Identification of Rotor Dynamic Parameters of Seals Using Impact Hammer Method

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ABSTRACT

A frequency domain technique for the experimental identification of rotor dynamic parameters of seals in rotor-bearing-seal systems by using the impact hammer method is presented. For the present configuration the seal is treated as floating and the rigid rotor is mounted on rolling bearings. Rotor- fluid-seal interaction has been modeled by ten linearised rotor dynamic parameters (four each for the stiffness and the damping and two for the inertia), which are assumed from the existing theoretical models for the numerical simulation of the proposed identification technique. A bell shape forcing function is used to model the impulse force. Displacement responses due impulse forces is generated in the time domain and transformed to the frequency domain by using the Fourier transform. By using these forces and responses, the seal rotor dynamic parameters are estimated by the proposed technique based on least squares estimation. Responses corresponding to different number of excitation frequencies are considered during the estimation. Good comparison of estimated parameters with assumed parameters that is used to generate responses does the validation of the present method. Adding noise in simulated responses checks the robustness of the technique.

Keywords: Seals, Rotor dynamic parameters, Identification, Fourier transform.

1. Introduction

Rotary seals in the high-speed and the high-pressure operations of compressors in the industrial applications and turbopumps of the space shuttle main engine lead to instability .The main factor, which governs the instability, is the rotor dynamic parameters (RDPs) of seals. Although the importance of seals RDPs is generally well recognized by the design engineer it is often the case that theoretical models available for predicting it are accurate for very specific cases. Moreover, RDPs of seals are greatly dependent on many physical and mechanical parameters such as the lubricant and working fluid temperatures, pressure drops, seal clearances, surface roughness and patterns, rotor speeds, eccentricity and misalignments and these are difficult to obtain accurately in actual test conditions. It is for this reason that designers of high-speed rotating machinery prefer experimentally estimated RDPs of seals in their calculations.

Nordmann and Scholhom (1980) described a procedure to identify the stiffness and damping parameters of journal bearings is the most economical and convenient. Chan and White (1990) used the impact method to identify bearing dynamic parameters in a rotor mounted on two symmetric bearings by curve-fitting frequency responses. Since in many applications rotor-bearing systems bearings are not symmetric, the assumption of the symmetry limits the application of the method. Ammugam *et al.* (1995) extended the method of structural joint parameter identification proposed by Wang and Liou (1991) to identify the linearised oil-film parameters utilizing the experimental frequency response functions (FRFs) and theoretical FRFs obtained by finite element modelling. Qiu and Tieu (1997) presented an algorithm for the identification rotor dynamic parameters of journal bearings from impulse responses. Tiwari *et al.* (2004 and 2005) gave a comprehensive survey of experimental identification of rotor dynamic parameters of bearings and seals, respectively. In this paper, a test rig and a procedure for identification of RDPs of seals is presented

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by using the impact hammer method. Numerical validation of the proposed identification procedure has been performed on the proposed test rig. The estimation is quite encouraging even in the presence of noise in responses, which proves the robustness of the present estimation procedure.

2. System Modelling

A simple schematic of the rotor-bearing-seal test rig is shown in Figure 1 and it is used in the numerical test example. The rigid rotor is supported in two rolling bearings. Two identical seals are placed in the stator assembly. The stator is supported by a support spring of the effective stiffness of $2k_s$. Let $2m_s$ be the mass of the stator, L is the distance between centers of seals, R is the radius of the seal and c_r is the radial clearance of seals. Bearings are fixed to rigid pedestals. From Figure 1, the linearized equations of motion of the stator system is

$$[M]\{\ddot{q}\} + [C]\{\dot{q}\} + [K]\{q\} = \{f_\xi\} \tag{1}$$

where, ξ denotes x or y directions, matrices $[M]$, $[C]$ and $[K]$ are respectively the system mass, damping and stiffness matrices and vectors $\{q\}$ and $\{f_\xi\}$ are respectively the displacement and force (impact) vectors. Table 1 summarizes all the matrices and vectors. Subscripts: s and sl represent stator and seal respectively, k_{xx} and k_{xy} are the direct and cross-coupled stiffness respectively, c_{xx} and m_{xx} are respectively the direct damping and inertia terms, k_s is the stiffness of the support system (including support spring, bellow seals, hose connections). In the next section, an estimation equation for the seal RDPs is derived.

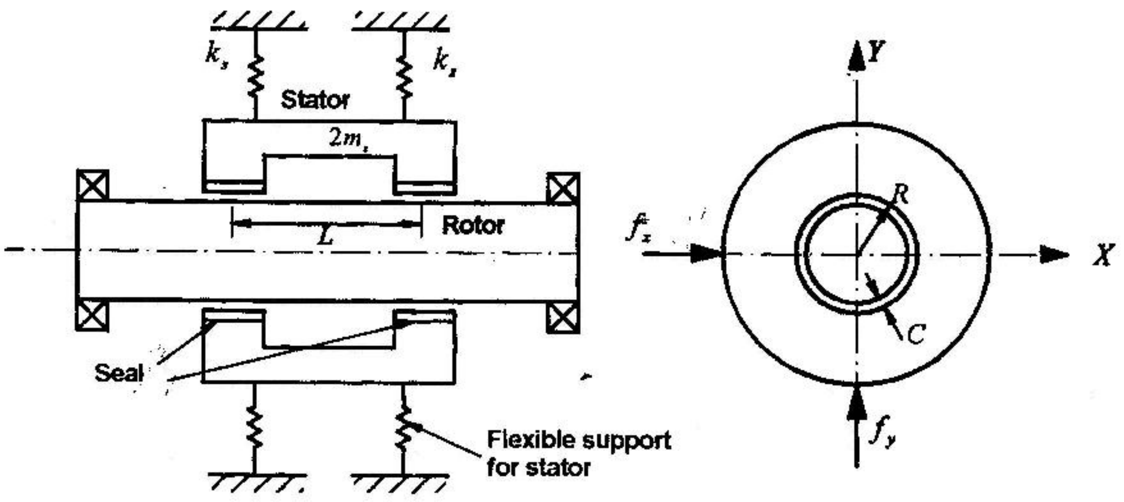

Figure1. A schematic diagram of the test rig

Table 1 System matrices and vectors

$[M]=[M_s]+[M_{sl}]=\begin{bmatrix} m_s & 0 \\ 0 & m_s \end{bmatrix}+\begin{bmatrix} m_{xx} & 0 \\ 0 & m_{yy} \end{bmatrix}$; $[C]=[C_{sl}]=\begin{bmatrix} c_{xx} & c_{xy} \\ c_{yx} & c_{yy} \end{bmatrix}$; $\{q\}=\{x \;\; y\}^T$
$[K]=[K_s]+[K_{sl}]=\begin{bmatrix} k_s & 0 \\ 0 & k_s \end{bmatrix}+\begin{bmatrix} k_{xx} & k_{xy} \\ k_{yx} & k_{yy} \end{bmatrix}$; $\{f_\xi\}=\begin{cases} \{f_x\}=\lfloor f_x(t)\; 0 \rfloor^T \\ \{f_y\}=\lfloor 0 \;\; f_y(t) \rfloor^T \end{cases}$

3. Identification Procedure

On performing fourier transform of equation (1) system equations in the frequency domain becomes

$$\left[([K_s]+[K_{sl}])-\Omega^2([M_s]+[M_{sl}])+j\Omega[C_{sl}]\right]\{Q_\xi(j\Omega)\}=\{F_\xi(j\Omega)\} \tag{2}$$

with

$$\{Q(j\Omega)\} = \frac{2}{NT}\sum_{n=0}^{N-1}\{q(nT)\}e^{-j\Omega nT} \quad \text{and} \quad \{F(j\Omega)\} = \frac{2}{NT}\sum_{n=0}^{N-1}\{f(nT)\}e^{-j\Omega nT} \qquad (3)$$

Equation (3) is separated into the real and imaginary parts, which gives

$$\{Q(j\Omega)\} = \{Q^r + jQ^i\} = \{X^r + jX^i \;\; Y^r + jY^i\}^T \quad \text{and} \quad \{F(j\Omega)\} = \{F^r + jF^i\} = \{F_x^r + jF_x^i \;\; F_y^r + jF_y^i\}^T \quad (4)$$

At frequency $\Omega = \Omega_p (p = 1,2,3...., m)$, where m indicates the number of excitation frequencies, on substituting equation (4) into equation (2) and separating the real and imaginary parts, we get

$$[K_{sl}]\{\Omega^r_{\xi p}\} - \Omega^r_p[M_{sl}]\{\Omega^r_{\xi p}\} - \Omega_p[C_{sl}]\{\Omega^i_{\xi p}\} = \{P^r_{\xi p}\} \qquad (5)$$

and

$$[K_{sl}]\{\Omega^i_{\xi p}\} - \Omega^2_p[M_{sl}]\{\Omega^i_{\xi p}\} + \Omega_p[C_{sl}]\{\Omega^r_{\xi p}\} = \{P^i_{\xi p}\} \qquad (6)$$

with

$$\{P^r_{\xi p}\} = \{F^r_\xi\} - [K_s]\{Q^r_{\xi p}\} + \Omega^2_p[M_s]\{Q^r_{\xi p}\} \text{ and } \{P^i_{\xi p}\} = \{F^i_\xi\} - [K_s]\{Q^i_{\xi p}\} + \Omega^2_p[M_s]\{Q^i_{\xi p}\} \quad (7)$$

where $\{P_{\xi p}\}$ is the vector of known terms. Equations (5) and (6) can be rearranged to give

$$[A_{\xi p}]\{Z\} = \{D_{\xi p}\} \qquad (8)$$

with

$$[A_{\xi p}] = \begin{bmatrix} X^r_{\xi p} & Y^r_{\xi p} & 0 & 0 & -\Omega_p X^i_{\xi p} & -\Omega_p Y^i_{\xi p} & 0 & 0 & -\Omega_p^2 X^r_{\xi p} & 0 \\ 0 & 0 & X^r_{\xi p} & Y^r_{\xi p} & 0 & 0 & -\Omega_p X^i_{\xi p} & -\Omega_p Y^i_{\xi p} & 0 & -\Omega_p^2 Y^r_{\xi p} \\ X^i_{\xi p} & Y^i_{\xi p} & 0 & 0 & \Omega_p X^r_{\xi p} & \Omega_p Y^r_{\xi p} & 0 & 0 & -\Omega_p^2 X^i_{\xi p} & 0 \\ 0 & 0 & X^i_{\xi p} & Y^i_{\xi p} & 0 & 0 & \Omega_p X^r_{\xi p} & \Omega_p Y^r_{\xi p} & 0 & -\Omega_p^2 Y^i_{\xi p} \end{bmatrix};$$

$$\{D_{\xi p}\} = \{P^r_{\xi p}(1) \;\; P^r_{\xi p}(2) \; P^i_{\xi p}(1) \; P^i_{\xi p}(2)\}^T \text{ and } \{Z\} = \{k_{xx} \;\; k_{xy} \; k_{yx} \; k_{yy} \;\; c_{xx} \; c_{xy} \; c_{yx} \; c_{yy} \; m_{xx} \; m_{yy}\}^T \quad (9)$$

Equation (8) is four linear simultaneous equations with ten unknowns, which is an under-determined regression equation. Writing equation (8) for d impacts (alternately in x and y directions) and for m frequencies, it yields $4dm$ linear simultaneous equations of the following form

$$[A]\{Z\} = \{D\} \qquad (10)$$

with

$$A = \{A_{11} A_{12} ..... A_{1m} \;\; A_{21} \; .... A_{dm}\}^T; \; D = \{D_{11} D_{12} ..... D_{1m} \;\; D_{21} \; .... D_{dm}\}^T; \; d \geq 2 \qquad (11)$$

Equation (10) is an over-determined regression equation and it can be solved by least squares method as

$$\{\widetilde{Z}\} = ([A]^T [A])^{-1} [A]^T \{D\} \qquad (12)$$

Equation (12) is the required regression equation for obtaining seals RDPs experimentally. In the present paper a numerical simulation has been performed to test the developed identification algorithm.

4. Numerical Example

To identify RDPs of seals by using equation (12), it requires measurement of stator responses due to impact forces. For numerical simulation, equations of motion (equation 1) are solved in the time domain. The impact force (Figure 2) is approximated by the bell-shape function of the following form

$$f(t) = f_o exp\,[-2(2 \text{ in } 10)\,(t - t_i)^2 / t_i^2] \qquad (13)$$

where f_0 is the amplitude of the applied impulse force, t_i is the instant at which the impulse force is applied and t is the instant time. The RDPs of seals have been generated from closed form expressions (Childs, 1983) and are listed in Table 2 corresponding to the following seal dimensions and operating conditions: L/D ratio=0.75, clearance=0.2 mm, pressure difference = 40 bar and rotor speed=14,000 rpm. For parametric study, the half of the stiffness of the stator support k_s has been varied from 0 to 10^5 N/m. In actual practice, the stiffness of the stator support system can be obtained by the impact test at baseline conditions (i.e. no load, without shaft rotation and at ambient fluid pressure). The resulting response measured in the time domain is transformed into the frequency domain to obtain the predominant frequency peak corresponding to the natural frequency of the stator support system. The stiffness of the stator support can be obtained by

$$k_s = m_s\,\omega_n^2 \qquad (14)$$

The data taken for the numerical simulation are as follows: mass of the stator = 2kg, impact force magnitude in the x- and y- directions are 125 N and 160 N respectively, sampling frequency=6375 samples/s and the instant at which impulse force is applied t_i = 0.014s.

5. Results and Discussions

Figure 2 shows impact forces applied to the stator in the x and y directions. Figure 3 shows the corresponding simulated responses obtained from assumed RDPs of seals. Due to high damping oscillations are not observed. The simulated forces and noisy responses are fed to the identification algorithm to get back the RDPs of seals. Effects of various parameters have been investigated as discussed in the following subsections.

5.1. Effect of number excitation frequencies and the noise level in the response

Estimated RDPs of seals are better when the number of excitation frequencies is increased in the identification algorithm. Especially, damping parameters are accurate when the number of excitation frequencies is increased. In practical situations, responses are always corrupted with noise, hence in the simulation, noise signals with a magnitude up to 5% of displacement amplitude is introduced in simulated responses. Estimated RDPs are consistent even in the presence of noise. This exercise shows the robustness of the identification algorithm. These results are summarised in Table 2.

5.2. Parametric study for the stator mass and the support stiffness

For different stiffness of the stator support k_s the variation in estimated RDPs is shown in Figures 4-6. It is observed that in all cases the percentage of error in the RDPs is quite low even with the noise. The increase in direct stiffness parameters (k_{xx}= k_{yy}) is nearly linear with the increase in stiffness. Cross-coupled stiffness (Figure 4) and damping parameters (Figure 5) remain constant with the increase in the k_s Similarly, direct inertia parameters affects significantly (Figure 6) with the increase in the k_s Hence, the stiffness of the flexible stator support should be very less, however, at the same time, it should be of such that it should keep the stator in desired eccentric positions.

For different stator masses, variation in the estimated RDPs of seals is shown in Figures 7-9. It is observed that in all cases the percentage of error is quite low even with noise. Estimated direct stiffness parameters (Figure 7) increase with the increase in the stator mass. The cross-coupled stiffness parameters k_{xy} are almost constant with the increase in stator mass value and k_{yx} is having less error at around m_s =5 kg. The direct and cross-coupled damping parameters (Figure 8) remain constant while increasing the stator mass. However, direct inertia parameters (Figure 9) are increasing with the increase in the stator mass. From the parametric study, it is observed that the stator mass should be as low as possible. For irrespective of the number of excitation frequencies, noise levels, support stiffness and stator masses the estimated direct inertia parameters have relatively high percentage of error, however, the estimated damping and stiffness parameters have less error (refer Table 2). Responses obtained from estimated RDPs of seals are compared with original signal generated in Figure 3. Both responses are almost close to each other and it shows the accuracy of the identification technique.

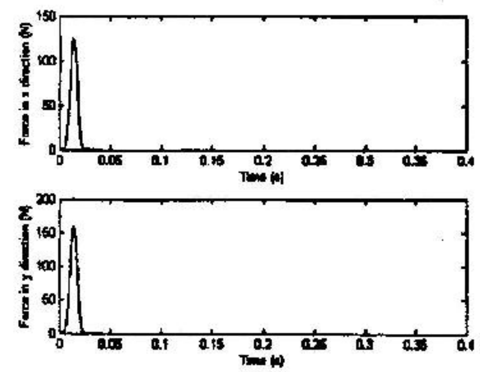

Figure 2 Impulse forces applied in the x and y directions.

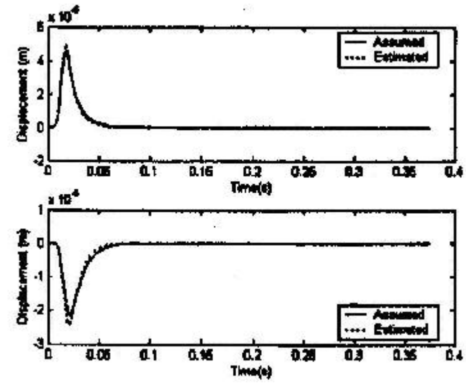

Figure 3 Assumed and estimated impulse responses in the x and y directions for the horizontal impact

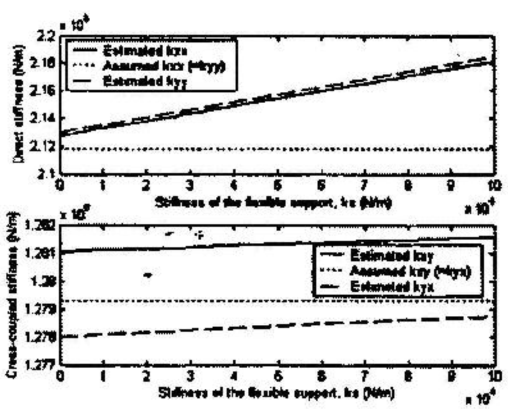

Figure 4. Effect of the flexible support stiffness (k_s) on the estimation of stiffness parameters of seals

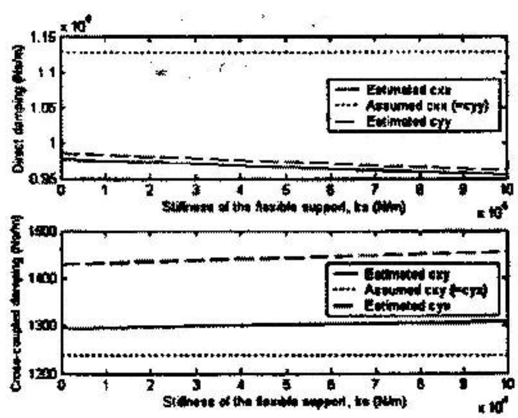

Figure 5. Effect of the flexible support stiffness (k_s) on damping parameters of seals

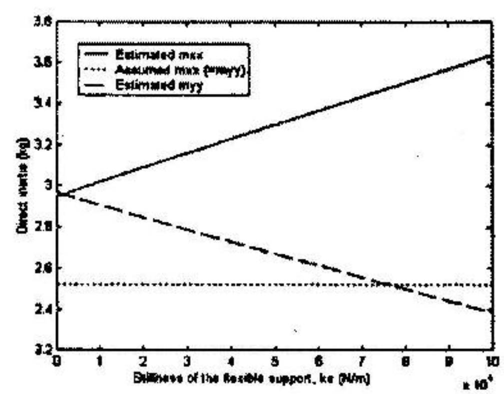

Figure 6. Effect of the flexible support stiffness (k_s) on direct inertia parameters of seals

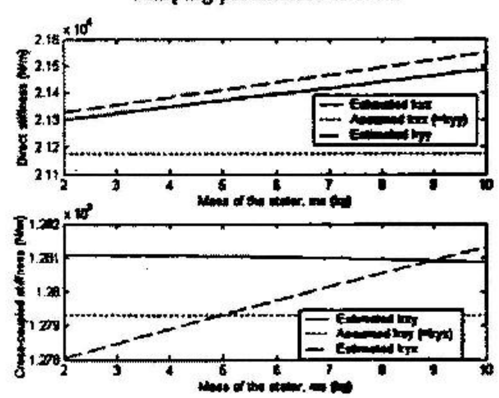

Figure 7. Effect of the stator mass (m_s) on stiffness parameters of seals

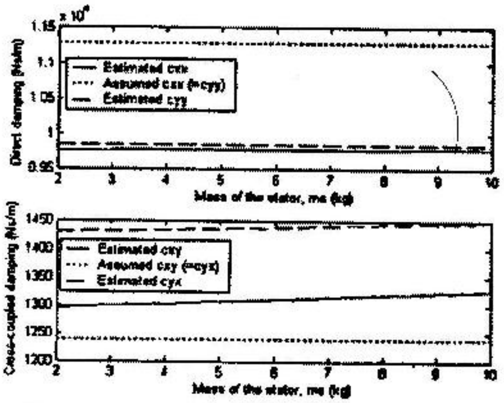

Figure 8. Effect of the stator mass (m_s) on damping parameters of seals

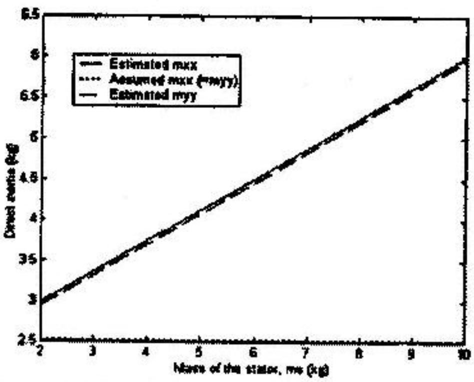

Figure 9. Effect of the stator mass (m_s) on direct inertia parameters of seals

6. Conclusions

A technique for the identification of rotor dynamic parameters of seals using the impact hammer method has been developed. Through numerical experiments effects of various test rig parameters (i.e. the stator support stiffness and the stator mass) and number of excitation frequencies and the measurement noise on the accuracy of the identified RDPs have been studied. Estimates of the stiffness have found to be excellent and that of the damping are fairly good and inertia parameters are poor. However, the presence of inertia parameters in estimation procedure improves the estimates of the stiffness and damping parameters. It is suggested in the actual test rig to provide lighter stator mass and flexible support stiffness to get better estimate of the RDPs.

Table 2. Estimated RDPs of seals with different excitation frequencies and noise level

Rotor dynamic coefficients of the seal	Assumed coefficients	Percentage error in the estimated coefficients for different excitation frequencies and noise level								
		8 excitation frequencies			16 excitation frequencies			64 excitation frequencies		
		No noise	1% noise	5% noise	No noise	1% noise	5% noise	No noise	1% noise	5% noise
k_{xx} (N/m)	2.12 e6	0.08	0.08	0.11	0.44	0.47	0.62	0.59	0.67	0.67
k_{xy} (N/m)	1.28 e6	-0.28	-0.30	-0.40	0.01	0.01	0.02	0.14	0.18	0.14
k_{yx} (N/m)	1.28 e6	0.16	0.15	0.11	-0.05	-0.04	0.03	-0.10	-0.05	-0.05
k_{yy} (N/m)	2.12 e6	0.41	0.38	0.24	0.62	0.62	0.59	0.72	0.74	0.49
c_{xx} (Ns/m)	11285.0	-21.32	-21.26	-21.02	-14.73	-14.61	-14.14	-13.51	-13.33	-12.94
c_{xy} (Ns/m)	1238.20	-8.19	-8.18	-8.14	2.16	2.20	2.36	4.59	4.78	2.73
c_{yx} (Ns/m)	1238.20	0.89	1.40	3.41	12.70	13.68	17.45	15.56	16.72	16.06
c_{yy} (Ns/m)	11285.0	-20.74	-20.81	-21.10	-14.07	-14.15	-14.49	-12.83	-12.88	-13.92
m_{xx} (kg)	2.52	5.89	6.61	9.49	20.38	22.44	30.64	18.40	21.47	19.26
m_{yy} (kg)	2.52	-39.75	-39.07	-36.35	2.35	4.13	11.25	16.59	19.96	24.27

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