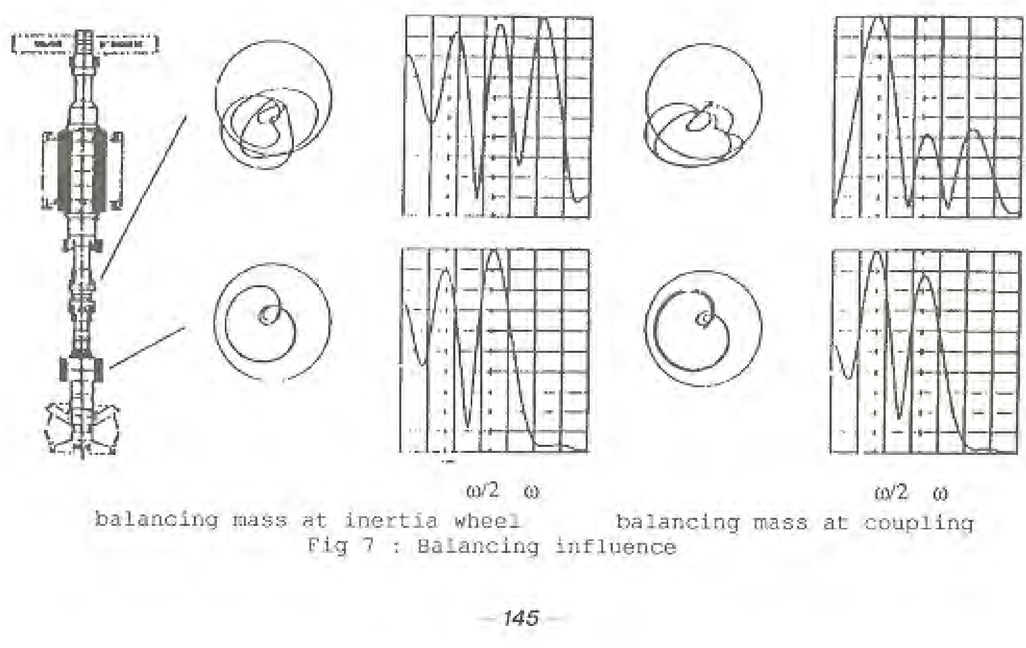
At this stage of the analysis we have been able to compare the coupling amplitude of observed trajectories (50 μm < A < 150 μm) and the simulated ones (A = 90 μm).

These results show us the non linear model ability to represent the unstable phenomenons. More simulations on different parameters variation indicate the unvarying unstable pump bearing behaviour (out of alignment bearing, motor bearings preloading, balancing or static loadings). We also observe the wrong correlation between coupling and pump bearing behaviour.

4.2.2 Balancing influence on coupling behaviour

So as to simulate balancing operations we apply several balancing masses configurations to the balancing levels (inertia wheel and coupling). The results give an accurate representation of observed phenomenons.

As observed on real machines at coupling location, balancing masses can reduce first harmonic component but rises subharmonic one. Coupling motion is the resulting combinations of the two phenomenons, giving rise to a first harmonic or a half harmonic component but keeping a constant global amplitude.



balancing mass at inertia wheel   balancing mass at coupling

Fig 7 : Balancing influence

Figure 7 summarizes obtained results with balancing masses located at the coupling and at the inertia wheel. The global amplitude at coupling is the same than in normal conditions (figure 6) but its frequential components are different. At the same time it can be seen that balancing has no effect, neither on vibration level nor on unstable behaviour of the rotor in pump bearing. This is an interesting information given by the simulations, because there is no monitoring instrumentation in this place.

We don't have to worry about these phenomenons for the rotor safety, finite amplitude orbites are obtained due to the non linear behaviour of fluid films in presence of dynamical loading. On the other hand it makes balancing and monitoring operations very difficult.

4.2.3 Change of the pump bearing

The model permits us to test some modifications in order to make the pump working in better conditions. The most effective is the change of the hydrodynamic circular pump bearing to an hydrodynamic three lobes bearing. Figure 8 shows the obtained results. For a stability test (short time duration loading), the rotor orbite in the pump bearing becomes quickly stable and its component is only an harmonic one. Then we apply to the rotor the hydraulic loadings, orbites amplitude levels are equivalent to those obtained with the circular bearing.

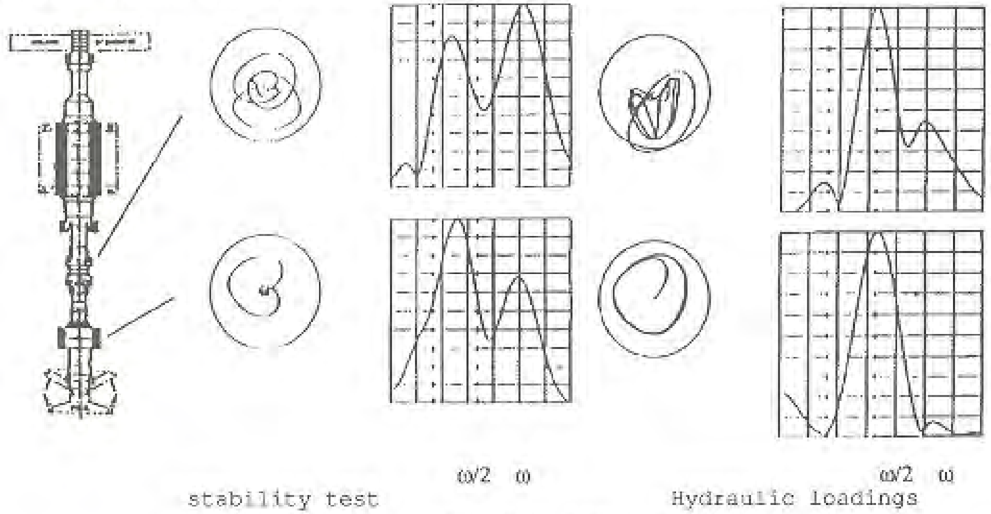


Fig 8 : Change of the pump bearing

The model enabled us to estimate the progress such a modification would bring for balancing and monitoring operations.

5 CONCLUSION

Rather good agreement between observed and simulated behaviours shows us the importance of taking into account the non linear effects due to bearing fluid films for better understanding of rotor dynamics of vertical machines. More over, in good accordance with the experiment, the calculations show that, if the lineshaft is not stable, it is difficult if not impossible, to reduce the vibratory level.