Présentation d'une campagne d'essais sur joints annulaires d'étanchéité

Presentation of a test series with labyrinth seals

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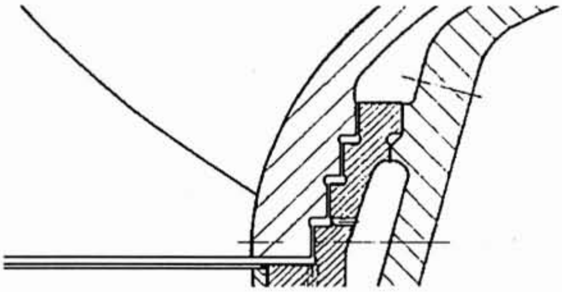
Dans une turbomachine, les joints d'étanchéité annulaires sont utilisés pour améliorer le rendement des machines en limitant les fuites de fluide. En pratique, un compromis est à trouver entre les conséquences de l'influence des joints sur le rendement de la machine d'une part et sur la stabilité dynamique de la ligne d'arbre d'autre part. Pour étudier systématiquement ces influences, une campagne d'essais a été lancée par le Professeur Bernard Chaix et réalisée à l'Ecole Polytechnique de Zürich. L'installation expérimentale a été construite comme un dispositif flexible permettant l'essai de joints de géométries différentes avec des excentricités fixes ou effectuant des précessions sur une orbite. L'accent a été mis plutôt sur l'analyse de l'écoulement que sur la reproduction des géométries industrielles des joints. Par comparaison aux joints des machines hydrauliques, les joints testés ont été réalisés avec des dimensions agrandies pour permettre une observation plus détaillée de l'écoulement. Des résultat sélectionnés des travaux des doctorants Graf, Kündig, Amoser et Spirig seront présentés.

I ■ INTRODUCTION - PURPOSE OF THE INVESTIGATIONS

The geometry of the different types of labyrinth seals generally employed in hydraulic machinery has to be - for manufacturing reasons - very simple. Fig.1 depicts the layout of one of the more sophisticated seal designs used for highly efficient pump-turbines.

The smaller a turbomachine is the larger becomes the contribution of the seal leakage loss to the overall hydraulic losses in a turbomachine. E.g. for small heating water circulation pumps the leakage flow may cause up to 50 % of the hydraulic losses. For the larger machines the leakage aspects may be of minor importance except for worn out, eroded seals and further improvement of the designs employed by experienced manufactures may bring only parts of a percent of efficiency improvement. For these machines, especially pumps, it is of far more importance that the seals are the major source of the radially acting fluid forces. In the worst case these forces may even destabilize the rotor with eventually detrimental outcome to the turbomachine.

Looking at the mentioned simplicity of geometry and remembering uncomplicated calculations of the laminar flow - which we probably all had to do in a first grade course of fluid mechanics - one would not expect major research going on on the subject of these seals. As a matter of fact there is a huge number of very recent publications from representatives of universities and industry. A recent review and summaries of actual research work are given by Childs, 1993, in his book "Turbomachinery rotor dynamics, phenomena, modeling and analysis".



1. Meridian view of a stepped labyrinth seal (taken from a drawing of Sulzer Escher Wyss).

The need for individual seal data with varying geometries and flow conditions leads to the development of more and more sophisticated flow models or numerical flow calculations. These models and calculations have as an objective the correct prediction of flow losses, forces, and torque. However, the value of these predictions can only be appreciated once the results have been validated with reliable experimental data.

For this reason Chaix (professor emeritus at the ETH-Zürich) initiated a project with the goal to increase the physical insight into the flow and its effects on forces. Furthermore, experiments were planned to provide experimental data for the validation of theoretical models and

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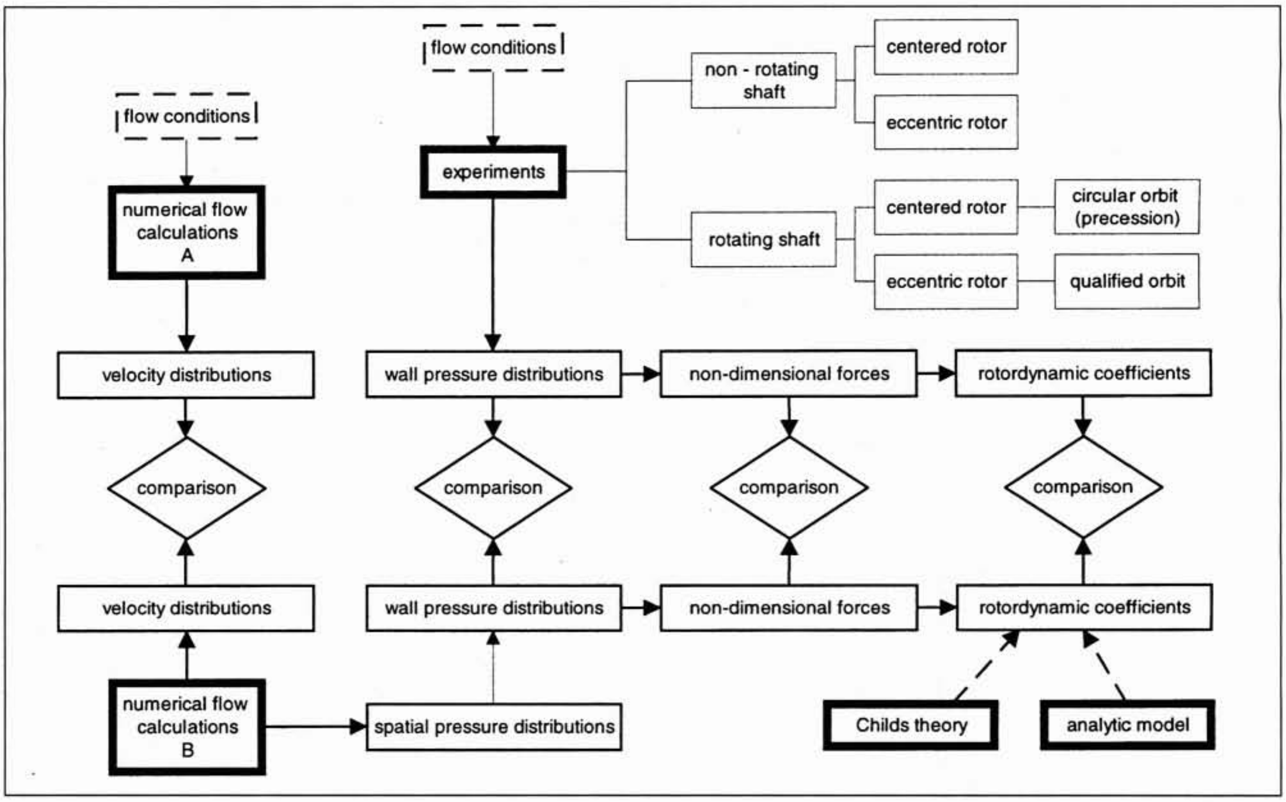


2. Seal tests and comparison with numerically calculated or analytically predicted data.

numerical codes. Under his guidance a multi-purpose test rig for investigation of seal configurations was designed.

The thought of simple laminar flow calculations immediately vanishes when considering that the rotor usually will not rotate on a fixed centric or eccentric position but, additionally, will vibrate or perform an arbitrary orbital motion around an eccentric position. This will accordingly lead to unsteady, three-dimensional flows with strong gradients due to abrupt geometrical changes within the seal configuration. Only in some special cases (circular orbit) one can find an appropriately rotating coordinate system where the problem may be considered steady ; however, the flow still is three-dimensional in nature and much dependent on upstream and also downstream flow conditions.

On this basis, systematic experiments (Fig. 2) were planned to augment insight into the effects responsible for the fluid forces. Different motions of the rotor within the surrounding seal had to be considered. In a first test series the orbital motion of the eccentricity was kept as much as possible on a circular orbit. This limitation was only released recently by Spirig, who demonstrated that rotordynamic coefficients may be determined from one single test run if a special qualified orbit is chosen [1].

For the validation of numerical codes or theoretical models it is also important to consider cases without rotation. In this case the forces act stabilizing in the sense that they tend to center the rotor (restoring forces due to the Lomakin-effect). Fig. 2 gives an overview of the experimental conditions and types of possible comparisons between locally or globally measured data and theoretically and numerically obtained data.

With respect to engineering applications the rotordynamic coefficients (global data) are most important since they form the basis for the calculations of the dynamics of the entire turbomachinery rotor. However, the detailed flow effects being the source of the forces can no more be interpreted on the basis of such coefficients. The correct modeling of forces on the other hand asks for knowledge of local flow effects and accordingly representative local data must be verified.

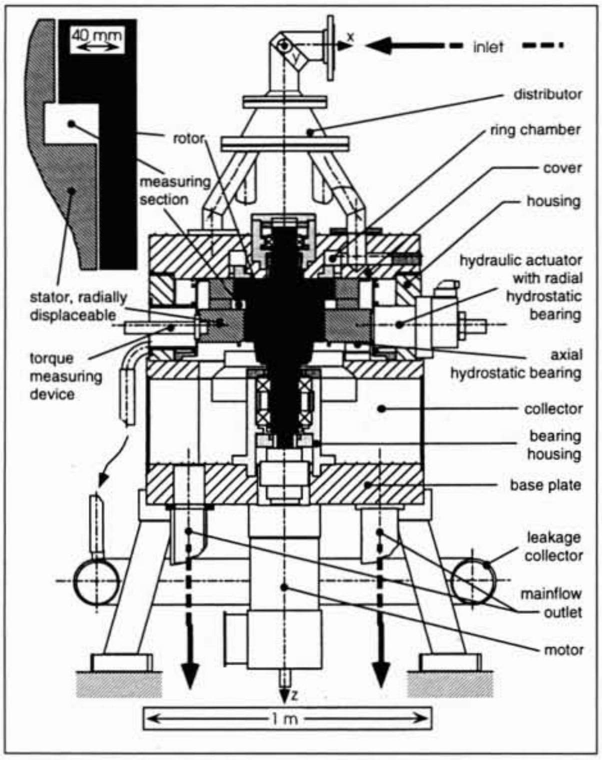
Only when the governing effects are known it is possible to formulate appropriate similarity and to transfer results from one seal configuration with given flow conditions to another configuration.

The experiments and the theoretical work of the doctorands Graf, [2], Kündig, [3], Amoser, [4] and Spirig, [5], described in the following cover the individual boxes given in the overview of Fig. 2.

II ■ TEST RIG

The construction of the labyrinth test apparatus depicted in Fig. 3 was initiated by Professor emeritus Bernard Chaix at the Swiss Federal Institute of Technology in Zürich. The installation was designed by his doctorands in close collaboration with Sulzer Hydro in Zürich. This flexibly constructed test apparatus allows investigation of enlarged versions of industrially important seal configurations. The enlargement of the seal cross section to be tested permitted more detailed flow surveys especially with respect to three-dimensionalities [6] and effects due to curvature within the seal flow. Furthermore, enlargement of the seal cross-section facilitates the accurate measurement of the rotor position and its motion. At the moment this test-rig is being transferred to the Zentralschweizerisches Technikum Luzern.

The main components of the test rig are the stator with the excitation mechanism and the rotor shaft and its driving



3. Test rig.

motor. The modular design allows the exchange of rings and according adaptation of various seal geometries. The rig is built very massively and mounted on vibration absorbers in order to prevent disturbing effects on its excited vibration.

Three hydrostatic actuators excite the stator vibrations, thus the relative displacement of the rotor to the stator. This design with strong external forces and the integrated control mechanism permits to prescribe shape and frequency of almost any harmonic displacement of the rotor eccentricity. Circular displacement orbits could be achieved with only 5 % deviation from a perfect circle. The central portion of the rig with the rings describing the investigated seal rests on axial and radial hydrostatic bearings. When floating on these bearings friction becomes negligibly small and the torque acting on the seal portion can be measured.

2.1 Technical specifications and example of an investigated seal geometry

Maximum feeding pressure	$p_{0,max} = 6.\ 10^5$ N/m²
Maximum discharge	$V_{0,max} = 0.03$ m³/s
Rotor speed	N_{max} = -3600...+3600 rpm
Axial Reynolds number	$Re_{axial} = C_m 2S_0/\nu = 5.10^4$
Circumferential Reynolds number	$Re_{tangential} = US_0/\nu = 5.\ 10^5$
Static eccentricity	$\varepsilon = E/S_0 = 0...\ 1$
Rotor displacement amplitude	$s_{max} = S/S_0 = 0...\ 0.5$
Rotor displacement frequency	$f_{max} = 5\ .\ .\ .\ 28$ Hz

2.1 Measured quantities

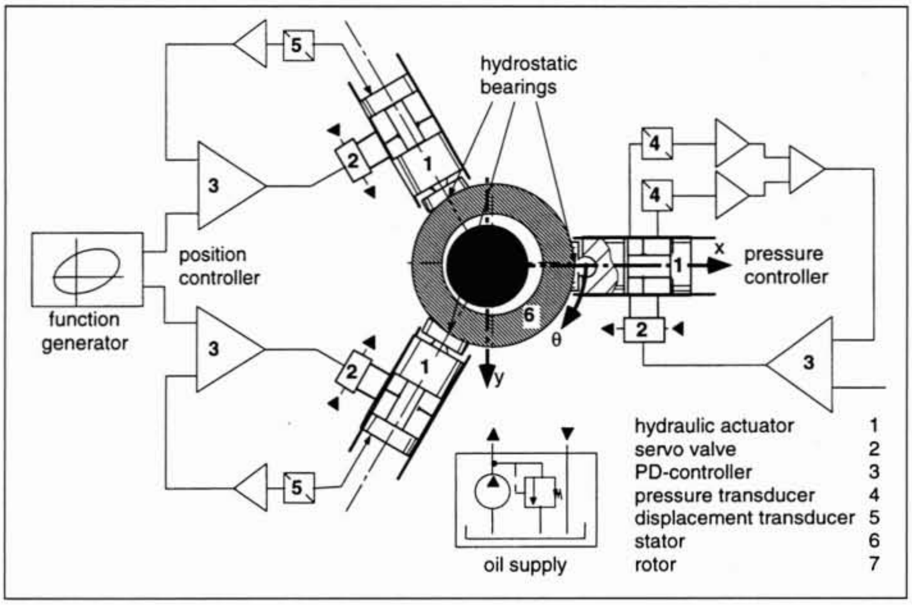
Static wall pressure distributions were measured with capacity type differential transducers mounted outside the test rig. Pressure taps had a diameter of 0.7 mm and overall measuring uncertainty of the pressure measurement was estimated to be about ±20 mbar after in situ calibration (±0.2 % of fullscale).

Discharge was measured electro-magnetically. Periodical recalibration of the transducer on site allowed to keep the relative measuring uncertainty below ±0.5 %.

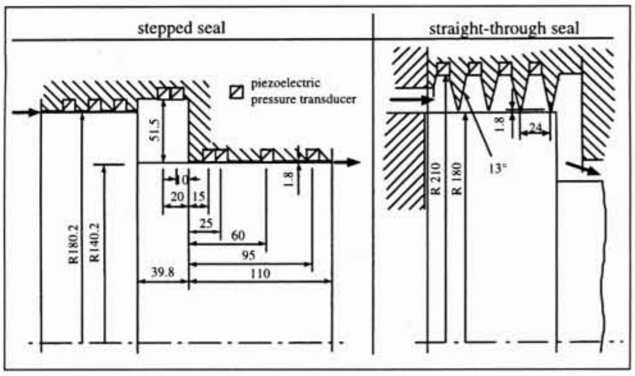
Due to the fact that the stator is floating on hydrostatic bearings the torque acting on the stator could be measured with force transducer.

The relative displacement of rotor and stator was measured by clearance measurement using eddy current transducers (uncertainty ±3 %). The angular position of the rotor could be determined with an accuracy of ±0.5 degree and the speed of rotation with an uncertainty of ±0.2 %.

Unsteady wall pressures were measured by Graf with piezoelectric (uncertainty of the measured harmonic fluctuation amplitudes ±5 %) and by Spirig with piezoresistive array transducers (uncertainty ±0.2 %).



4. Excitation mechanism with three hydraulic actuators for stator vibration and the relative rotor-stator motion.



5. Typical model seal geometry with piezoelectric pressure transducers for unsteady pressure measurement [2].

The mounted array transducers developed by Spirig are depicted in Fig. 6. Goal of his development was to accomplish not only good temporal but also good spatial resolution of the unsteady pressure measurement within the labyrinth seal. The spatial resolution of these transducers comes down to 2.4 mm allowing 16 transducers to be mounted over the labyrinth chamber height.

III ■ LEAKAGE, FRICTION, AND FORCES UPON ECCENTRIC ROTORS

The thesis of Kündig gives answers to general questions related to seals operating in a position of steady eccentricity. Various geometries of typical labyrinth seals employed for hydraulic machinery were investigated experimentally by measuring static wall pressure distributions. For a given geometry energy dissipation and fluid-elastic or stiffness coefficients were determined as a function of discharge and rotor speed from measurements with varying fixed eccentricities. These coefficients were integrated from the measured axial and circumferential pressure distributions along the stator walls. Special effort was put on the separation of the loss coefficients caused by the heavily disturbed flow at the inlet portion to the plain seals, the contributions of the chambers, and the developed flow within the plain seals. Kündig, [3], analyzed the overall power losses and compared data of the plain seal portions with the data obtained by other authors in order to verify the similarity laws.

IV ■ CIRCULAR ORBIT OF THE ROTOR ECCENTRICITY AND THE ROTORDYNAMIC COEFFICIENT MATRIX

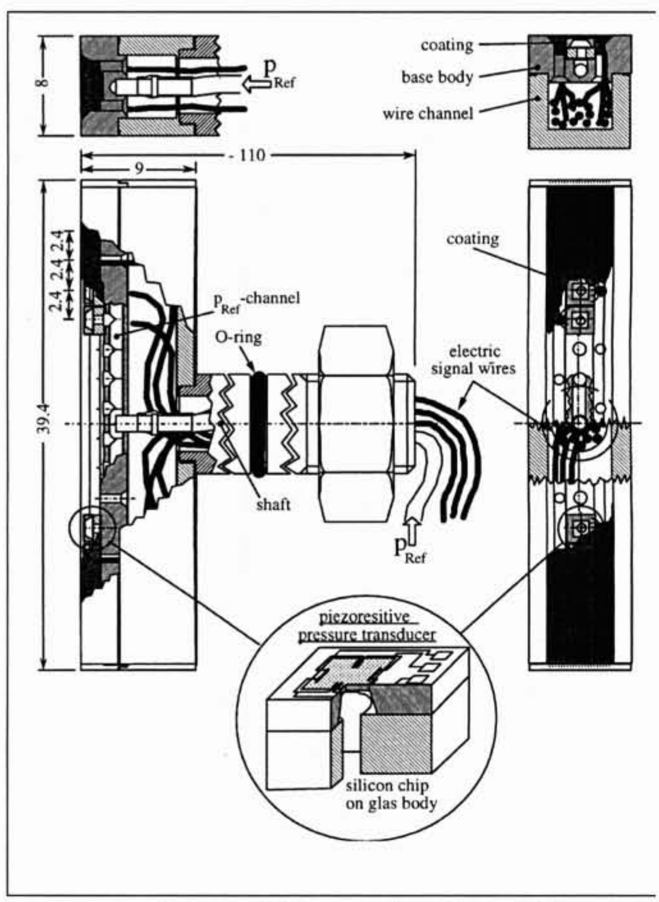
In case of an orbital motion of the rotor relative to the stator the local pressures on a fixed position on the rotor or stator will be time-dependent. Graf, [2], measured the fluctuating component of these pressures on selected positions with piezo-electric transducers (Fig. 5). In order to improve the spatial resolution Spirig, [5], developed a measuring system with narrowly packed piezo-resistive transducers, which also allow measurement of the steady pressures in addition to the fluctuation component (Fig. 6).

Graf determined the unsteady forces acting upon the rotor from integration of the measured pressure fluctuations varying the rotational frequency of the eccentricity orbit independently of the rotor speed. He tuned the test rig in order to achieve the best possible circular (cylindrical) orbits. Furthermore, with small amplitudes of eccentricity around a centered position he could assume an isotropic force-displacement model. This model assumes a linear relation between force and displacement, velocity, and acceleration. It is described by the stiffness, the damping, and the inertia coefficient matrices in the linear differential equation below :

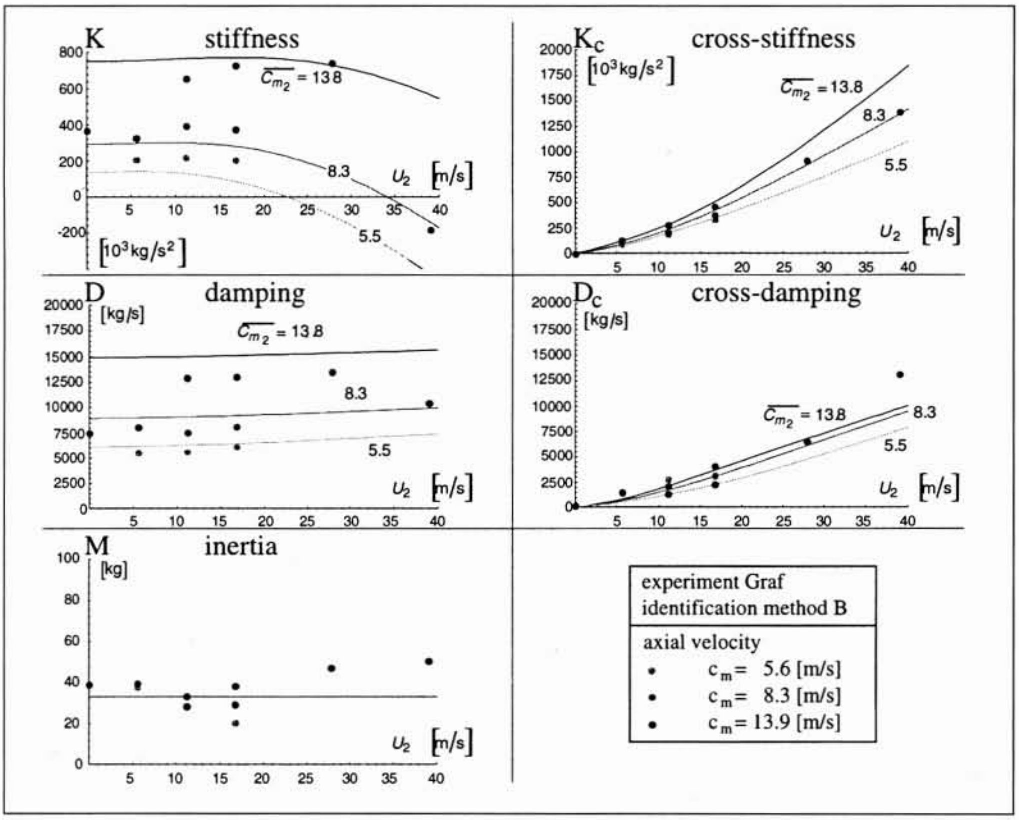
$$\begin{Bmatrix} -F_x \\ -F_y \end{Bmatrix} = \begin{bmatrix} K & K_c \\ -K_c & K \end{bmatrix} \begin{Bmatrix} x \\ y \end{Bmatrix} + \begin{bmatrix} D & D_c \\ -D_c & D \end{bmatrix} \begin{Bmatrix} \dot{x} \\ \dot{y} \end{Bmatrix} + \begin{bmatrix} M & M_c \\ M_c & M \end{bmatrix} \begin{Bmatrix} \ddot{x} \\ \ddot{y} \end{Bmatrix}$$

In order to identify the rotor dynamic coefficients Graf performed a series of measurements varying the orbit motion frequency. For the plain seal sections he employed two different methods of identification of rotordynamic coefficients [7]. Further test parameters were the discharge through the seal, the rotor speed, and the inlet swirl entering the first plain seal of the test section.

The results of Graf's study confirm that the cross coupled terms of the inertia matrix for the tested seal configurations are small and can be neglected. The damping and inertia coefficients show to be independent of the rotor speed as well as of the inlet swirl. Thus only the stiffness coefficients vary weakly with the circumferential velocity of the fluid. In contrast, the cross coupled terms of stiffness and damping increase with growing circumferential velocity and inlet swirl. All coefficients increase with increasing axial velocity with the exception of the cross coupled damping and also the inertia coefficients which are independent of through-flow.



6. Piezoresistive array transducers [5].



7. Comparison of experimentally determined rotordynamic coefficients [2] with the analytic approximation (full and dotted lines) by Amoser, 1993.

Amoser, 1993, performed detailed numerical computation of the three-dimensional flow in a stepped labyrinth seal and compared results with experimental data. In his calculations he put great effort into calculating the ensemble of the seal meaning calculations of a plain seal followed by a chamber and a second plain seal. In this way he could avoid making insecure estimates of inlet and outlet effects to the chamber and the second plain seal.

Analyzing numerical 3D flow data and experimentally obtained data he succeeded in formulating an analytical approach to assess the dynamic forces rising from thin plain seals with an eccentricity performing a circular orbit. In his model he assumes a superposition of three effects : the stiffness force (Lomakin-force) that is restoring the rotor, the inertia force that is divergent, and the force due to viscosity (as in journal bearings) that is perpendicular to the eccentricity. He interpolates the influence of geometry and flow conditions (e.g. inlet swirl) between known special cases. With this approach he is able to model broad cases of industrially employed seal designs. He verifies his results by comparison with the theoretical calculations of Childs and with measurements published in literature. Fig. 7 displays a comparison of his analytic approach with measurements of Graf.

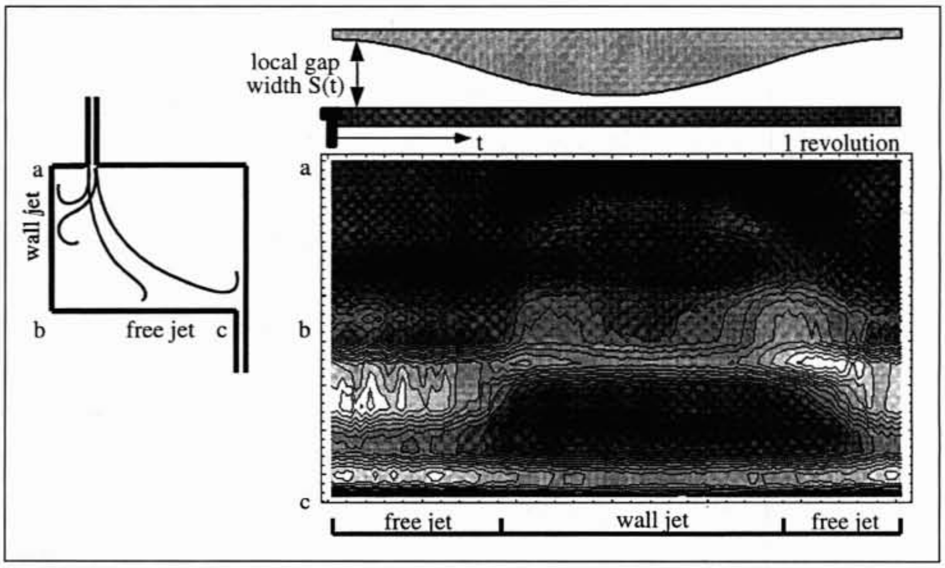
Using a representative reference velocity he succeeds in finding a non-dimensional formulation for Eq. (1). Employing the non-dimensional representation of the equation of motion, forces of model tests with different physical and geometrical conditions may be conferred and their influence on prototype runners assessed.

Since the flow exiting from the chamber of the labyrinth seal into the following plain seal provides inlet conditions which have an important or even dominating effect whether the seal is considered to add stabilization to the rotor or not, it is most important to know more about the three-dimensional flow effects occurring in the seal (Staubli, Amoser, 1991). The thesis of Spirig, [5], focuses with emphasis on this aspect. With his spatially well resolved pressure measurements he is able to detect a flip-flop characteristic of the jet exiting from a plain seal section into a chamber as depicted in Fig. 8.

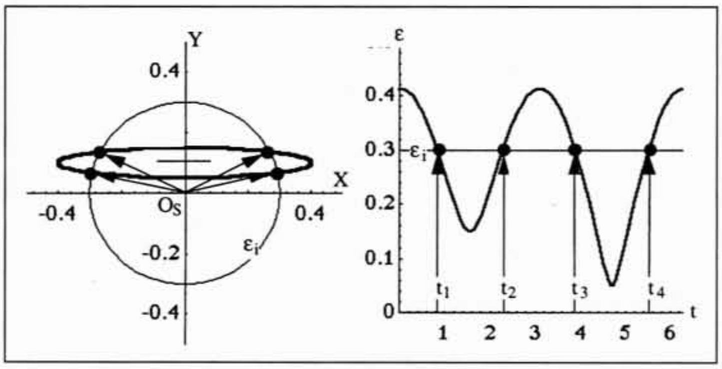
V ■ NON-CIRCULAR, ECCENTRIC ORBITS AND IDENTIFICATION OF ROTOR-DYNAMIC COEFFICIENTS

Analyzing the spatial and temporal variations of pressure distributions Spirig, [5], came to the conclusion that it is possible to determine all rotordynamic force coefficients from one single test run without variation of rotor or orbit speed. Necessary condition are multiple circumferentially and axially distributed measurements of unsteady pressure. The procedure leading to the coefficients bases on a two-dimensional (time and space) Fourier transform.

In order to determine the coefficient for one specific magnitude of the rotor eccentricity the orbit has to pass this eccentricity four times per cycle, thus specially qualified orbit motions are required. An example of such a qualified orbit is depicted in Fig. 9. Since this procedure allows evaluation of the rotordynamic coefficients as a function of displacement ε linearity has no more to be assumed. This is



8. Unsteady pressure distribution Spirig, [5] indicating a flip-flop from wall jet to free jet and vice versa in the labyrinth chamber.



9. An example of a qualified rotor/stator displacement with an eccentric eliptical orbit and harmonic variations of the non-dimensional eccentricity e(t).

what is of special importance in situations with orbits of larger amplitudes of the rotor eccentricity $\varepsilon(t)$.

VI ■ CONCLUSION

In a series of successive research work [2, 3, 4, 5], insight into the fluid dynamic phenomena responsible for energy dissipation and forces could be considerably enhanced. These findings showed to provide a sound basis for improvement of force prediction models and non-dimensional coefficients allowing to scale the forces to a specific geometry and flow condition.

Abandoning the assumption of linearity Spirig proposes a new procedure for identification of rotordynamic coefficients from one single test run using multiple circumferential pressure measurements. A necessary condition for his procedure are qualified (non-circular and eccentric) orbits of the rotor eccentricity.

Although knowledge on labyrinth flows was improved during the last decade there are still many details to be enlightened in these complex, three-dimensional, unsteady flows with extreme gradients due to abrupt geometrical changes. Especially with respect to numerical calculation these flows will be a challenge for future research.

References

[1] Spirig, M., Staubli, T., 1997, "Identification of non-linear rotor dynamic coefficients using multiple circumferential pressure measurements", ASME Conference on Flow Induced Vibration, Dallas.

[2] Graf, K., 1991, "Spaltströmungsbedingte Kräfte an berührungslosen Dichtungen von hydraulischen und thermischen Turbomaschinen", Dissertation ETH Nr. 9319, Zürich.

[3] Kündig, P., 1993, "Gestufte Labyrinthdichtungen hydraulischer Maschinen, Experimentelle Untersuchung der Leckage, der Reibung und der stationären Kräfte", Dissertation ETH Nr. 10366, Zürich.

[4] Amoser, M., 1995, "Strömungsfelder und Radialkräfte in Labyrinthdichtungen hydraulischer Strömungsmaschinen", Dissertation ETH Nr. 11150, Zürich.

[5] Spirig, M., 1997, "Instationäre Druckfeldmessung zur Ermittlung nichtlinearer rotordynamischer Koeffizienten hydraulischer Labyrinthdichtungen", Dissertation ETH to be published, Zürich.

[6] Amoser, M., Staubli, T., 1991, "Three-dimensional flow phenomena in labyrinth seals", 9th conference on Fluid Machinery, Budapest.

[7] Graf, K., Staubli, T., 1990, "Comparison of two identification models for rotordynamic coefficients of labyrinth seals", Interfluid 1st International Congress on Fluid Handling Systems, Essen.

[8] Amoser, M., Staubli, T., 1996, "Analytische Approximation rotordynamischer Koeffizienten von Spaltdichtungen", Pumpentagung Karlsruhe.

[9] Childs, D. W., 1993, "Turbomachinery rotor dynamics, phenomena, modeling and analysis", J. Wiley & Sons, New York.