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DYNAMIC COEFFICIENTS OF STEPPED LABYRINTH GAS SEALS

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ABSTRACT

Rotor-fluid interactions can cause self-excited shaft vibrations of high density turbomachinery. Often the amplitude of the vibrations reaches unacceptably high amplitudes and the scheduled power or running speed cannot be achieved. One of the most important sources of excitation is the flow through labyrinth seals. For a reliable design it is necessary to predict these forces exactly, including not only stiffness but also damping coefficients.

As the forces in labyrinth gas seals are rather small only minimal experimental data is available for the comparison and validation of calculations. Meanwhile a new and easy-to-handle identification procedure enables the investigation of numerous seal geometrys. The paper presents dynamic coefficients obtained with a stepped labyrinth and the comparison with other seal concepts.

NOMENCLATURE

a	Amplitude of the vibrational mode
C, E, K	Direct damping, inertia, stiffness coefficients
c, e, k	Cross damping, inertia, stiffness coefficients
c_{u0}	Entry swirl
F_T	Lateral force
Δr	Change in direct stiffness of the magnetic bearing
Δp	Pressure difference
Δq	Change in magnetic excitation at the stability limit
x, y	Coordinates of displacement
Ω	Whirling frequency
STS14	Nomenclature of the stepped labyrinth
TOS24	Nomenclature of the look-through labyrinth (tooh on stator)

INTRODUCTION

High-density turbomachines often operate close to the stability limit which is characterized by a violent vibration behavior. In case of instability, the vibration due to unbalance is overlaid by a rapidly increasing vibration close to an eigenfrequency of the rotor. The unstable vibrations are of the self-excited type and they are mainly generated by hydrodynamic bearings and by the flow of the working fluid in the turbomachine. All exciting mechanisms are induced by lateral forces which transfer rotative energy into the rotor bending vibrations.

A mechanism which leads to fluid-generated forces, so-called steam whirl, was first described by Thomas [1958]. The forces are traced to non-uniform leakage losses which cause variable bladeforces and in result in a lateral and therefore exciting force. Alford [1965] discovered an additional effect which may increase the excitation. When the blades are shrouded, an asymmetric pressure distribution in relation to the rotor deflection results in a second excitation component. Meanwhile, it turned out that one of the main parameters influencing the asymmetry of the pressure distribution is the swirl of the flow at the entrance to the labyrinth seal.

The rotor-fluid interaction in gas seals is usually described by a linearized relation. The inertia coefficients neglected here are considered only in case of liquid seals (Childs [1993]).

$$\bar{F} = -\begin{pmatrix} K & k \\ -k & K \end{pmatrix}\begin{pmatrix} x \\ y \end{pmatrix} - \begin{pmatrix} C & c \\ -c & C \end{pmatrix}\begin{pmatrix} \dot{x} \\ \dot{y} \end{pmatrix} \quad (1)$$

With small motions around the centered position it can be assumed that the matrices of the stiffness coefficients K and k and the damping coefficients C and c are symmetric in the main diagonal and skew-symmetric in the secondary diagonal. Experimental investigations (Wagner and Steff [1996]) indicate, that the simplification mentioned above is justified even for small deviations out of the centered position. The linearity of the equation is given up to an eccentricity ratio of 0.5.

The destabilizing force caused by the cross-coupled stiffness is counteracted by the direct damping; it is therefore also important to know and to take into consideration the damping. The experimental identification of all the dynamic coefficients requires a motion of the

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rotating rotor relative to the stator. Most of the test rigs use heavy and stiff rotors and therefore the realization of a predefined motion is an ambitious task. The seal forces influence the dynamic behavior of the system. The response caused by the motion is an unsteady reaction force or the corresponding circumferencial pressure distribution, which must be measured with regards to amplitude and phase (Childs et al. [1986], Hawkins et. Al [1989]... Yu and Childs [1996]), Wright [1983], Millsaps and Martinez-Sanches[1993]).

Recent arrangements use active magnetic bearings to carry the rotor and to measure the forces simultaneously (Matros, Neumer and Nordmann [1994], Wagner and Pietruszka [1988]). The latter succeeded in measuring dynamic coefficients even under extreme conditions in pressure and rotational speed (Wagner and Steff [1996]). Presumably, only few results of this test facility will be published in the near future and will therefore be accessible to a broader public.

However only few experimental dynamic coefficients for gas seals are available at the moment. One of the reasons might be that the forces generated in gas seals are at least one order in magnitude smaller than in liquid seals. A survey of the experimental approaches and the investigated seal geometries is given by Childs [1993].

EXPERIMENTAL PROCEDURE

The new approach developed by the author tries to avoid some of the restrictions and difficulties caused by the rigid rotor concept (Kwanka and Mair [1995]). The starting point of the identification procedure is a Jeffcott-type rotor carried in fluid film bearings with a test seal and a magnetic bearing placed at the midspan position. The shaft is flexible.

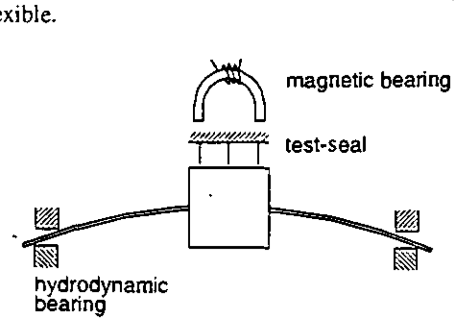


Fig. 1. Jeffcott rotor with test seal and magnetic bearing

The flow through the test seal causes a change in the dynamic behavior of the rotor. The (unknown) nonconservative acting coefficients as the cross-coupled stiffness and the direct damping alter the stability limit of the system. The conservative working coefficients, namely the direct stiffness and the cross-coupled damping, influence the vibrational frequencies. Both, the change in the stability and the frequency can be measured, which finally helps to find the dynamic coefficients.

An important role in the identification procedure is played by the magnetic bearing. It helps to simulate flow-like, displacement-dependent forces thus enabling an excitation of the rotor up to the stability limit; also, the frequency alteration caused by the flow can be compensated. Note, that the magnetic bearing is not used as a bearing in the traditional sense, but as an excitation source. The actual use of the magnetic bearing was inspired by Ulbrich [1988].

Two measurements are necessary to find the changes caused by the flow through the test seal. First, a measurement without flow through and second, a measurement with flow. The first measurement helps to identify the stability behavior of the rotor which is controled mainly by the dynamic chracteristics of the involved hydrodynamic bearings.

$$q_0 = \frac{(C\Omega a - ka)_{BEARING}}{a} \quad (2)$$

The flow through the test-seal alters the stability limit (and the whirl frequency) of the rotor. Again the acting magnetic excitation at the stability limit q can be measured.

$$q = q_0 - q_{SEAL} \quad (3)$$

The changes in magnetic excitation at the stability limit Δq and the magnetic direct stiffness Δr which is necessary to compensate the change in whirl frequency are known due to a previous calibration of the magnetic bearing. The following relations between the dynamic coefficients in eq. 1 and the measured changes Δq and Δr exist, when a circular whirling orbit is assumed.

$$\Delta q = q_{SEAL} = k - C\Omega \quad (4)$$

$$\Delta r = -K - c\Omega \quad (5)$$

The influence of the flow on the stability limit depending on the whirling frequency is displayed in fig. 2. As the stiffness of the shaft and the whirling frequency increases, less energy is transferred to the vibrational bending mode. The magnetic excitation at the stability limit q of the forward and backward vibration increases, too (continues line). The assumed forward excitation of the flow reduces the stability of the forward mode and elevates the stability of the backward mode (broken line). If the aerodynamic excitation is high, a magnetic stabilization becomes necessary (positive whirl frequency, area below the frequency axis).

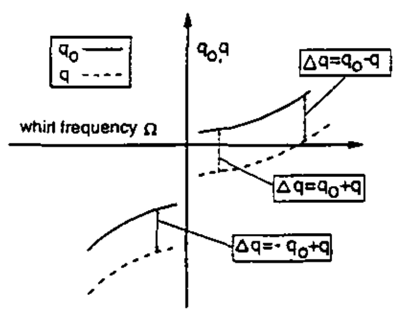


Fig. 2 Change in stability of the forward and the backward mode caused by the seal

When the measurement is repeated for at least two whirl frequencies, then all unknown coefficients can be calculated (eq. 4 and 5). The measured changes in magnetic stiffness come to lie on a straight line. The intersection with the ordinate gives the stiffness value and the slope of the line determines the damping value (s. fig. 3)

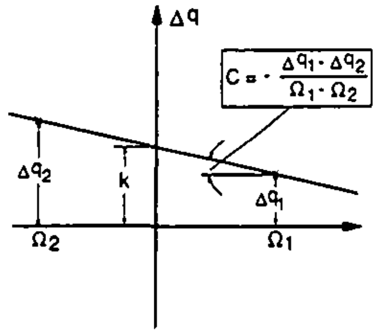


Fig. 3. Identification of the cross-coupled stiffness and the direct damping

To enhance the fitting reliability of the line needed for identification, in this paper the measurements were repeated for three forward and three backward whirling frequencies. An exact determination of the stability limit is essential in order to be able to obtain accurate coefficients. Usually, this limit can be fixed within ±1-2 N/mm (Kwanka [1996]).

The stability limit is characterized by a sudden and distinct change of the vibration amlitude and the orbit shape. In the frequency domain the vibrational signal contains in addition to the rotating frequency the unstable eigen-frequency. The spiral orbit of this band-filtered eigen-vibrations registered in about 0.5 seconds is documented in fig. 4.

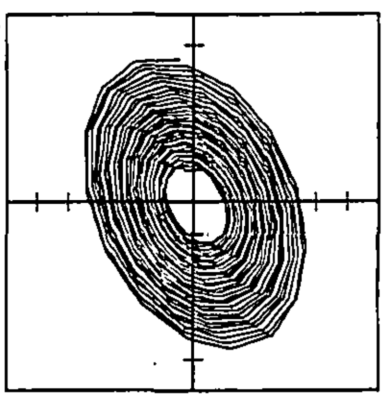


Fig 4 Spiral orbit of the unstabel eigen-vibration

The adjusted gain of the control system at the stability limit corresponds to a known cross-coupling (or direct stiffness) of the magnetic bearing. The linearized magnetic characteristics were extracted out of a static and a dynamic calibration procedure. The used configuration for the calibration was identical to that used in the subsequent experimental investigations. The maximum deviation of the force-current factor from the linear least square fit is about 5%. Since the vibrational orbit is frequency dependent a variable part of the calibrating range is involved and the average error will be considerably lower than the maximum value mentioned above. One identification run includes measurements with different frequencies and therefore the error will vary within the identification process.

The measurement accuracy and reproducibility were discussed in Kwanka [1997]. First measurements with the identification procedure are displayed in fig. 5 and should give an impression of the fluctuations abuot the fitted line which are representative for all of the following results (Kwanka and Mair [1995]). Each of the single dots represent the average of five Δq's measured with one whirl frequency. The used seal geometry is of the staggered type with two cavities. As expected the cross-coupled-stiffness (intersection with ordinate) and the direct damping (slope of the fitted line) increase along with the pressure ratio Δp.

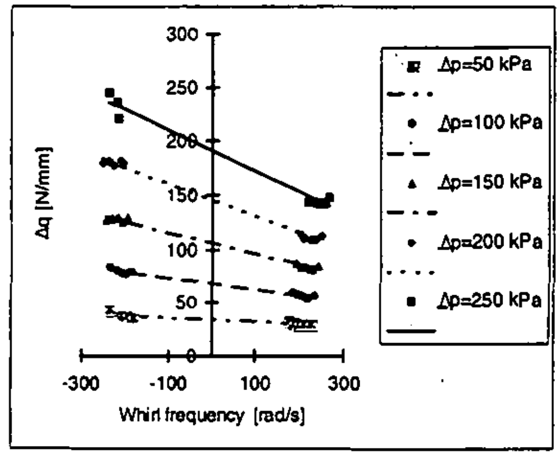
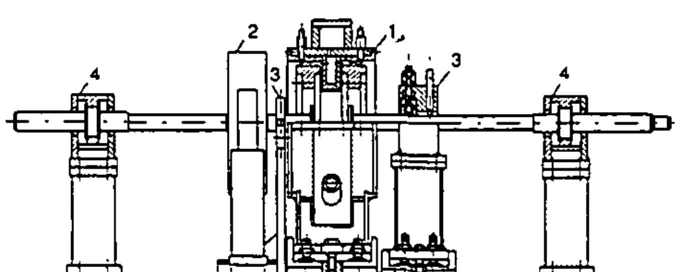


Fig. 5 Identification of the cross-coupled stiffness and the direct damping (dots – measurement; lines – curve fitting)

Without difficulties the procedure can be extended for the additional identification of the inertia coefficients. In this case, measurements with at least three whirling frequencies are necessary and a quadratic curve must be fitted.

As expected, the test rig build up to identify the coefficients is completely different from the other arrangements employed in this matter (fig. 6). The test seals including the casing are located there, at midspan position. Two identical seals are placed symmetrically to the guide vane ring which generates the entry swirl. Both, the seal rotor\stator and the guide vane ring can easily be replaced and changed to another geometry. The arrangement with two identical seals helps to double the small forces and average them at the same time. The magnetic bearing is mounted as close as possible to one site of the seal casing. Because the position of the magnetic bearing differs from the seal-position it is necessary to transfer the forces to a common point of application (Kwanka [1995]). The coefficients in fig. 5 include two seals without transfer.



1 test seal   3 catcher bearing
2 magnetic bearing   4 journal bearing

Fig. 6 Test rig for the identification of dynamic coefficients

The rotor with a diameter of 23 mm is carried in fluid film bearings and flexibly connected to a variable d.c. motor. The variation of the frequency is realized by varying the spacing between the bearings. To avoid violent vibrations with high amplitudes in the unstable vibration regime, retainer bearings are placed at both sides of the test-seal housing. After each change in the whirl frequency, which means a dismounting and remounting of the journal bearing pedestals, both measurements (with and without flow through the test-seal) are repeated.

EXPERIMENTAL RESULTS

As mentioned above, the test-seal internals and the guide vane ring can be easily removed and changed to other labyrinth geometries or guide vane angles. In this paper the main emphasis lies on a stepped labyrinth seal (STS14) with four cavities and strips on stator (s. fig. 7a). The experimental findings are compared to a tooth on stator labyrinth seal (TOS24; fig. 7b). Both seals have the same geometrical data in terms of clearance (0.5 mm) and number of cavities (4). The inflow region and the cavity length are identical. The only difference between the seals consists in the step of 1.5 mm, which leads to a reduction of the clearance area from the entrance of the seal to the exit.

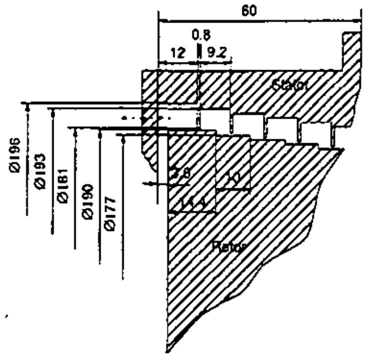


Fig. 7a Geometrical data of the stepped labyrinth seal (STS14)

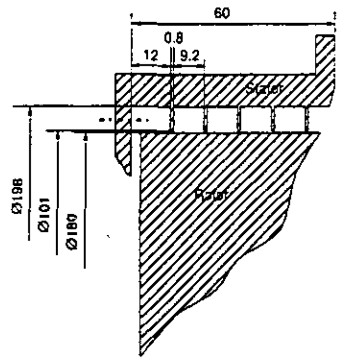


Fig. 7b Geometrical data of the tooth on stator labyrinth seal (TOS24)

The main task of labyrinth seals is still to minimize the leakage flow. First steps to improve the sealing behavior are: tightening the clearance, increasing the number of cavities or hindering in some way the flow in the cavity (e. g. by changing from the look-through to the stepped design). The expected decrease of the leakage of the stepped seal as compared to the looked- through seal is displayed in fig 8.

Unfortunately, the leakage flow of the TOS24 seal was measured only with two lower pressure differences. The mass flow through the stepped seal is approximated by a polynomial fit. The test medium is compressed air which is cooled, dried and metered by an orifice before entering the seals. The air is expanded to ambient conditions.

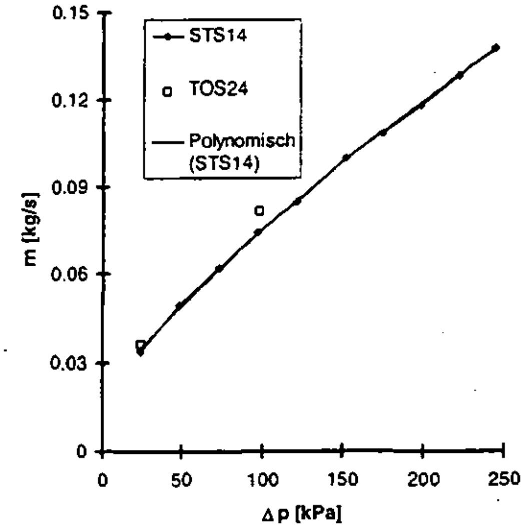


Fig. 8. Leakage loss of labyrinth seal versus the pressure difference.(n = 750 rpm)

The dynamic coefficients are measured in dependence of the entry swirl. When the pressure difference on the seal is held constant, the mass flow through the seal slightly decreases with higher swirl values (fig 9). Again, the better sealing performance of the stepped seal becomes obvious.

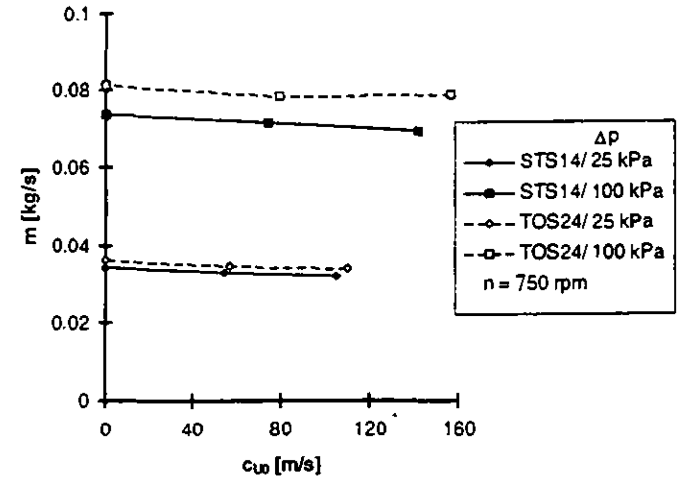


Fig. 9. Leakage rate in dependence of the entry swirl

The nonconservative coefficients have a considerable impact on the dynamics of the rotor. The cross-coupled stiffness excites the rotor, whereas the direct damping counteracts this excitation. The nonconservative coefficients of the stepped seal with two pressure differences are displayed in fig. 10.

The cross-coupled stiffness shows the usual almost linear behavior in dependence of the swirl condition at the entrance of seal. A higher pressure difference also leads to higher stiffness values. The dependencies of the direct damping on the pressure difference and the swirl is not that clear. Noteworthy is, however, the fact that the flow generates direct damping even without swirl.

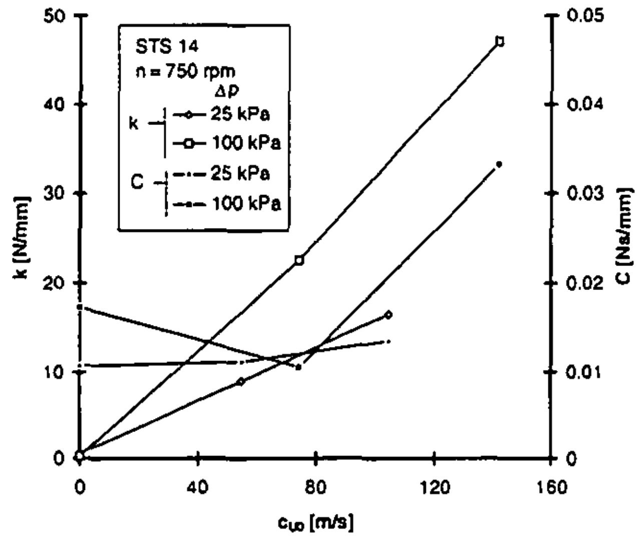


Fig. 10 Nonconservative dynamic coefficients of a stepped labyrinth seal

The conservative acting coefficients do not influence the stability of the rotor directly, but indirectly via the deflection of the vibrational mode. When the modal deflection increases due to a weakening influence of the seal, then more energy is transferred to the rotor and the stability limit decreases. The stepped labyrinth seal generates a positive direct stiffness and therefore stabilizing effect (fig. 11)..

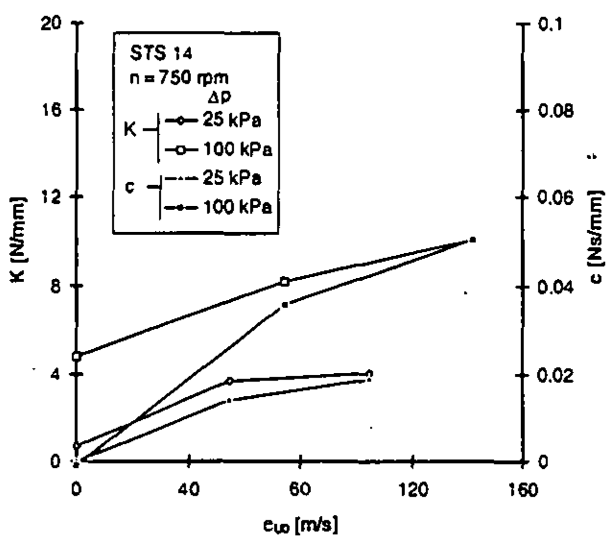


Fig 11. Conservative dynamic coefficients of a stepped labyrinth seal

Solely the influence on the frequency might be of interest to avoid a coincidence with the operating frequency. An elevation of the pressure difference or the entry swirl on the conservative coefficients leads to a moderate increase

Finally, a comparison of the nonconservative coefficients of the stepped seal with the very common look-through seal should indicate which seal concept is more suited from the dynamic point of view. In fig. 12, the cross-coupled stiffness and the direct damping of the stepped seal are related to the numbers of the tooth on stator seal. The cross-coupled stiffness of the stepped seal is distinctly smaller; unfortunately, at the same time the reduction of direct damping is more pronounced. As a consequence the better sealing labyrinth posses the poorer dynamic suitability. This behavior is in some respects comparable with the interlocking seal which also shows low direct damping values (Kwanka [1998]).

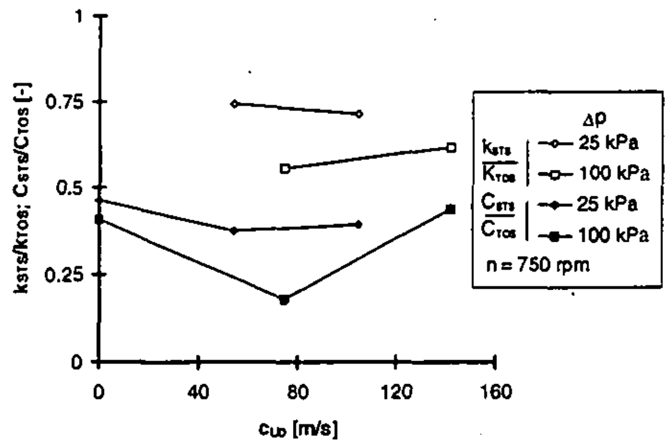


Fig. 12. Comparison of a stepped and a tooth-on-stator seal

CONCLUSIONS

A new easy-to-handle approach for the identification of the dynamic coefficients of labyrinth gas seals allows the direct and systematic comparison of seal concepts. The objective is to obtain an optimized seal concept which combines a good sealing and good a rotordynamic performance.

As expected, the stepped labyrinth seal leaks less than a look-through seal. The dynamic properties of the stepped seal aren't that favorable. When stepped labyrinth seals are used it must ensured that there is still enough damping in the system.

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