Rotor vibration reduction using rotor balancing and command shaping

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Abstract. Mechanical vibrations are intrinsically characteristic of rotating machines. Apart from that, other turbomachine characteristics, such as rotor unbalance, non-linearity and shaft variable stiffness can introduce severe disturbances in the operation of the machine. In this paper two complementary methods to reduce rotor vibration, rotor balancing and command shaping, are applied to a flexible rotor machine. The results will show that the combination of the two methods produces a clear reduction of the vibration amplitude.

Keywords

Turbomachines, mechanical vibrations, command shaping, rotor balancing.

1. Introduction

When driven by an electric motor, the vibration pattern of a rotor is affected by both mechanical and electric factors. Mechanical factors include the shaft construction, the location of the supports, and the distribution of masses such us impellers, spacers and couplings; electrical factors include the electric power wave, namely concerning waveform and frequency (or frequencies), which relates directly to the speed drive type used and/or the electrical network power quality parameters.

The stiffness and damping characteristics of the mentioned whole rotor structure and its supports will play an important role in the behaviour of a flexible rotor. In addition, the relation between the operating speed range and the rotor natural frequencies is critical due to resonance phenomena. Given the importance of rotor flexibility, it is not surprising that an enormous amount of research and development in both fields mechanical and electrical has gone into dealing with improving flexible rotor dynamics. Mechanical solutions include: stiffening the mechanical structure of the own rotor, adding damping to supports, reviewing the distribution of stages and supports, rotor field balancing, and finally using specially shaped reference commands for the velocity profile to reach the desired operating speed [1-4]. Electrical solutions include star-delta starters, autotransformers, rotoric resistors, a variety of speed drives, and even programmable waveform power sources [5,6].

This work presents a case study implementation of the rotor balancing and command shaping techniques in order to reduce rotor vibration. It discusses methods for designing and selecting a motion command generator for rotating machinery without command-induced unwanted dynamics, while focusing on the implementation of these two techniques simultaneously.

2. Laboratory equipment and tests

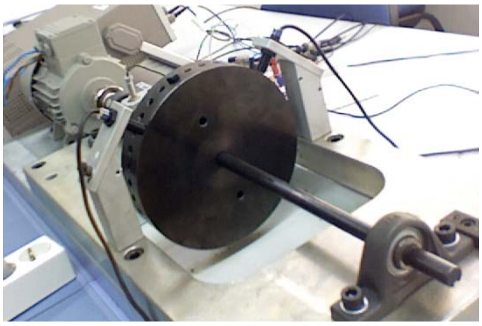


Fig. 1. A view of the OSIFRO model

In order to study the influence of the electric drive on the overall behaviour of a mechanical load, a one-stage inertia flexible rotor (OSIFRO) set was specially designed, dimensioned and built. This provided a simple mechanical load with a well-fitted (and well-known) dynamic model with which to work. Figs. 1 and 2 show a view of the OSIFRO model and a representation of the rotor shaft on the bearings.

To drive the OSIFRO, a 0,37 kW squirrel-cage induction machine was connected using a three-phase input, three-phase (sinusoidal PWM) output, TeleMecanique Altivar 71, 1.5 kW variable speed drive. The OSIFRO mechanical load is constituted by a flexible rotor with a cylindrical inertia. This mechanical load is intrinsically slightly unbalanced, and adequate correction masses can be added to balance the rotor.

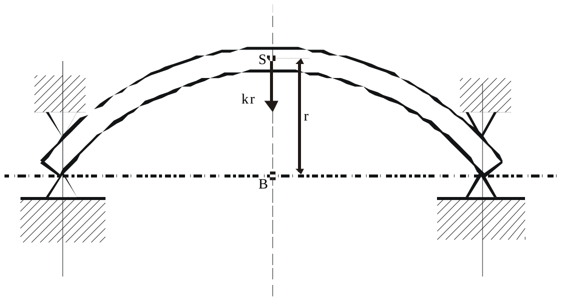


Fig. 2. A representation of the flexible rotor shaft on the bearings

This work required a number of electrical and mechanical quantities to be measured. In order to acquire and accommodate the several quantities to be measured, a PC-based, high-precision data acquisition card was used, together with voltage and current active probes, accelerometers, proximity inductive sensors and a keyphasor.

3. Case study modelling and rotor field balancing

A one-stage inertia flexible rotor can be modelled as a one degree of freedom system, and thus has one vibration mode. This is obviously a simplified model, yet an effective one for this purpose.

This results in an orbit-like motion of the shaft around its central axis, which will be measured using the inductive proximity sensors installed in the OSIFRO, as depicted in Fig. 3.

Although the mechanical modelling of the rotor is beyond the scope of the present paper, it can be stated that the resolution of the motion equations provides as oscillating system, with a natural frequency of around 23 Hz. Also, because the OSIFRO shaft has a slot along its length, the elasticity is variable, causing a subcritical natural frequency to occur.

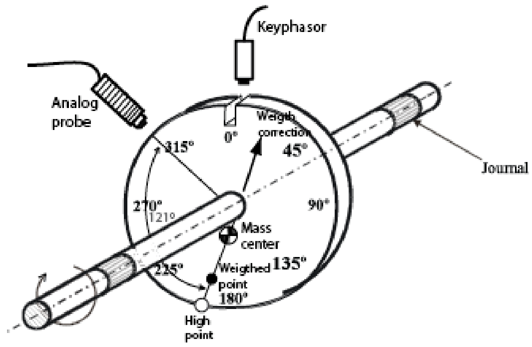


Fig. 3. A schematic of the OSIFRO, showing one of the analog probes measuring the distance to the shaft.

In addition, it is customary that some frequency harmonic of around half of the first critical velocity appear. This is common for horizontal rotors, showing the influence of a non uniform rotor stiffness associated to gravity as the primary cause. This is the case of the OSIFRO considered here. The cross section of the modelled shaft is not completely circular because a slot along the shaft length has been trimmed, and so two principal directions in it can be distinguished, one of maximum and one of minimum stiffness.

The contribution of the slot’s whole surface to the second moment can be estimated as a percentage of the whole shaft cross section second moment, implying nearly the same percentage of variation for the shaft transversal stiffness during the rotation. For the OSIFRO such margin of variation is of around the 15%, while stiffness parameter k varies in the range of [170200, 196380] N/m.

The method to determine the traditional rotor field balancing parameters is also beyond the scope of this paper, and only a brief review will be added; it includes solving two well-known problems:

i) The unbalance phase location.
ii) The estimation of the correction weight.

It should also be noted that a keyphasor signal is essential, and that the process of balancing a rotor is valid for a determined rotation speed. In this case, the rotor was balanced for a mechanical speed of 20 Hz (1200 rpm). The results are shown in Table I and Fig. 4:

TABLE I. – Rotor field balancing results

Calibration weight added	Measured amplitude of vibration (volt)	Estimated Sensitivity vector	Estimated Unbalance (unit: 1 weight)
W=0	8.93	-	-
W=1	7.22	0.29	2.09
W=2	3.87	0.395	3.52
W=3	0.83	0.37	3.30

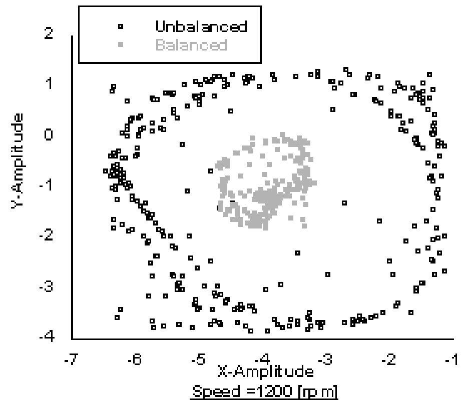


Fig. 4. Unbalanced and balanced rotor orbits: comparison

4. **Command shaping for rotor startup**

Command shaping is a customary method for reducing both transient sway and residual vibration when moving lightly damped systems. It offers several clear advantages over other conventional approach:

i) Designing an input shaping profile does not require a complex analytical model of the system. It can be generated from simple, empirical measurements of the actual physical system: the rotating machine [7].

ii) Input Shaping basically tries not to 'insert' energy into the machine on its natural frequencies [8,9].

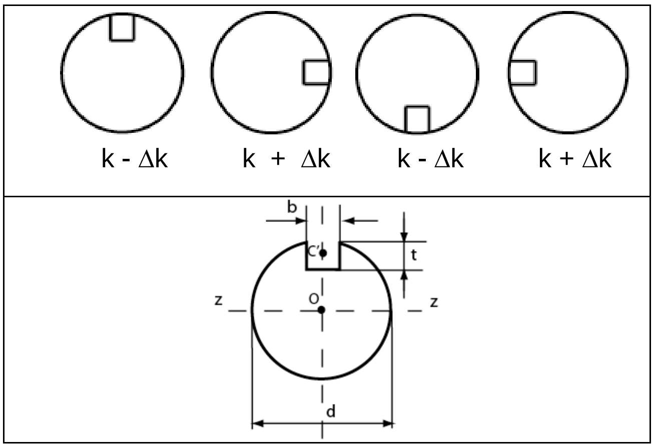


Fig. 3. Variable stiffness of the rotor induced by the slot along its length. Conditions: b=5mm, t=2.65mm, d=15mm. Iz=2485-331.25 10^{-12} m^4

One of the goals of this work was to design an input shaped velocity profile that would drive the rotating machine from a velocity under the subcritical velocity to a nominal velocity over the subcritical, thus mitigating the vibration induced by the variable stiffness of the rotor, and reducing the severity of the vibration for the overall motion.

The developed methodology includes:

i) Analytical and experimental study of the rotor dynamics due to its variable elasticity and it associated natural critical and sub-critical frequencies.

ii) Derivation of the time delay filter for the turbomachine natural frequency, also at half of critical velocity due to the variable elasticity of the rotor.

iii) Shaping of the trapezoidal velocity profile command, by convolving the ramp with the Time Delay Filter, in real time (Fig.4).

iv) Measurement of the experimental results. The measurement plane coincides with the correction plane.

It is expectable the flexible dynamics of the rotating machine shaft traversing the subcritical during the start up might be slightly improved through the use of command shaping. This process filters the input command by convolving the ramp up with a series of impulses known as Time Delay Filter (TDF) [10,11], summarized by the following equation:

$$TDF = A_1 + A_2 \cdot \delta(t - T_1) + A_3 \cdot \delta(t - T_2) \qquad (1)$$

where δ states the Dirac delta function or unit-impulse function, Ai is the impulse amplitude and Ti the instant when the impulse is applied.

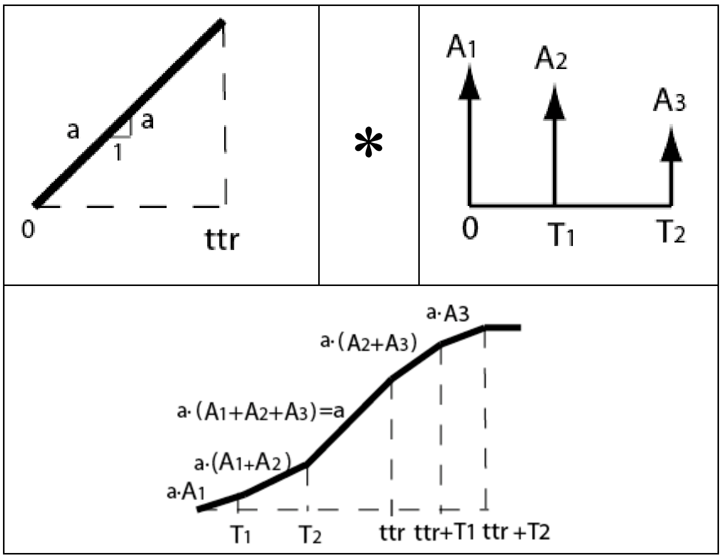


Fig. 4. Command Shaping of a ramp by convolving it with a Time Delay Filter.

A straightforward method to generate a TDF will be introduced here. It involves the minimization of the objective equation corresponding to the residual energy of the flexible system, the shaft on its rigid bearings, after the time of the last applied impulse, measuring time with a digital clock with period Ts, given by:

$$E_{res} = \frac{1}{2}\omega_n^2 \cdot m\left(x(T_n + T_i) - 1\right)^2 + \frac{1}{2} m \cdot \left(\frac{\dot{x}(T_n + T_i) - \dot{x}(T_i)}{T_i}\right)^2 \quad (2)$$

Therefore, the optimization problem to design the time delay filter consists in minimizing equation (2), subject to the constraints of the Jeffcott machine dynamic model.
The Time Delay filter synthesized for the modelled rotor natural frequency 23 Hz is thus obtained:

$$\begin{bmatrix} A_i \\ T_i \end{bmatrix} = \begin{bmatrix} 0.476446 & 0.394896 & 0.128658 \\ 0 & 0.020199 & 0.027285 \end{bmatrix}_{\omega_n = 23\,Hz} \quad (3)$$

where Ai and Ti state the solution of the optimization problem.

The velocity control profile synthesized moves the turbomachine from zero velocity to a velocity close to the nominal. Fig.5.a illustrates the variation of the residual energy at move end as a function of the rotor natural frequency. The blue curve corresponds to the solution obtained by Non Linear Programming (NLP) problem, solved using GAMS, while the gray curve corresponds to the unfiltered case. The solution is robust for a frequency range of around 5 Hz [20, 25], where the residual energy remains below a tolerable level that includes the expected range of variation. Fig.5.b illustrates the simulated response of the machine vs time while comparing the rotor vibration that is mitigated by the TDF shaped input.

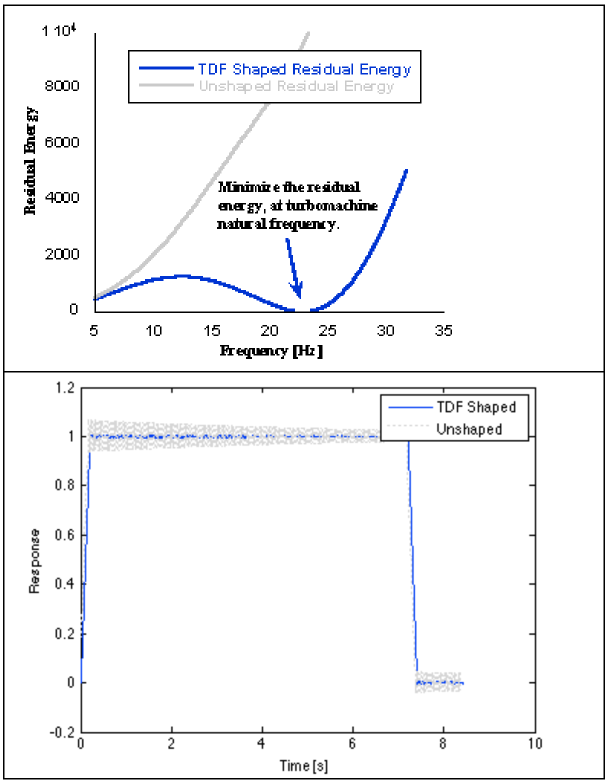
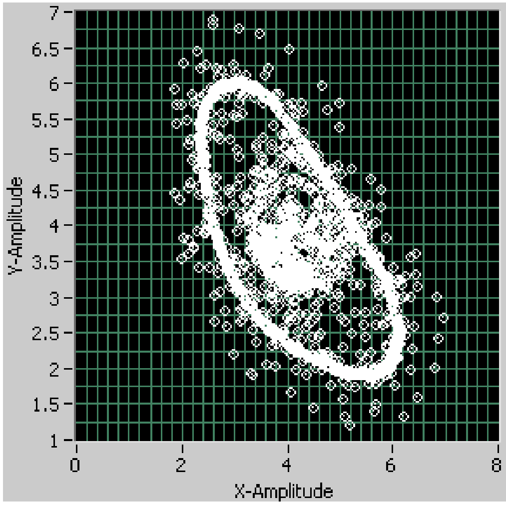


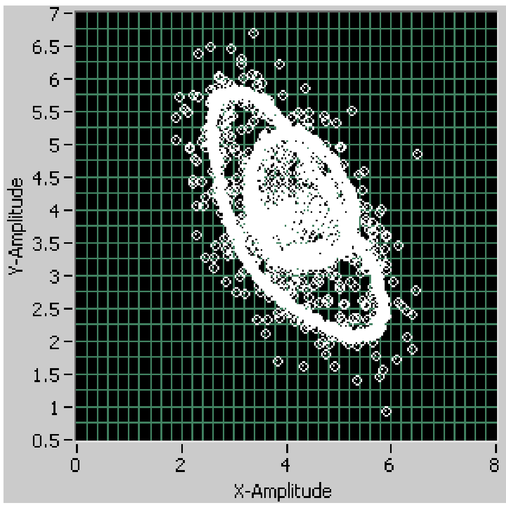
Fig. 5. (a) Minimization of the residual energy at the natural frequency. Conditions wn= 23.5 Hz.
(b) Unshaped and Time Delay Filter shaped response (simulation).

For the sake of exactitude, in each scenario the turbomachine was started seven times. The following table compares the maximum value, the variance and the standard deviation of the experimental results for the unshaped and TDF shaped cases, given that the mean value is unimportant here. The TDF filtered response keeps the values of the statistic parameters fairly below the corresponding values of the unfiltered responses.

Note that command shaping undergoes the worst conditions due to the rotor sub-harmonic resonance, non-linear behaviour, and synchronous forces (mitigated by rotor balancing) acting as disturbances. However, no energy is injected in the turbomachine at is natural frequency by the TDF shaped velocity profile. Thus, the actual rotor dynamic response is fairly improved for this case of the shaped input. The TDF-shaped velocity input performs better because the rotor orbit is less scattered than the unshaped case, as is overall depicted in Fig.6.



(a)



(b)

Fig. 6. Balanced rotor orbits:
(a) Unshaped rotor orbit (b) shaped rotor orbit.

For the shaped case the rotor orbit is less scattered than for the unshaped case, and correspondingly the final velocity is reached in a more stable mode, and less rotor vibration takes place. The final goal is that the rotor uses the highest percentage of the supplied energy in productive work instead of in mechanical vibrations.

Certainly the improvements are modest, but the only price that is necessary to pay for them is a little longer command input due to the delay (of only 0.027 [s]) induced by the filter size, plus the availability of a suitable motor drive that guarantees the reliability of the velocity value reference put into the rotor at each sampling time.

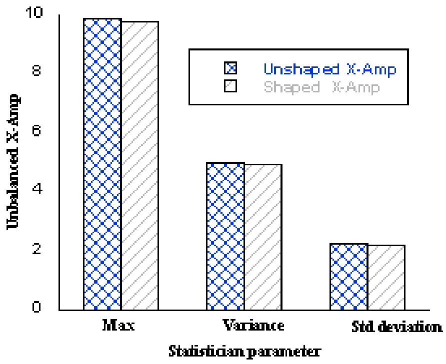
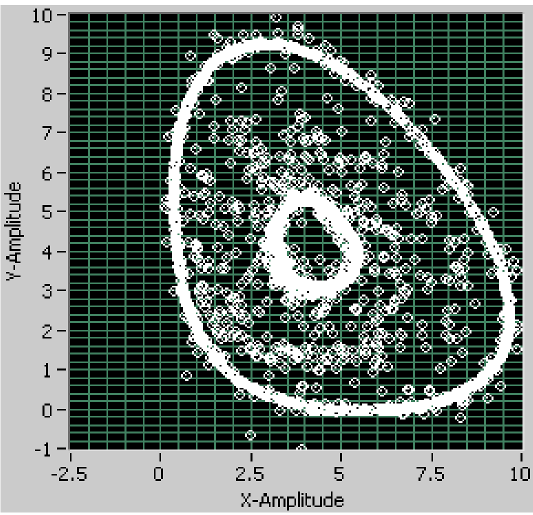


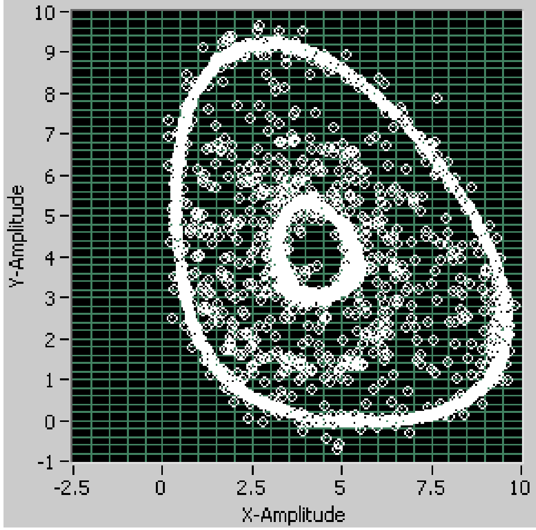
Fig. 7. Statistic parameters for balanced rotor orbits

If the rotor is unbalanced, the disturbances due to the synchronous forces increase. Thus a wide orbit radius is expected when traversing the half critical velocity for the reasons previously stated. Also, the inner orbit corresponding to the final velocity appears on Fig. 8.

Between both orbits the number of points is increased in the shaped case, while in the unshaped the points travel to the boundary of the widest orbit indicating that they are fairly more scattered than the TDF shaped orbits. The statistic parameters shown reafirm this assessment.



(a)



(b)

Fig. 8. Unbalanced rotor orbits:
a) Unshaped rotor orbit b) shaped rotor orbits.

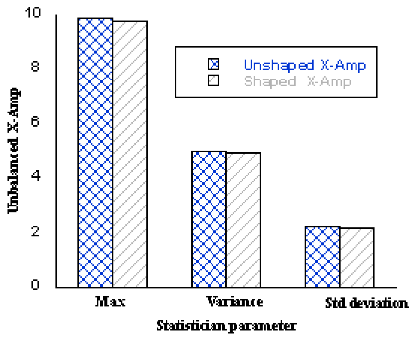


Fig. 9. Statistic parameters for unbalanced rotor orbits

5. Conclusions and outlook

In this work both rotor balancing and command shaping techniques are used to minimize rotating machinery vibration, namely due to disturbances introduced by unbalance and non-linearity due to shaft variable stiffness.

Command shaping techniques avoid inserting energy into the machine at the flexible rotor natural frequency, thus improving dynamic performances. This is achieved at a low cost, as only an accurate speed drive is necessary for both command shaping implementation, and it is compatible with rotor field balancing.

While somewhat limited in scope and results, this work demonstrates the beneficial dynamic effects of command shaping when associated with a suitable electrical speed drive that inserts modulated energy into the machine, improving its performance mainly at start-up. Thus, command shaping contributes fairly to the final goal, which consists in the rotor spending the highest percentage of the supplied energy in productive work rather than mechanical vibrations.

Steady-state vibrations (at a determined speed) are better addressed using rotor balancing techniques, thus adding to the advantages of the concomitant use if these two techniques.

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