Experimental techniques for determining rotordynamic coefficients of gas seals: results for short staggered labyrinth seals and comparison with CFD

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ABSTRACT

This work deals with the rotodynamic characterization of gas labyrinth seals. The experimental determination of stiffness and damping coefficients at two different test rigs is described including undertaken modernization of the experimental environment. Test results on leakage and rotodynamic coefficients are shown for the three-teeth-on-stator staggered labyrinth seal with the tooth radial clearance of 0.5 mm. The results are compared to previous tests with a 0.27 mm clearance seal of the same design and CFD predictions.

1 INTRODUCTION

A large part of the world’s electrical energy supply is provided by power plants equipped with turbomachinery. Rising requirements for better efficiency lead to high power densities, more flexibility and challenging operating conditions. Reduction of leakage between rotating and stationary parts is one of the most important aspects of improvement of efficiency and ecological compatibility of modern turbomachines.

The most commonly used seal technology used at machine locations with high relative velocities are labyrinth seals. Constructed as contactless seals, they provide leakage reduction without mechanical losses and without excessive heat generation which might be caused by friction. Labyrinth seals are well known to generate destabilizing forces acting on the shaft [1], [2]. The resulting shaft vibrations depend on the operating conditions like pressure drop, swirl velocity, and rotational speed. The prevention of such seal-induced vibration problems is an important and challenging task [3].

This work presents experimental and numerical results on the leakage performance and rotodynamic coefficients for a three-teeth-on-stator staggered labyrinth seal with a rotor band under the second tooth (see Figure 1). Updated experimental environments and identification techniques are described in detail. Experimental investigations were performed for the labyrinth seal with a radial clearance of 0.5 mm over a wide range of operating conditions. The results are compared with available experimental data for the same sealing configuration, but different tooth radial clearance of 0.27 mm. Finally, theoretical rotodynamic coefficients predicted by a CFD method are presented demonstrating good agreement with the experiments.
EXPERIMENTAL SETUP AND TEST PROCEDURE

Over many years, the Institute for Energy Systems at the Technische Universität München conducts experimental investigations on leakage and rotordynamic coefficients of gas seals. There are two test rigs which use the same modular seal environment, but different measuring principles. In the no-whirl test rig, the rotating shaft does not undergo whirling motion, while in the dynamic test rig the shaft freely vibrates within the clearance of journal bearings. The no-whirl test rig is used to identify local direct and cross-coupled stiffness coefficients. The dynamic test rig is designed for determination of global stiffness and damping coefficients by the use of a magnetic actuator which is a proved method for rotordynamic experiments [1], [4].

Pressure differential, inlet swirl velocity, and shaft rotational speed can be varied during the tests. The operation parameters of both test rigs are summarized in Table 1. Compared with the no-whirl test rig, the range of operating parameters available on the dynamic test rig is limited by the capabilities of the magnetic actuator and stability of the tested seals.

Both test rigs use the same testing assembly. The working medium is air. The testing assembly consists of the inflow casing and two identical seals. Positioning the seals in double-flow configuration compensates axial forces acting on rotor and supports.

Figure 2 shows the schematic of the test rig air line including a leakage-independent swirl generator system. Compressed air coming from the faculty line is injected into the swirl chamber. The system pressure level is regulated by a pneumatic valve (1). Two bypass valves (2) are used to handle highly variable mass flow rates occurring during the tests of different seals (e.g. contact brush seals vs. labyrinth seals). The leakage mass flow passing the tested sealing

<table>
<thead>
<tr>
<th>Table 1 Operating conditions of the test rigs</th>
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<tr>
<td>Inlet pressure [MPa]</td>
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<tr>
<td>0.1 – 1.0</td>
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<tr>
<td>Inlet swirl [m/s]</td>
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<td>Shaft rotational speed [rev/min]</td>
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<td>Shaft relative eccentricity [-]</td>
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* Operating parameters depend on the rotordynamic stability characteristics of the seal.
configur**ation** is measured by a Rheonik Coriolis mass flow meter at the beginning of the air supply system (3). Pressure fluctuations occurring in the air supply grid and regulation system are buffered by a 500 l buffer vessel (4).

Depending on the control parameters of a throttle valve (6) and a bypass valve (7), one part of inlet air is led through a preswirl ring equipped with angled nozzles for tangential deflection into the tested seal system and is then discharged to ambient. The rest of air is exhausted through bypass channels back into the swirl generator circuit which is driven by a screw compressor (5). A single preswirl ring is used (see Figure 1), but the configuration of the air line with the air vessel and compressor located after the testing assembly allows of generating the preswirl value between 100 and 300 m/s independently of the inlet pressure. This is achieved by changing the mass flow rate through the bypass channels.

For the ongoing measurement campaign, a substantial modernization of the test rig equipment has been performed affecting many components: data acquisition system, pressure and leakage measurements, displacement sensors, magnetic actuator, air supply line, bearings and electric infrastructure. Data acquisition system of both test rigs was significantly revised. Signals from the displacement and temperature sensors, mass flow meters, and rotational speed sensor are handled now by several NI modules. The data acquisition program was redeveloped within the LabVIEW environment.

An old Scanivalve pressure sensor with forty eight pressure taps was replaced by two PSI 9116 pressure scanners with thirty two channels in total to enable a simultaneous measurement of all pressure locations with higher accuracy. The pressure taps are distributed on the circumference in the seal cavities 1 and 2, ten taps in each cavity (see Figure 1). In both inlet cavities, the pressure level is determined by averaging pressure measurements at four locations. The inflow swirl velocity is calculated in the real-time from the static and total pressure measurements. Total pressure is measured with a Pitot tube placed in the inlet zone direct before the seal entrance. The rotor position is determined by eddy current sensors mounted in the outlet rings of both sealing configurations.

Partially damaged displacement sensors were replaced by new Brüel & Kjær IN081 eddy current sensors for position and vibration measurements in both seals and in the magnetic actuator system.
Both test procedures require stationary air supply without any pressure fluctuations during experiments at different mass flow rates resulting from various tested seal geometries and test parameters. Two bypass valves were installed before and after the main pressure valve to enable constant regulation of the inlet pressure level for both low-leakage and high-leakage seals.

Procedures for experimental identification of the rotordynamic coefficients on the no-whirl and dynamic test rigs assume the following linear rotordynamic model

\[
\begin{align*}
F_r/e &= -K - c \Omega \\
F_t/e &= k - C \Omega
\end{align*}
\]

where \(F_r\) and \(F_t\) are radial and tangential components of the seal’s force; \(e\) is shaft eccentricity; \(K\) and \(k\) are direct and cross-coupled stiffness coefficients; \(C\) and \(c\) are direct and cross-coupled damping coefficients; \(\Omega\) is whirl frequency.

2.1 Tested seals
The tested seal is a short staggered three-teeth-on-stator labyrinth seal with a rotor band placed under the second tooth. Such a sealing configuration is typical for blade tip locations.

Geometry of the generic sealing configuration is shown in Figure 1. The mean diameter of the inlet is 203 mm and its height is 2 mm. The rotor band mounted as a shrink ring is 6 mm long and 3 mm high. The tooth width at root and at tip is 1 mm and 0.3 mm respectively. The tooth taper at tip is 75°. The radial tooth clearance is either 0.5 mm or 0.27 mm.

2.2 No-whirl test rig
Figure 3 shows a schematic of the no-whirl test rig and the used measurement method. A rigid rotor is supported by high precision ball bearings. This rotor arrangement guarantees a precise rotor-stator alignment and prevention of any whirling motion of the shaft. After the inspection, the oil lubrication was replaced by the use of sealed ball bearings with grease lubrication to avoid aerosol emissions at high rotational speeds. The rotor position within the seal clearance can be exactly preset by hydraulic alignment of the seal casing. To reduce rotor misalignment due to thermal growth of the bearing blocks at high speed operation, a new forced air cooling system was developed for the ball bearings. The rotor is driven by a
variable-speed direct-current motor with the maximal rotational speed of 20000 rpm which is limited to 12000 rpm by the bearings.

Integration of the circumferential pressure distribution measured in two sealing cavities at eccentric rotor alignment gives the aerodynamic radial and tangential forces acting in the seal

\[
F_r = F_r^1 + F_r^2; \quad F_r^{1,2} = -Rt \int_0^{2\pi} p^{1,2} \cos \varphi \, d\varphi
\]

\[
F_t = F_t^1 + F_t^2; \quad F_t^{1,2} = -Rt \int_0^{2\pi} p^{1,2} \sin \varphi \, d\varphi
\]

(2)

Determination of forces is based on the assumption that the circumferential pressure distribution remains relatively constant along the cavity axial length \( t \).

Knowing the shaft eccentricity, the local direct and cross-coupled stiffness coefficients can be easily calculated

\[
K = -F_r/e \quad \text{and} \quad k = F_t/e
\]

(3)

The no-whirl test rig experimental procedure is to perform several measurements of the circumferential pressure distribution at different shaft eccentricities. The local stiffness coefficients are obtained from the least square approximation of the radial and tangential forces. The identification procedure does not take into account inflow and outflow cavities, as well as regions under the labyrinth teeth; therefore the obtained stiffness coefficients are called local coefficients.

2.3 Dynamic test rig

Figure 4 shows a schematic of the dynamic test rig. Measurements on a whirling rotor system allow determination of global stiffness and damping coefficients [4]. For that purpose, a flexible shaft is supported by fluid-film bearings in the dynamic test rig. The bearing span can be slightly changed. A magnetic actuator is used for the rotor excitation. Catcher bearings with adjustable clearance protect the tested

Figure 4. The dynamic test rig
seals and magnetic actuator for the case of highly unstable behaviour or failure. The magnetic actuator equipped with new eddy current displacement sensors was anew calibrated. The dynamic rotor-shaft unit was balanced.

For the determination of stiffness and damping coefficients, the rotor is excited up to its stability threshold by applying the magnetic tangential force. The stability of the rotor is controlled by the orbit shape and frequency. The first run is performed without pressurization to obtain rotordynamic coefficients of the whole testing assembly except the tested seals. The second run is performed with pressurization. The forces acting in the seal alter the stability threshold and the whirl frequency of the rotor. Therefore, the magnetic normal force must be applied to compensate the whirl frequency shift compared with the run without pressurization. Magnetic forces are controlled by two coefficients – magnetic excitation coefficient $q$ and magnetic stiffness $r$.

Increasing the positive excitation leads to unstable conditions and forward whirl, while negative excitation effects a stabilization of the rotor system up to a backward whirl direction which can be damped by positive excitation.

Positive magnetic stiffness has a stiffening effect on the rotor system and leads to a higher natural frequency level. Negative magnetic stiffness has a softening impact on the rotor system.

The magnetic actuator is positioned close to the testing assembly, however not in the mid-span position. Therefore, the specific magnetic forces must be converted with the help of the beam-based finite element model of the dynamic test rig rotor system.

The difference between the control parameters of the calibrated magnetic actuator for the test runs with and without pressurization describes qualitatively the influence of the seals on rotordynamics, i.e. direct and cross-coupled stiffness and damping coefficients of the seal according to Eq (1)

$$\frac{F_r}{e} = r^0 - r^1$$
$$\frac{F_l}{e} = q^0 - q^1$$

As in the no-whirl test rig experimental procedure, the dynamic test rig procedure takes into account several measurements with varied whirl frequency $\Omega$. In addition to the switching the whirl direction (forward or backward), the frequency value is varied by changing positions of the bearing pedestals.

3 RESULTS AND DISCUSSION

This section compares the performance of the short staggered labyrinth seal with the tooth radial clearance of 0.5 mm (SSS 0.5) and 0.27 mm (SSS 0.27) respectively. The SSS 0.5 is tested on the updated test rigs, while the results for the SSS 0.27 were obtained in the previous works [5,6].

The leakage and rotordynamic coefficients of the tested labyrinth seals are also calculated by the computational fluid dynamics method. The gas flow within the sealing cavities is modelled using the Reynolds-Averaged-Navier-Stokes analysis with the Shear-Stress-Transport turbulence model. The rotordynamic coefficients are predicted form the same linear model in Eq. (1) by the whirling rotor method using three different whirl frequency values (zero, synchronous forward, and synchronous backward whirl). The detailed description and analysis of the CFD method can be found in [7].
Figure 5 provides a comparison of the experimental leakage for the SSS 0.5 and SSS 0.27 seals, as well as predictions for the SSS 0.5 seal (continuous line). The results are shown versus inlet pressure. As compared to SSS 0.27, the leakage increase in SSS 0.5 is about 40%. The predicted leakage performance of the SSS 0.5 seal is in good agreement with the measurements.

3.1 Local stiffness coefficients
The local coefficients determined on the no-whirl test rig are divided into the radially-oriented direct stiffness and the cross-coupled stiffness acting in tangential direction. Figure 6 provides the experimental local stiffness coefficients for the SSS 0.5 and SSS 0.27 seals, as well as predictions for the SSS 0.5 seal. Predictions and experimental results for the SSS 0.5 seals are presented for three preswirl velocities (small, medium, and high). The local stiffness coefficients for the SSS 0.27 are shown for various preswirl ranging from 20 m/s to 200 m/s. The experimental results shown in Figure 6 are for non-rotating conditions. Predictions are obtained at the shaft rotational speed of 750 rpm. The error bars plotted in the graphs illustrate variations of stiffness coefficients between tests with different eccentric rotor positions. Averaged data points were obtained from the different tests by the least square method. The range of scattering reaches a maximum at low preswirl conditions. The highest deviations are typically at the smallest rotor eccentricity.

Regarding the local direct stiffness, both seals show linear pressure dependence. The local direct stiffness remains negative for all operating conditions. The SSS 0.5 seal demonstrates a growth of the local direct stiffness (in absolute values) with increasing the preswirl. As compared to the SSS 0.27 seal, the SSS 0.5 seal has lower absolute levels of the local direct stiffness.

Similar to the local direct stiffness, the local cross-coupled stiffness levels of the SSS 0.5 seal are lower than for the SSS 0.27 seal and influenced obviously by the preswirl velocity. The local cross-coupled stiffness is shown in Figure 6 versus a product of the inlet pressure by the preswirl. The advantage of this presentation is that the local cross-coupled stiffness appears to be almost linear and the seals with different clearance can be directly compared. From the rotordynamic point of view, seals with higher amounts of cross-coupled stiffness tend more to instability which makes the SSS 0.5 seal to be the better choice regarding only local stiffness coefficients and ignoring disadvantages in leakage performance. For an overall evaluation, the knowledge of the damping properties is necessary which is included in the global results obtained by the dynamic experiment.
The predicted local stiffness coefficients of the SSS 0.5 seal match the experimental values quite well, with the exception of the low preswirl case for the direct stiffness and high preswirl case for the cross-coupled stiffness.

3.2 Global stiffness and damping coefficients

Additionally to the seal-induced stiffness coefficients, the global coefficients determined by the dynamic experimental procedure include the damping coefficients oriented in radial (cross-coupled damping) and tangential (direct damping) directions. Compared to the local results that only take into account aerodynamic forces acting in the cavities between the labyrinth fins, global rotordynamic coefficients concern the whole seal system. This includes areas under the labyrinth fins, as well as seal inlet and outlet regions.

Figure 7 provides the experimental global stiffness and damping coefficients for the SSS 0.5 and SSS 0.27 seals, as well as predictions for the SSS 0.5 seal. Predictions and experimental results for the SSS 0.5 seals are presented for three preswirl velocities (small, medium, and high). The global coefficients for the SSS 0.27 are shown for the preswirl velocity of about 140 m/s. The results shown in Figure 7 are obtained at the shaft rotational speed of 750 rpm.

As seen for the local results, the global direct stiffness is also negative. Radial clearance and preswirl do not have influence on the global direct stiffness for the tested operating conditions. Furthermore, both seals have a minimum in the direct stiffness curve at the inlet pressure of 3 bar.

As opposed to the global direct coefficients, the global cross-coupled stiffness strongly depends on the tangential velocity of the inflow. Both seals show linear dependence on the preswirl and inlet pressure. Compared to the SSS 0.27 seal, the SSS 0.5 seal has a slightly decreased global cross-coupled stiffness.

The direct damping is much higher in the SSS 0.27 seal. The SSS 0.5 seal demonstrates small influence of the preswirl on the direct damping. The cross-coupled damping of the SSS 0.5 seal depends strongly on the preswirl. At small preswirl, the cross-coupled damping of the SSS 0.5 seal is inversely proportional to the inlet pressure and changes its sign at about 2 bar. The cross-coupled damping of the SSS 0.27 seal is similar to the results obtained for the SSS 0.5 seal at medium preswirl.

At SSS 0.5, small preswirl levels of 100 m/s change slope and sign at cross-coupled damping to the negative. As global direct stiffness, cross-coupled damping is not affected by the clearance variation.
The predicted global stiffness and damping coefficients of the SSS 0.5 seal are also plotted in Figure 7. The global direct stiffness is in agreement with the experiments up to the inlet pressure of 3 bar. The drop in the experimental values at 3.5 and 4 bar for the low and medium preswirl is not reproduced by the predictions. The global cross-coupled stiffness is underpredicted for the high preswirl. For the global direct damping, the predictions are in good agreement with the experiments. The negative experimental values of the cross-coupled damping are not reproduced by the CFD method.

4 CONCLUSIONS

The experimental results on leakage and rotordynamic coefficients obtained on two test rigs using different measurement methods are presented for the short three-teeth-on-stator labyrinth seal with the tooth radial clearance of 0.5 mm. The experimental results for the same seal but different tooth radial clearance (0.27 mm) obtained in previous works are also shown.

Generally, the experimental data shows strong preswirl sensitivity of the cross-coupled stiffness. Direct damping is mainly influenced by the inlet pressure. Therefore, for the seals operated under high preswirl conditions, the cross-coupled stiffness grows faster with raising the pressure drop than the direct damping which increases the risk of instability problems.

The local direct stiffness depends linearly on the inlet pressure whereas the global direct stiffness decreases (in absolute values) after a minimum at 3 bar. The cross-
coupled damping is inversely proportional to the inlet pressure at the small preswirl.

The comparison between the different seal clearances shows a slightly lower level of the cross-coupled stiffness for the SSS 0.5 seal as opposed to clearly higher direct damping for the SSS 0.27 seal. The direct stiffness and cross coupled damping acting in radial direction are not affected by the clearance variation. Concerning the leakage increase of 40% caused by enlargement of clearance, the SSS 0.27 seal outperforms the SSS 0.5 sealing geometry regarding rotordynamics and leakage reduction.

The modernization of the test rigs provides more time-efficient and precise experiments. The real-time visualization and storage of experimental data simplifies the adjustment of operating parameters and keeps more information of the experiments.

Regarding the CFD method, the predictions are in reasonable agreement with the measurements. The discrepancy between the numerical results and experiments for the global direct stiffness and global cross-coupled damping indicates that an accurate prediction and measurement of the normal force is more of a challenge than of the tangential force.

5 ACKNOWLEDGEMENTS

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6 REFERENCE LIST