On the Experimental Identification of Dynamical Characteristics of Annular Gas Seals

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Abstract: Annular gas seals are important components of rotating machinery. In the industry, some machines have presented instability problems due to the pressure and velocity distribution in the rotor-seal clearance. Thus, it is important to characterize the behavior in order to guarantee safety operation conditions. Aiming at this goal, a test rig is being built at the Federal University of Rio de Janeiro, int the Acoustics and Vibration Laboratory. This paper shows some preliminary experimental results.

Keywords: annular gas seals, rotordynamics, identification methods

1 INTRODUCTION

Annular gas seals are important components of rotating machinery. Their function is to avoid the backwards flux from a high to a low pressure stage in centrifugal compressors. Smith [6] and Cochrane [2] showed some instabilities caused by seals. Some theoretical models has been proposed by considering one or two control-volumes approach, and applying conservation laws, as shown by Iwatsubo [4], Childs [1], Wyssman [9] abnd Dietzen and Nordmann [3]. Experimental validation of the models has been carried out by General Electrics [7], Siemens [8] and Texas A&M University. They have a common procedure to identify the coefficients. The test rigs used are composed of a rigid rotor, bearings, actuators, honeycomb/hole-pattern/labyrinth seals and instrumentation. The rotor is excite transversely and different measurements are taken. The coefficients depends on several variables such as pressure drop, gas density, rotor speed, tangential speed, among others.

The purpose of this paper is to show the procedure of the identification of a labyrinth seal in a test rig built at the Federal University of Rio de Janeiro [5] and show some experimental results. The stiffness and damping coefficients are determined by computing an impedance matrix composed of seal-rotor displacements and magnetic forces.

2 ANNULAR GAS SEALS

The force model of the annular seals are consider linear on a certain operation condition, as shown in Equation (1). The model is reasonable to explain the synchronous instabilities related to the four parameters: Crossed-couple stiffness and damping and direct stiffness and damping. Nevertheless, in frequency domain, their values depend on several variables, such as pressure drop, rotor speed, inlet circumferential speed, clearance, among others.

$$\begin{bmatrix} -f_{xs} \\ -f_{ys} \end{bmatrix} = \begin{bmatrix} K(\Omega) & k(\Omega) \\ -k(\Omega) & K(\Omega) \end{bmatrix} \begin{bmatrix} u_{xs} \\ u_{ys} \end{bmatrix} + \begin{bmatrix} C(\Omega) & c(\Omega) \\ -c(\Omega) & C(\Omega) \end{bmatrix} \begin{bmatrix} \dot{u}_{xs} \\ \dot{u}_{ys} \end{bmatrix} \tag{1}$$

3 MAGNETIC ACTUATORS

The magnetic actuators are responsible for applying forces to the rotor in order to excite the seals dynamics. The magnetic actuators are composed of a set of 8 coils arranged in a circular configuration and a ferromagnetic steel. When an electric current is applied to the coils, electromagnetic forces are created between the actuator and the rotor. The force interaction can be modeled as shown in Equation (2), where the coefficients are detailed in [5].

$$f_k = \frac{\xi_k \beta \mu_o N^2 A_g}{2} \left(\frac{(i_0 + i_k)^2}{(g_0 - k)^2} - \frac{(i_0 - i_k)^2}{(g_0 + k)^2} \right) \ , \ k = \{x, y\} \tag{2}$$

4 IDENTIFICATION METHOD

The parameters are identified by simultaneously exciting and measuring the magnetic forces and seal-rotor displacements. Then, the impedance matrix

$$\mathbf{H}(\Omega) = \mathbf{F}(\Omega)\mathbf{U}(\Omega)^{-1} = \begin{bmatrix} K_{xx} + i\Omega C_{xx} & K_{xy} + i\Omega C_{xy} \\ K_{yx} + i\Omega C_{yx} & K_{yy} + i\Omega C_{yy} \end{bmatrix}, \tag{3}$$

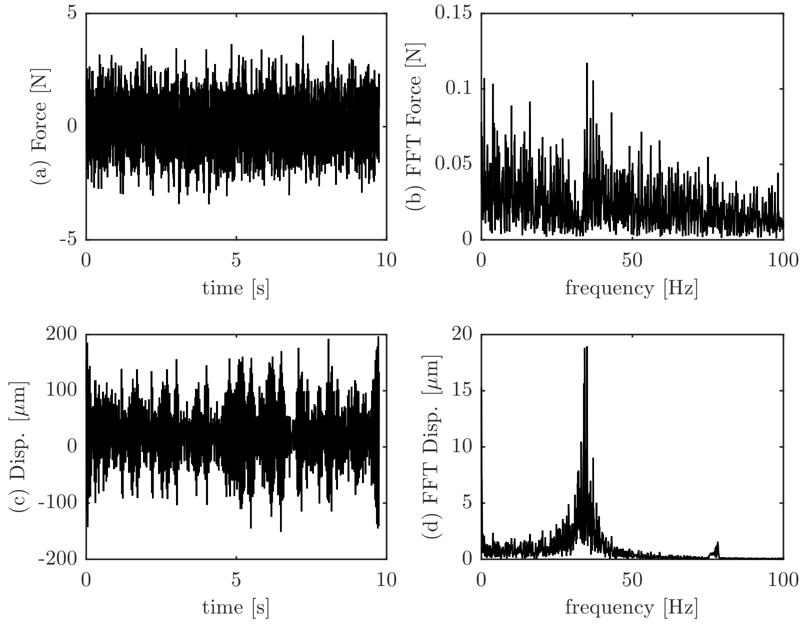

Figure 1 – Force and displacements signals.

computed in the frequency domain, is formed by the force and displacements matrices

$$\mathbf{F} = \begin{bmatrix} F_{xx} & F_{xy} \\ F_{yx} & F_{yy} \end{bmatrix}, \mathbf{U} = \begin{bmatrix} U_{xx} & U_{xy} \\ U_{yx} & U_{yy} \end{bmatrix}, \tag{4}$$

where the first subscript corresponds to the excitation direction while the second subscript is related to the direction of the measurement. The stiffness coefficients K and k are obtained by averaging K_{xx} and K_{yy}. The damping coefficients C and c are obtained by averaging C_{xy} and C_{yx}.

Since the displacement measurements are noisy, the inversion of the matrix may induce errors on the estimation. Thus, the estimator

$$\mathbf{H}(\Omega_k) = \left(\frac{1}{N_b}\sum_{l=1}^{N_b}\mathbf{F}(\Omega_k)^{(l)}\mathbf{U}^*(\Omega_k)^{(l)}\right)\left(\frac{1}{N_b}\sum_{l=1}^{N_b}\mathbf{U}(\Omega_k)^{(l)}\mathbf{U}^*(\Omega_k)^{(l)}\right)^{-1}, \tag{5}$$

is used to reduce the influence of the noise on the parameters value by averaging power spectral densities, where N_b are the number of blocks for the averages.

A two-step procedure is carried out to filter the effects of other components from the machine. First, a baseline impedance matrix is obtained with no pressure. Then, the test is realized with a pressure drop and the test impedance matrix is computed. Finally, both matrices are substracted in order to obtain only the effects from the seals. Equation (6) shows the seal impedance matrix.

$$\mathbf{H}_{\text{seal}}(\Omega) = \mathbf{H}_{\text{exp}}(\Omega) - \mathbf{H}_{\text{base}}(\Omega) \tag{6}$$

5 EXPERIMENTAL RESULTS

In this section, experimental results from the test rig are shown for a certain operation conditions. The tests were carried out with a labyrinth seal at a 3-bar inlet-pressure, at atmospheric outlet-pressure, with a 30° pre-swirl ring and with a rotor speed of 4500 RPM. The excitation signal used is a white noise from 0 Hz-100 Hz. All measurements were acquire at 5.128 KHz and the spectra were calculated with 5128 samples and with a resolution of 1 Hz. The impedances were averaged with 30 blocks of measurements.

The experiment results for the baseline are shown in Figures 1, 2 and 3. Figure 1a shows the excitation force along the x-direction and Figure 1b its spectrum. It can be seen that the signal excites all the frequency components. Figure 1c shows the seal displacement in the x-direction and Figure 1d its spectrum. The signal excites different frequency components, but the amplitude is greater at the first critical speed (39 Hz). The seal stiffness coefficients can be seen in Figure 2. Figure 2a shows the direct stiffness for 0 Hz-50 Hz. It clearly shows that the coefficient value increases with the excitation frequency. On the other hand, the cross-coupled stiffness from Figure 2b remains constant with a small value. Figures 3a and 3b show the direct and cross-coupled damping coefficient. In general, the seal does not exhibit a high damping coefficient.

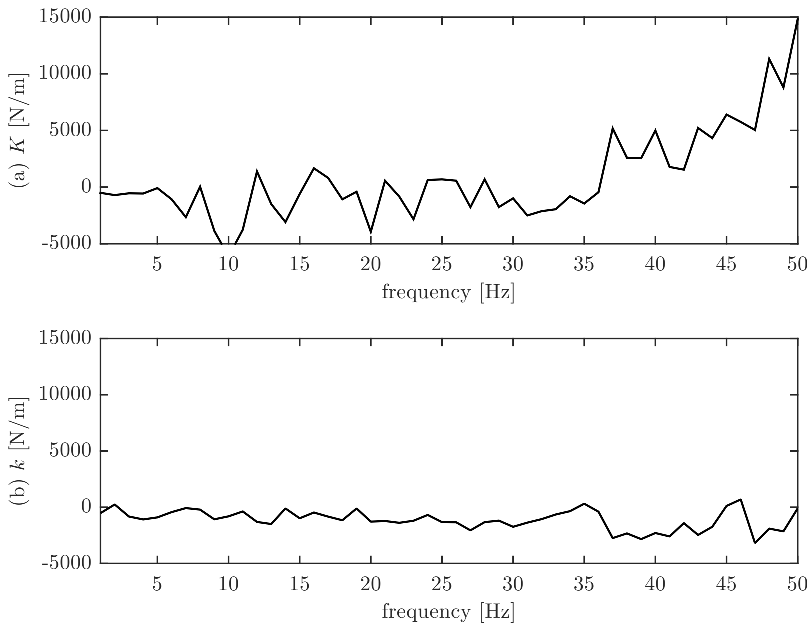

Figure 2 – Direct and cross-couple stiffness coefficients of the seal.

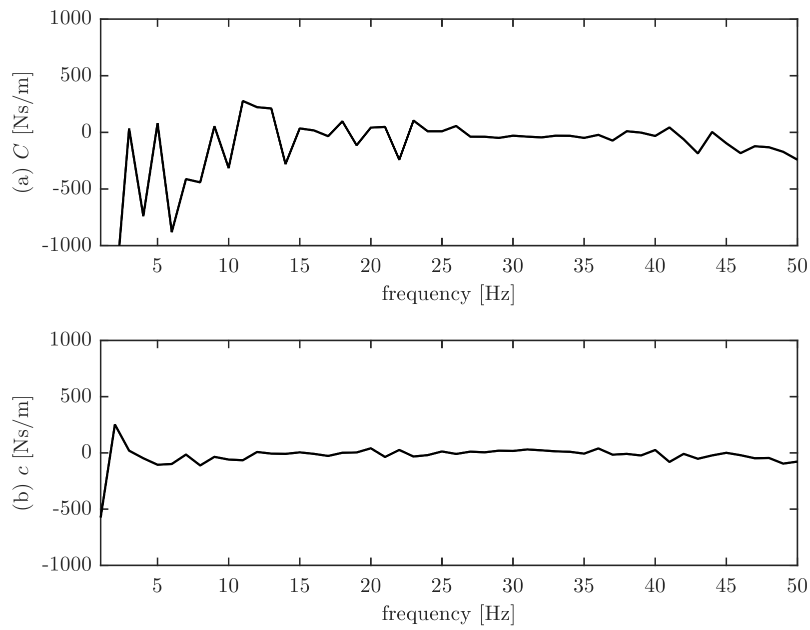

Figure 3 – Direct and cross-couple damping coefficients of the seal.

6 CONCLUSIONS

This paper showed the experimental results of a test rig for the identification of annular gas seals coefficients. The coefficients for a labyrinth seal were obtained for 0 Hz-50 Hz. The direct stiffness grew with the excitation frequency. On the other hand, the cross-coupled stiffness presented a low value and constant with the frequency. Finally, the damping coefficients were close to zero.

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