PARAMETRIC STUDY ON THE AEROELASTIC STABILITY OF ROTOR SEALS

QINGYUAN ZHUANG

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Qingyuan Zhuang

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Department of Energy Technology
Division of Heat and Power Technology
Royal Institute of Technology
100 44 Stockholm, Sweden
ABSTRACT

Labyrinth seals are widely used in rotating machinery and have been shown to experience aeroelastic instabilities. The rapid development of computational fluid dynamics now provides a high fidelity approach for predicting the aeroelastic behavior of labyrinth seals in three dimension and exhibits great potential within industrial application, especially during the detailed design stages. In the current publication a time-marching unsteady Reynolds-averaged Navier-Stokes solver was employed to study the various historically identified parameters that have essential influence on the stability of labyrinth seals. Advances in understanding of the related aeroelastic (flutter) phenomenon were achieved based on extensive yet economical numerical analysis of a simplified seal model. Further, application of the same methodology to several realistic gas turbine labyrinth seal designs confirmed the perceived knowledge and received agreements from experimental indications. Abbott’s criteria in describing the labyrinth seal aeroelastic behaviors were reaffirmed and further developed.
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NOMENCLATURE

Latin Symbols

\( c \)  
speed of sound  
\( n \)  
nodal diameter number  
\( v_{\text{slip}} \)  
slip velocity  
\( \omega_e \)  
mechanical (natural) frequency  
\( \omega_{ac} \)  
acoustic frequency

Greek Symbols

\( \delta \)  
logarithmic decrement  
\( \eta \)  
frequency ratio, \( \eta = \omega_e / \omega_{ac} \)

Subscripts

\( \text{ac} \)  
aoustic  
\( \text{cyc} \)  
per cycle

Abbreviations

AIAA  
American Institute of Aeronautics and Astronautics  
BC  
Boundary Condition  
CFD  
Computational Fluid Dynamics  
FE  
Finite Element  
FSI  
Fluid-Structure Interaction  
HCF  
High Cycle Fatigue  
HPS  
High Pressure Support  
LPS  
Low Pressure Support  
ND  
Nodal Diameter  
TWM  
Traveling Wave Mode  
URANS  
Unsteady Reynolds-Averaged Navier-Stokes
1 INTRODUCTION

1.1 Background

Labyrinth seals are commonly found in turbomachinery. They are comprised of one rotating and one stationary component. Together with the annular cavities which are formed by the coaxial rotor-stator cylindrical surfaces and a series of sealing fins, they keep the high and low pressure cavities separated. The sealing fins, or so-called teeth, grow radially outward from the rotor cylinder surface towards the stator (sometimes in a reverse direction when the fins are attached to the stator) and maintain a certain clearance between the two. A typical labyrinth seal configuration is shown in Figure 1.1. The members directly opposite to the fin tips (or knife edges) are usually coated with abradable and soft materials such as honeycomb structures. They are in place to accommodate the variable tip clearance during engine operation, especially when the two parts come into direct contact.

Labyrinth seals are primarily designed to isolate high and low pressure cavities and to serve as an interface between the rotor and stator. The sealing function is achieved by forcing the air flow to go through a series of fin tips, in a similar manner as throttled by orifices, and the kinetic energy is therefore dissipated. Being a sealing device and when engine performance is meeting ever more stringent requirements, it is of paramount importance that leakage flow is reduced to its minimum.

Certain seal designs have experienced aeroelastic instabilities dating back to the 1960s. The underlying flutter mechanism is still not as well understood as blade flutter yet the consequent HCF failures are just as catastrophic. Furthermore, the desire for lighter and more efficient gas turbines is now making the design of such structures even more challenging.
1.2 Literature Review

A brief review of labyrinth seal research is presented in this section. Core concepts are explained in detail in Section 1.3. The failure descriptions and investigations related to existing literature are further elaborated in section 6.1 for a better discussion together with the results obtained from the current analysis.

Alford (1964) first addressed the problem of labyrinth seal flexural vibration and reported that labyrinth seals had encountered both resonance and self-excited vibrations, which eventually led to fatigue failures. For the non-resonance failures Alford noticed the important effect of seal support location on stability. Attempts were made to describe how the support location, either at flow entrance (high pressure) side or discharge (low pressure) side (Figure 1.2), could affect the modulation of the flow during seal vibration, leading to unsteady pressure perturbations that could eventually contribute to instability. The research for seal flutter mechanism was not continued in Alford’s subsequent two papers (1967 and 1975) yet mechanical damping solutions were provided.

![Figure 1.2 LPS & HPS configurations; adapted from Alford (1975)](image)

Ehrich (1968) developed a 1D model based on Alford's hypothesis. One stability parameter and its accompanying criteria are derived according to the geometric, structural and flow information of the cavity element (Figure 1.4). The analysis later showed deviations from component testing.
Lewis et al (1978) investigated the same problem during the development of F100 engines. Through extensive engine tests the instability was finally attributed to the tight clearance of the first (upstream) knife edge and could be eliminated by adjusting it. An analytical model that includes the circumferential dimension is proposed in detail yet the instability in the failed design could not be predicted by the model.

Abbott (1980) developed a similar analytical model as Lewis et al. The model was extended from Ehrich’s and accounted for the circumferential pressure variation within the inter-fin cavities. From experiments as well as analytical analysis, Abbott made a coherent and most interesting discovery that the mechanical frequency of the seal, or the rigidity, affects the stability characteristic in combination with the support location. Results from the analytical computing program are shown in
Figure 1.4. The notion of a frequency ratio, essentially a comparison between the mechanical frequency of the seal structure and the acoustic wave frequency in the annular space, proved to be a useful parameter to describe and characterize the stability curve. Approximately when the frequency ratio equals to one, the stability of seals as described by aerodynamic damping for both high pressure support (HPS) and low pressure support (LPS) cases would switch sign.

Srinivasan et al (1984) took the existing analytical analyses to a further step and included in the model the influence from both the stator and rotor, i.e. both members are considered flexible and vibrate in traveling wave modes with diametral nodes (previously only the rotor member is included). The effect of swirl flow velocity within the annular cavity is also accounted for, which means that forward and backward traveling waves of the structures would interact differently with the flow and thus resulting in different aeroelastic behaviors. A parametric study of the rotor frequency for an LPS seal shows a comparable stability curve with Abbott’s results (Figure 1.5), even though it does not share the peak feature found in Abbott’s when frequency ratio nears one.

![Figure 1.5 Influence of rotor rigidity on seal stability; Srinivasan et al (1984)](image)

Phibel et al (2009) for the first time applied CFD techniques to labyrinth seal instability problems. A coupled fluid-structure code based on non-linear time accurate unsteady Reynolds-averaged Navier-Stokes equation is employed to determine the damping of the seal. A phase-shifted periodic boundary condition could be applied so that only a small sector of the complete annulus needs to be modeled in order to save computational cost. The configuration under study is a simplified model adapted from a large-diameter aero engine turbine labyrinth seal disk. A streamline plot of the studied domain and the Mach number contour could be seen in Figure 1.6. In essence, the influence of the seal natural frequencies and the support location are studied. The acquired stability curve (Figure 1.7) of a low pressure support configuration agrees well with Abbott’s criteria.
Di Mare et al (2010) furthered the study of Phibel et al and applied the same methods to what appears to be two variant configurations (Figure 1.8) of the same large-diameter aero engine labyrinth seal, but with detailed and realistic upstream and downstream cavity geometry. The detailed stability (aerodynamic work per cycle) contributions from various parts of the seal surface (Figure 1.10), especially from the inter-fin cavities, do not appear readily explainable. Perhaps a better understanding would be gained if the work per cycle is not presented in percentage but in absolute values, and contributions from each of the inter-fin cavities are further separated. The final damping curve for the seal disk at different ND TWMs is presented in Figure 1.9, which renders good prediction as compared with testing.
So far the underlying mechanism of the labyrinth seal flutter problem is not well understood, or at least not found in open literature. However, the continuation of the research is promising based on the available knowledge in terms of identified stability parameters, stability criteria, analytical models and experimental data.
1.3 Fundamental Concepts

1.3.1 Flutter Assessment Stability Parameter

A structure moving in fluid flow would result in pressure perturbations in the fluid surrounding it. According to Verdon (1987) small-amplitude harmonic oscillation of the structure would lead to harmonic unsteady pressures oscillating about a steady mean pressure field. These unsteady forces would in turn act on the structure and exert either positive or negative work, which describes the fluid-structure system as either unstable or stable respectively.

Following Verdon’s derivation, the aerodynamic work done on the structure per cycle of oscillation can be expressed as:

\[
W_{\text{cyc}} = \pi \cdot \Im \{ Q \} = \pi \cdot \Im \{ u^* F \} = \pi \cdot \sum_{i=1}^{n_a} [\Re \{ u_i \} \Im \{ F_i \} - \Im \{ u_i \} \Re \{ F_i \}]
\]  \hspace{1cm} (1.1)

where \( Q \) is the modal force, or generalized force, essentially a projection of the unsteady force vector \( F = \{ F_i \} \) onto the specific mode shape of the structure \( u = \{ u_i \} \). \( n_a \) is the total number of discretized grid points on the surface of the structure. \( \Re \) and \( \Im \) denote the real and imaginary parts respectively and * represents the complex conjugate transpose. Knowing the amplitude of the unsteady force and mode shape deflection and their respective phase lag, the aerodynamic work per cycle can be equally expressed as:

\[
W_{\text{cyc}} = \pi \cdot \sum_{i=1}^{n_a} \left[ |u_{ix}| |F_{ix}| \sin(\phi_{ix} - \mu_{ix}) + |u_{iy}| |F_{iy}| \sin(\phi_{iy} - \mu_{iy}) + |u_{iz}| |F_{iz}| \sin(\phi_{iz} - \mu_{iz}) \right]
\]  \hspace{1cm} (1.2)
where \( |u_x| \) is the magnitude of nodal displacement in the \( x \) direction and \( |F_x| \) the nodal force in the same axis. \( \phi_{F_x \rightarrow u_x} \) denotes the phase difference between the nodal force and displacement, i.e. how much the excitation leads the response.

To compare among different cases, the \( W_{\text{cyc}} \) could be further normalized by \( \pi \) and the maximum oscillation amplitude of the structure \( |u_{\max}| \), and thus arriving at the flutter stability parameter (normalized aerodynamic work per cycle):

\[
\Xi = \frac{W_{\text{cyc}}}{\pi \cdot |u_{\max}|} \quad (1.3)
\]

### 1.3.2 Determination of Logarithmic Decrement

The assessment of flutter could also be done by calculating damping, e.g., in terms of logarithmic decrement (LogDec, \( \delta \)). With such a single scalar, the results are easily condensed and could be readily compared across different nodal diameters or inter-blade phase angles (IBPAs) for a tuned bladed disk, blisk, or in case of labyrinth seal flutter, simply a disk. As a damping measure, the sign of LogDec characterizes the stability of the flow in the opposite manner compared to aerodynamic work per cycle, such that a negative LogDec value denotes instability (vibration amplifying with time) when negative \( W_{\text{cyc}} \) refers to stable flow, and vice versa. Furthermore, the magnitude of LogDec provides direct information on the severity of flutter in relation to material and friction damping effects etc. Logarithmic decrement is defined as:

\[
\delta = \ln \left[ \frac{x_i}{x_{i+1}} \right] = \frac{2\pi\zeta}{\sqrt{1-\zeta^2}} \approx 2\pi\zeta \quad (1.4)
\]

where \( \zeta \) is the damping ratio \( c / c_{cr} \), i.e., actual damping over critical damping. \( x_i \) represents the amplitude of the \( i^{th} \) vibration cycle. It is noted that for small levels of damping, \( \sqrt{1-\zeta^2} \approx 1 \), thus making \( 2\pi\zeta \) a very good approximation of the exact logarithmic decrement.

It is shown by Carta (1988) that the logarithmic decrement can be computed from the aerodynamic work per cycle \( W_{\text{cyc}} \) exerted on the structure and the averaged kinetic energy \( \bar{K}_E \) of the system during the same oscillation cycle:

\[
\delta = \frac{W_{\text{cyc}}}{4\bar{K}_E} \quad (1.5)
\]

Assuming a linear system and regarding the kinetic energy of the system as oscillating harmonically about \( \bar{K}_E \) between 0 and \( K_{E,\max} = 2\bar{K}_E \), Equation 1.5 could be rewritten as:
The calculation of aerodynamic work per cycle is already elaborated in Equation 1.2. The estimation of the maximum kinetic energy for a discretized finite element model could be expressed as:

\[ K_{E,\text{max}} \approx \frac{1}{2} \sum_{i=1}^{n} m_i |v_i|^2 = \frac{1}{2} \rho \sum_{i=1}^{n} V_i |v_i|^2 \]  

(1.7)

where \( m_i \) is the local mass, \( \rho \) the density and \( V_i \) the local grid volume. \( |v_i| \) is the local velocity amplitude and \( n \) is the total number of mesh points constructing the structure. Considering a harmonic oscillation of all the nodes the local velocity could be replaced by \( |v_i| = \omega_c |u_i| \), with \( \omega_c \) being the vibration frequency and \( |u_i| \) the nodal displacement amplitude:

\[ K_{E,\text{max}} \approx \frac{1}{2} \rho \omega_c^2 \sum_{i=1}^{n} V_i |u_i|^2 \]  

(1.8)

or expressed in finite element matrix form:

\[ K_{E,\text{max}} = \frac{1}{2} \tilde{\mathbf{u}}^* \mathbf{M} \tilde{\mathbf{u}} = \frac{1}{2} \omega_c^2 \mathbf{u}^* \mathbf{M} \mathbf{u} \]  

(1.9)

In the above equation \( \tilde{\mathbf{u}} \) and \( \mathbf{u} \) refer to the nodal velocity and displacement vectors of the mode shape. \( \mathbf{M} \) is the mass matrix.

Normally during flutter CFD calculations the original FE mode shape \( \psi \) is scaled in order to qualify the linear flutter assumption. Replacing \( \mathbf{u} \) with \( q^* \cdot \psi \) where \( q^* \) denotes a scaling factor, Equation 1.9 becomes:

\[ K_{E,\text{max}} = \frac{1}{2} \omega_c^2 (q^*)^2 \cdot \psi^* \mathbf{M} \psi \]  

(1.10)

The term \( \psi^* \mathbf{M} \psi \) is recognized as the modal (generalized) mass \( \hat{m} \) of the respective mode. In practice most FE solver would output a normalized mode shape vector since mode shape is arbitrary in nature. Most commonly the normalization is made such that either the maximum mode shape displacement becomes one or the modal mass \( \hat{m} = 1 \). In the latter case, the estimation of maximum kinetic energy finally comes down to:

\[ K_{E,\text{max}} = \frac{1}{2} \omega_c^2 (q^*)^2 \]  

(1.11)
1.3.3 Traveling Wave Mode

The vibration of a disk-like structure could either form a standing wave or a traveling wave. A standing wave can actually be interpreted as two traveling waves with the same frequency but rotating in opposite directions. The one which is rotating in the same direction as the rotor is called forward traveling wave and the other called backward traveling wave. Different wave modes are characterized by the number of nodal diameters and/or nodal rings (circles), if there are any. For structures like labyrinth seals, the most noticeable behavior when they are experiencing flexural vibration in TWM is the difference in deflection magnitude along the circumferential direction. Typical disk modes with diametral nodes are shown in Figure 1.11.

![Figure 1.11 Typical disk TWM flexural vibration modes; Alford (1975)](image_url)

Aeroelastic instabilities found in labyrinth seals all indicated that traveling waves are the culprit. During realistic operations, the forward and backward TWMs that are identical in theory (except for their traveling directions) would experience different damping due to the rotation of the rotor disk itself and the swirling flow within the inter-fin cavities. Thus after certain amount of time one of the TWMs would be pronounced while the other dissipated. The TWM motion and its influence on the seal flutter are further explained in Section 5.1.

1.3.4 Inter-Fin Cavity Acoustic Frequency

Another important concept for the labyrinth seal flutter problem is the so-called inter-fin cavity acoustic frequency. It was first proposed by Abbott (1980) and defined as:

\[ \omega_{ac} = \frac{nc}{2\pi r} \]  

(1.12)

where \( n \) denotes the number of circumferential waves or nodal diameters, \( c \) is the speed of sound and \( r \) is the inter-fin cavity mean radius.

The acoustic frequency describes the number of “acoustic waves” one (standing on the rotor) experiences in a second. Such an “acoustic wave” is assumed to travel in the same direction as the mechanical TWM in the structure. To account for the tangential velocity of the swirling air flow inside the inter-fin cavities due to rotation, the expression is modified by di Mare (2010) as:
In Equation 1.13, \( v_{\text{slip}} \) is the slip velocity of the flow (i.e. the difference in velocity between the rotating structure and the fluid). The plus sign denotes a backward TWM and a minus sign forward TWM.

The acoustic frequency later proves to be very useful when characterizing the stability of labyrinth seals, especially in relation to the mechanical frequency. The definition of a frequency ratio simply serves as a dimensionless comparison between the two frequencies:

\[
\eta = \frac{\omega}{\omega_{\text{ac}}}
\] (1.14)

When the above parameter approaches one, there is however no resonance found in the inter-fin cavities as could be seen both in the literatures as well as in the current analysis. Nevertheless, the stability of the seal is undergoing a drastic change when the mechanical frequency is crossing the boundary of the “acoustic frequency”.
2 OBJECTIVES

The research of labyrinth seal aeroelastic instability started in the 1960s and a series of milestone papers have been published. Several key parameters are proven to play an important role in the perceived instability. These parameters, including the support side of the seal, the mechanical frequency, nodal diameters and so forth, have been studied based on analytical models, experiments and most recently CFD techniques. Nevertheless, the underlying flutter mechanism is still not fully understood and little reference exists in open literature on the detailed effects of those parameters. Consistent and comprehensive guidelines are also favored to ensure more reliable labyrinth seal designs.

The current thesis aims at investigating in detail the historically identified stability parameters. On top of that, it addresses the industrial applicability of time-accurate CFD methods to the aeromechanical design of labyrinth seals, both in terms of computational cost and accuracy. The ultimate goal is to gain a deeper and broader insight into the problem by mapping the flutter characteristics in various scenarios, validating the numerical results for a set of realistic designs, and applying Abbott's criteria in describing labyrinth seal aeroelastic phenomenon in general.
3 METHOD OF ATTACK

In essence a time-marching unsteady Reynolds-averaged Navier-Stokes solver was employed for the fluid-structure interaction simulation. Due to the relative lack of knowledge for labyrinth seal aeroelastic problems, the calculations were first done on a simplified test model for a comprehensive parametric study. Afterwards the exact same method was applied to the case study of several realistic industrial gas turbine sealing designs.

For each case, the aeromechanical analysis was applied and streamlined as shown in Figure 3.1. First of all a cyclic symmetry modal analysis was performed using ANSYS Mechanical, from which the complex mode shape vectors of the prescribed TWMs were exported. The CFD mesh was created using ICEM and then assembled in CFX, reading in the complex mode shape data for mesh morphing along with other boundary conditions. Finally, the aerodynamic work per cycle or damping (LogDec) was calculated with Matlab scripts to determine the overall stability of the seal structures.

![Figure 3.1 Labyrinth seal flutter analysis flow chart](image-url)
4 NUMERICAL TECHNIQUES

The numerical analysis is based on decoupled fluid-structure interaction and mainly consists of two parts, namely the cyclic symmetry modal analysis in ANSYS APDL Mechanical (included in ANSYS Workbench platform) and CFD flutter calculation (FSI) in ANSYS CFX.

4.1 Structural Model

Considering the axisymmetric feature of the analyzed component, i.e. a complete rotor disk, a cyclic symmetry model was used for efficient simulations. Firstly a one-degree sector model is prepared, with defined upper and lower boundaries in the circumferential direction as required by such cyclic symmetry analysis. To this end, ANSYS Workbench was found to provide a streamlined and integrated design process within one package, which includes CAD modeling, pre-processing (meshing) and post-processing. After solutions are generated, it is convenient to select the value of the specific harmonic index (corresponding to the respective nodal diameters or nodal rings/circles) and get the related modal displacements.

4.2 Fluid Model

Unlike the cyclic symmetry structural modal analysis with phase-shifted boundaries, a time-accurate axisymmetric fluid model featuring periodically alternating patterns in the circumferential dimension (such as TWMs) requires a periodic boundary condition, i.e. the sector angle is large enough to accommodate at least one complete wave length. In case of a 4ND mode for instance, a 90-degree sector must be made. Similarly for a 5ND mode the model must be 72 degrees or its multiples in sector angle and so forth.

Structured hexahedral meshes are created for all cases. For transient CFD calculations, especially with large sector models, a structured mesh is extremely advantageous regarding memory and CPU usage.

4.3 Simple Model Parametric Study

4.3.1 Numerical Setup

The simple test model is designed to maintain the essential features of a labyrinth seal while having a clean geometry. The seal is cantilevered from one side and comes with three fins, forming two inter-fin cavities. The flow channel is almost mirrored upside down, the thinking behind which is that by simply switching the inlet and outlet assignment a low pressure support configuration could be conveniently changed to high pressure support, with all other setups unaltered. This makes it an optimal test case to study the effects of support location of seal as proposed by both Alford and Abbott. A one-degree mesh is presented in Figure 4.1. Within the domain the entire right hand side is the stator member and the left side the rotor, separated by the inlet and outlet.
Below are the boundary conditions and some key parameters for the baseline case.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Inlet pressure</td>
<td>4 bars</td>
</tr>
<tr>
<td>Inlet static temperature</td>
<td>420 °C</td>
</tr>
<tr>
<td>Outlet pressure</td>
<td>1 bar</td>
</tr>
<tr>
<td>Rotational speed</td>
<td>6600 rpm</td>
</tr>
<tr>
<td>TWM direction</td>
<td>Forward traveling wave</td>
</tr>
<tr>
<td>Mechanical frequency</td>
<td>500 Hz</td>
</tr>
<tr>
<td>Nodal diameter</td>
<td>4</td>
</tr>
<tr>
<td>Cavity mean radius</td>
<td>0.43725 m</td>
</tr>
<tr>
<td>Seal support side</td>
<td>Low pressure support</td>
</tr>
</tbody>
</table>

Table 1 Baseline case BCs and parameters

A number of parameters are varied to check their effects on stability. To change the support side of the seal, inlet and outlet positions are switched as mentioned before. The complex mode shape displacements are initialized as profile data into CFX. For the \( i \)th node the coordinate is \((x_i, y_i, z_i)\). Its real and imaginary part of the displacement values in the \( x \) axis are \( \mathbb{R}\{u_{ix}\} \) and \( \mathbb{I}\{u_{ix}\} \) respectively. The real time actual (physical) displacement could then be expressed as:

\[
  u_{ix} = \mathbb{R}\{u_{ix}\} \cdot \cos(\omega f) - \mathbb{I}\{u_{ix}\} \cdot \sin(\omega f) \quad (4.1)
\]
where $t$ is the time variable in CFX and $\omega_e$ is the aforementioned mechanical frequency of the seal. $u_y$ and $u_x$ can be expressed in a similar way. The displacements altogether dictate the movement of the mesh at each and every time step. The mechanical frequency of the seal can thus be tuned by changing the value of $\omega_e$. The effect of pressure ratio across the seal can be modified by changing the inlet pressure.

Different TWMs, i.e. the number of nodal diameters and traveling directions, are directly controlled by the data as extracted from the modal analysis. The effect of the maximum deflection could be experimented by adding a scaling factor to the mode shape either within CFX or before the import of data.

### 4.3.2 Convergence Studies

First a grid convergence study is conducted among four mesh configurations. The mass flow rate normalized by the 3rd configuration (baseline case, marked in red) is presented in Figure 4.2. As the number of mesh points increase the mass flow rate is converging. Even the coarsest mesh has merely a 0.5% deviation of mass flow rate away from the baseline case.

![Figure 4.2 Simple model grid convergence mass flow rate](image)

The meshes are refined in terms of two criteria, namely the overall mesh density in the main flow channel and more importantly the mesh resolution across the seal fin tip clearance. As indicated in the numbers in green above the figure, the increased mesh density relates to the tip clearance regions as follows: the first number denotes the number of cells along the knife edge of the fin tip in the flow (axial) direction, and the second number represents the radial cell resolution.
across the tip clearance between the rotor and the stator. The influence of mesh density on computed pressure is also check as shown in Figure 4.3. Pressure or nodal forces are the ultimate data used for flutter calculation and from the figure it clearly shows a well converged grid even for the coarsest mesh. Flutter calculation robustness is therefore very well maintained.

![Figure 4.3 Grid convergence surface pressures](image)

Some of the key numerical parameters are presented in Table 2. As for the transient calculations, a reasonable choice of total simulation time will ensure a “stabilized” unsteady flow, i.e. a good transient periodic convergence. In order to check this, the baseline case of 4ND is first run for a relatively long time, encompassing in total 30 vibration cycles. Then the time history of pressure (on a selected monitor point) at each cycle is compared against the next one, and a percentage error could be calculated to assess the deviations. A root mean square deviation (RMSD) is defined as:

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Turbulence model</td>
<td>SST</td>
</tr>
<tr>
<td>Advection scheme</td>
<td>High resolution</td>
</tr>
<tr>
<td>Transient scheme</td>
<td>2nd order backward Euler</td>
</tr>
<tr>
<td>Inner coefficient loops</td>
<td>5</td>
</tr>
<tr>
<td>Total simulation time</td>
<td>16 vibration cycles</td>
</tr>
<tr>
<td>Time step</td>
<td>32 per vibration cycle</td>
</tr>
<tr>
<td>Sampling interval</td>
<td>Every 2 time steps</td>
</tr>
</tbody>
</table>

Table 2 Numerical Parameters
\[ \text{RMSD}(X_1, X_2) = \sqrt{\frac{1}{n} \sum_{i=1}^{n} (X_{1,i} - X_{2,i})^2} \]  

(4.2)

where \( X_1, X_2 \) refer to the two sets of pressure time history samples of the current and next vibration cycle. \( n \) denotes the number of samples per vibration cycle. The RMSD is further normalized by the mean pressure of the considered two cycles. A plot of the errors for the entire 30 oscillation cycles is shown in Figure 4.4. At cycle number 29 the error is 0.009\%, suggesting a very good periodic convergence. To strike a balance between the computation time and accuracy, the total simulation time is chosen as 16 number of oscillation cycles, which corresponds to an acceptable error of 0.07\%.

\[ \begin{align*}
\text{X: } 16 & \quad \text{Y: } 0.07034 \\
\text{X: } 29 & \quad \text{Y: } 0.00896
\end{align*} \]

\[ \text{SIMPLE MODEL PERIODIC CONVERGENCE} \]

\[ \text{RMSD AS PERCENTAGE OF MEAN PRESSURE [\%]} \]

\[ \text{NUMBER OF OSCILLATION CYCLE [-]} \]

\[ Figure \ 4.4 \ Simple \ \text{model} \ \text{periodic} \ \text{convergence} \ \text{history} \]

### 4.4 Industrial Application – Realistic Designs

Three realistic gas turbine rotor disk sealing designs are evaluated using the same methodology as applied on the simple model parametric study. The three designs are dubbed the brush seal design, the original labyrinth design and the modified labyrinth design.

The geometric layout with steady-state velocity vectors for the brush seal design is presented in Figure 4.5. Two large cavities (upstream and downstream) are included. The complete left side boundary of the fluid domain constitutes the disk rotor and the right side the stator. Apart from the normal seal fins, a brush seal
design is utilized which situates itself on the stator. This is simply modeled by having an operationally small clearance across the gap.

For the original labyrinth design, the brush seal is eliminated and all sealing elements are labyrinth seals. The modified labyrinth design has the same rotor disk as the original while featuring an increased gap clearance over the 3rd and 4th fins, as counted along the fluid channel starting from the upstream. The surface mesh for the main sealing area of the modified labyrinth design is shown in Figure 4.6, along with a close-up view of the increased-clearance regions. Smooth transitions of the mesh density are well maintained from the large cavities to the detailed fin surroundings. The meshes are created with a grid density comparable to the simple test model.

By revolving the surface mesh a three-degree grid building block (base mesh) is first generated in ICEM. The building block for each design is then imported into CFX to construct the complete fluid model, with a sector angle corresponding to the specific nodal diameter. The three-degree brush seal design base mesh and the assembled 120-degree mesh (for the 3ND model) are shown side by side in Figure 4.7. Key numerical parameters are also inherited from the simple test case.

Figure 4.5 Brush seal design geometry and steady velocity vectors
Figure 4.6 Modified labyrinth design mesh overview (upper) and fin regions (lower)

Figure 4.7 Brush design assembled 3ND mesh (left) and base mesh (right)
5 RESULTS

A total number of 318 cases were simulated, including both the simple model and the realistic designs. A typical 4ND (90-degree sector) simple model case with approximately 1.8 million nodes could be accomplished in 10 hours using 24 CPUs. The most demanding task of a 1ND (complete annulus) realistic design case with around 10 million nodes required 33 hours using 36 CPUs. All simulations finished with good convergence.

5.1 Simple Model

5.1.1 Effects of Support Side of Seal and Mechanical Frequency

The support side of a labyrinth seal has a significant influence on its stability characteristics. The combined effect with mechanical frequency is shown in Figure 5.1 and Figure 5.2, representing an LPS and an HPS configuration respectively. Both TWMs involved are 4 nodal diameter modes.

The charts could be interpreted as analogical to a frequency response, for which the mechanical or natural frequency of the seal structure is artificially varied from 0 Hz to 3000 Hz in the same nodal diameter mode (4ND). The abscissa denotes the mechanical frequency and the ordinate represents the related normalized aerodynamic work per cycle as exerted on each part of the seal in 360 degrees, marked by different colors and also illustrated schematically. The total work, which is a sum of all work contributions and which ultimately determines the stability of the entire structure, is marked with red asterisk. Two vertical dashed lines, one in blue and one in green, refer to the acoustic frequencies of the 1st and 2nd cavity.

Compared with Figure 1.4, the results of the LPS and HPS 4ND affirm Abbott’s criteria. For an LPS configuration, approximately when the mechanical frequency is lower than the acoustic frequency, the overall work is positive, which means the flow is feeding energy into the structure and thus renders it unstable. When the mechanical frequency is above the acoustic frequency, the seal becomes stable. For the HPS case, the curves are to some extent mirrored to LPS but with distinct shapes.

It is also useful to examine the work contribution from each of the surfaces. For the LPS case, at first glance the curves representing the 1st and 2nd cavity are behaving in a similar fashion, except that for the 1st cavity the work is always negative, even when the mechanical frequency is lower than the acoustic one. Hence among the two inter-fin cavities, only the 2nd cavity exactly falls into the description of Abbott’s criteria, as is the case regarding the overall stability. Work characteristics on the upstream and downstream surfaces appear to be similar to each other, but only the upstream member is showing peaks during sign switch near 600 Hz. The upstream surface is also deemed describable by Abbott’s criteria. On the other hand the downstream surface could only be partially described by the criteria. This is because when the mechanical frequency is over 1000 Hz, the aerodynamic work per cycle is very close to the stability boundary and is not always negative. For the HPS case, work characteristics from all surfaces are having a similar trend. The major sign switch position of the upstream
Figure 5.1 Nodal diameter 4 LPS stability characteristic

Figure 5.2 Nodal diameter 4 HPS stability characteristic
The downstream work curve is still very close to the stability boundary for frequencies over 1000 Hz. It is interesting to point out that in this HPS configuration the behavior of the 1st cavity could be well explained by Abbott’s criteria as opposed to the LPS configuration.

For the LPS case the high peaks near 600 Hz in the upstream curve are believed to be the direct result of an acoustic resonance phenomenon. Details are further explained in Section 6.3. It is also believed that at the same range of frequencies the dents towards the stability boundary in the 1st and 2nd cavity curves are caused by the same resonance phenomenon. This feature of the stability curves could not be observed in Abbott’s results, which is understandable since acoustic resonance is primarily dependent on the specific geometries and the ambient temperatures.

Overall, it is concluded that the majority of the work contribution is from the inter-fin cavities, especially considering the size of these cavity surfaces compared with the upstream and downstream ones. Therefore Abbott’s criteria could be readily applied to these inter-fin cavities in the same way as applied to the entire seal structure. This observation holds true with the exception of the 1st cavity for the LPS case.

5.1.2 Effects of Nodal Diameter

A parametric study of nodal diameter’s effect on stability is shown in Figure 5.3 for the LPS cases, expressed in percentage LogDec. The curves are the interpolation of discrete data points. Six nodal diameter scenarios are checked, from 0ND all the way up to 36ND. It should be noted that across the different nodal diameters the same maximum deflection of the structure is maintained. This is to make sure that all results are comparable against each other since at this stage it is not yet clear if this type of seal flutter phenomenon has linear behaviors.

The 0ND case has a distinctive all-stable characteristic, for which the details are discussed further in Section 6.2. On the other end of the spectrum is the 36ND case, which is also stable throughout the entire analyzed frequency range. The major features of these curves (excluding the two extreme ends), i.e. the crossing position at the stable-unstable boundary (zero LogDec line) and the peak-valley shape patterns, are shifting (and scaled) towards higher frequencies as the NDs grow larger. In general the magnitude of the maximum LogDec value for each ND curve is also decaying with increasing NDs. When examining individual ND curves, the aerodynamic damping values are negative at lower frequencies. As frequencies go up, the damping values cross the zero LogDec line with a steep slope and then converge back towards zero after some oscillations. From the above observations, lower order NDs and lower range frequencies should be of greater significance when dealing with LPS labyrinth seal flutter problems.

As mentioned before, to accurately describe the behavior of stability curves Abbott had proposed the idea of acoustic frequency in relation to the natural frequency of the vibrating rotor seal. This criterion has been reaffirmed when looking at the 4ND LPS (Figure 5.1) and HPS (Figure 5.2) characteristics, both for the overall stability and the individual inter-fin cavity behaviors. To check its validity across different
NDs, the work per cycle on the 1st cavity surface is compared in Figure 5.4. It seems that as the NDs grow larger, the peak-valley patterns are scaled towards higher frequencies in a linear fashion.

Figure 5.3 Nodal diameter stability characteristic

Figure 5.4 First cavity stability characteristic across NDs
However, since the 1st cavity is always stable and does not behave exactly as described by Abbott’s criteria, a 2D plot of the work per cycle on the 2nd cavity surface is presented in Figure 5.5 and with better visualization. Two dashed lines are drawn to highlight the stability crossing boundary (black) and the acoustic frequencies (blue) respectively. The acoustic frequency line is already defined as a linear function of NDs (Equation 1.13). The stability crossing line is obtained by interpolating the available data on the curves.

![Second Cavity Nodal Diameter Stability Characteristic](image)

*Figure 5.5 Second cavity stability characteristic across NDs*

To investigate if the stability crossing line is in fact linear, Figure 5.5 is re-plotted into Figure 5.6, where the mechanical frequencies belonging to the same ND family are normalized by their respective nodal diametral acoustic frequency. It shows that all the available ND curves are crossing the zero LogDec line at approximately the same position (0.6) after normalization. If the stability crossing line is a linear function of ND (assuming it is across the origin), the stability crossing frequency for each ND family could be expressed as:

$$\omega_c = k_c n$$  \hspace{1cm} (5.1)

where $k_c$ is the slope and $n$ the nodal diameter. Putting Equation 5.1 and 1.13 into Equation 1.14, the stability crossing frequency ratio then becomes constant:

$$\eta_0 = \frac{\omega_c}{\omega_{ac}} = \frac{2\pi r k_c}{c \pm v_{slip}} = \text{const}$$  \hspace{1cm} (5.2)
The second cavity is further checked regarding the stability crossing positions for different NDs. Back to the original definition of aerodynamic work per cycle for flutter calculations (Equation 1.2), for any node on the structure the underlying parameters that dictate the stability are the amplitude of the nodal force, the amplitude of the nodal displacement, and the phase lag between the force vector and the displacement vector. To this end a monitor point is chosen on the horizontal surface of the 2nd cavity, and the amplitude of the nodal force and the phase lag between the two vectors are plotted (Figure 5.7) across the NDs. The amplitude of the nodal displacement is neglected because it is maintained the same for all NDs.

It appears that when ND increases the nodal force curve is essentially shrinking with regard to the point at approximately (0.67, 0.018), in both horizontal and vertical directions. The phase curves are more important because they ultimately determine the stability crossing positions of the work characteristics. Similar to the nodal force curves, with increasing NDs the phase curves are also shrinking in reference to one point at around (0.75, -0.28). This point is not located on the stability boundary and therefore the stability crossing positions are not the same for all NDs, even though they are quite close to each other.

Based on the above analysis with respect to the stability crossing positions, it is concluded that Abbott’s criteria and in particular the concept of acoustic frequency are very useful and practical in describing the stability characteristics of rotor seals regardless of the nodal diameter mode. After the acoustic frequency normalization the stability crossing positions for all NDs are approximately the same.
As for the relative positions of the peak-valley features, they are not referenced to a common frequency ratio value (as is the stability crossing position) after the normalization by the corresponding acoustic frequencies. However, they do appear linear in Figure 5.5. One possible explanation is that they are linear but not crossing the origin. Another reason might be that the peak-valley shape patterns are not the results of motion-induced flow perturbations alone. As mentioned before, it is suspected that the “valleys” as seen in Figure 5.6 and Figure 5.7 is not the underlying stability characteristic of the 2nd cavity itself, but rather a result and interplay of the acoustic resonance phenomenon formed in the upstream cavity at specific frequencies (see Section 6.3). For the 4ND case, it takes place at around 600 Hz. For the other NDs, the resonance frequencies are slightly different, shifted towards higher values with increasing NDs. Without these acoustic-resonance-induced “valleys”, it is a possibility that all NDs would have the maximum damping reached at the same frequency ratio.
Figure 5.8 Nodal diameter stability characteristic vs. frequency ratio

Figure 5.3 is re-plotted in Figure 5.8 with acoustic frequency normalization, which resembles Figure 5.6 to a great extent. This again proves that in the context of rotor seal aeroelasticity, inter-fin cavities are the dominating factor compared with other seal surfaces and dictate the overall behavior in spite of their smaller surface size. Abbott’s criteria are valid in describing these cavities.

The detailed stability characteristics for 3ND, 5ND and 36ND are shown in Figure 5.9, Figure 5.10 and Figure 5.11. The importance of acoustic frequencies in referencing the stability characteristics is again well displayed.
Figure 5.9 Nodal diameter 3 LPS stability characteristic

Figure 5.10 Nodal diameter 5 LPS stability characteristic
5.1.3 Effects of Traveling Wave Direction

Another important parameter is the TWM direction. The underlying reason is that during rotation, the flow between the rotor and the stator is traveling slower than the rotating member (where slip velocity is defined as the difference between the two). This determines that the forward and backward TWMs would in fact experience different fluid velocities if one travels with the TWMs.

In another interpretation, the speed of sound is different as felt by a person riding on a forward or backward TWM. Imagine one is in the reference frame of a backward TWM: the speed of sound regarding this frame of reference is actually the sum of the speed of sound in the fluid’s frame of reference and the slip velocity. On the other hand for a forward TWM, it is the speed of sound in the fluid’s frame deducted by the slip velocity. This is well represented in Equation 1.13 for the calculation of acoustic frequency.

Shown in Figure 5.12 is the stability characteristic for a 4ND backward TWM. All setup is the same as the baseline case except for the TWM direction. The major features in Figure 5.1 are also exhibited here for all the surfaces, except that at lower frequencies, the overall work is crossing the stability boundary twice. Other than this the difference between the forward and backward TWMs for LPS seals is essentially a shift or scaling towards the right hand side in the horizontal axis. It is noted here that the acoustic frequency is still serving as a good reference to mark the relative axial positions of the curve.
5.1.4 Effects of Pressure Ratio

Five pressure ratio configurations are investigated at two chosen mechanical frequencies, namely 300 Hz and 1000 Hz. One is below the acoustic frequency and one is above. The pressure ratio ranges from 2 to 6, as shown in Figure 5.13 and Figure 5.14.

The normalized aerodynamic work per cycle is growing essentially linearly as function of pressure ratio for the majority of the seal surfaces, including both the inter-fin cavities and those upstream and downstream larger surface areas. However, the linearity is not exact and small deviations do exist for higher pressure ratio cases, depending on the mechanical frequency and the specific surfaces. For small damping values especially near the stability boundary, it is noted that the behavior could not be well described. This is anyhow a general issue when predicting stability near the boundary.

With this analysis it could be concluded that a change of pressure ratio in a considerably large range would not significantly affect the phase relationships of labyrinth seal aeroelastic behaviors, rather a mere amplification of the nodal force magnitude and consequently the aerodynamic work per cycle.

The following hypothetical scenario could arise as a result of this behavior. During engine startup, the power is ramping up together with the temperatures and pressures. If a seal disk is inherently unstable, this could cause a magnitude increase in negative aerodynamic damping. When material, friction and other sources are unable to provide enough available positive damping, the startup could ultimately trigger a disk flutter event.
Figure 5.13 Nodal diameter 4 pressure ratio stability characteristic at 300 Hz

Figure 5.14 Nodal diameter 4 pressure ratio stability characteristic at 1000 Hz
5.1.5 Effects of Maximum Deflection

To check the linearity of the system with respect to oscillation amplitude, the maximum deflection of the seal structure is varied starting from one quarter of the baseline maximum deflection. The process was easily achieved by applying a scaling factor on the mode shape data before being imported into CFX.

The normalized aerodynamic work per cycle is plotted against several maximum displacements as fraction of the baseline (reference) case (Figure 5.15). The aerodynamic work per cycle is normalized by maximum deflection, thus analogous to the unsteady force magnitude. Apparently as the vibration magnitude increases, the unsteady force magnitude is increasing in a linear manner.

![Figure 5.15 Nodal diameter 4 flutter linearity](image)

The linearity assumption is again checked by tabulating the percentage LogDec. As elaborated in Section 1.3, if the nodal force is a linear function of the nodal displacement, then the LogDec would remain constant whatever the maximum displacement value is. In Table 3 the damping values are in principle the same although with a slight increase for larger displacements. It is also noted that the linearity assumption is valid in a relatively wide range. The maximum displacement for the baseline case corresponds to approximately 20% of the fin gap clearance. Further, the proof of the linearity for labyrinth seal systems suggests that linear harmonic solvers could be used for this type of analysis.
5.1.6 Effects of First Fin Clearance

One additional geometric modification is made on the first fin clearance. The gap clearance is doubled compared to the baseline case. The stability characteristics are shown in Figure 5.16.

![Figure 5.16 Nodal diameter 4 LPS stability characteristic, doubled first clearance](image)

Compared with Figure 5.1, all curves are essentially the same except for the one associated with the 1st cavity. For the baseline case this part of the seal is always stable, whereas for the doubled clearance case it seems that the curve is shifting upwards and over the boundary, particularly at the mid-range frequencies around 600 Hz. The situation for the lower and higher frequencies is just the opposite. The work is almost doubled towards the stable side. Serving as a boundary between upstream and inter-fin cavities, the double-clearance first cavity could have unique characteristics and probably a mix between the two sides. Imagining an overly increased clearance on the 1st cavity, it would very likely behave the same way as the upstream cavities and no longer exhibit the characteristic of an inter-fin cavity.

The effect of the doubled clearance is further checked in Figure 5.17, where the work difference between the double-clearance case and the baseline case for each of the surfaces is plotted. As far as the total work difference is concerned, it confirms that the change in clearance has made the mid-range frequencies (approximately from 400 Hz to 1400 Hz) more unstable while pushing the lower and higher frequencies towards the stable side. The contribution to the work difference from each surface is rather different. It appears that the first cavity is affected the most, constituting the majority of the total work difference. The second cavity has a very similar pattern as the first one but with smaller amplitudes. The upstream surface is making an even smaller contribution, whose curve oscillates...
around the zero line at mid-range frequencies. It is unknown what causes such oscillations but it should be pointed out that these oscillations are to some degrees correlated to the second cavity curve. The contribution from the downstream surface is very small and almost non-existent.

![Normalized Aerodynamic work per cycle difference diagram](image)

*Figure 5.17 Work difference, double-clearance case over baseline case*

In order to further examine the change in stability characteristics brought by the increased first fin clearance, the nodal force amplitude and the phase lag between the nodal force and displacement vectors are shown in Figure 5.18 for a monitor point sitting on the 1st cavity. It is seen that both the nodal force amplitude and phase lags are affected.

At frequencies below 400 Hz the increase in nodal force amplitude is significant for the double-clearance case. This contributed in part to the increase of the negative work amplitude and made the seal more stable. It is also discovered that double-clearance nodal force amplitude curve looks a lot like the one from an upstream monitor point (refer to Figure 6.9). The peak nodal force value is reached at the resonance frequency near 600 Hz where the phase lag becomes 180 degrees.

In general the clearance of the first fin is essential to the stability characteristics of the seal structure. An increase in clearance not only has the largest effect on its direct downstream cavity but can influence other surfaces as well. It is also found that the acoustic resonance in the upstream cavity has propagated to the 1st cavity due to such an increased clearance, which makes the behavior of the 1st cavity shifted towards its upstream neighbors.
5.2 Realistic Designs

Figure 5.20 and Figure 5.21 show the detailed work contributions from the divided seal surfaces (marked by different colors) for the brush seal design, 3ND backward and forward TWMs. The first bar denotes the overall work. The divided seal surfaces are listed in a clock-wise sequence corresponding to the geometry (Figure 5.19). Surfaces belonging to the same inter-fin cavities are grouped by rectangles and seal fin tip surfaces (knife edges) are indicated by black triangles.

Looking at each rectangle group individually, it is discovered that the left and the right bars have opposite work contributions, and the overall work for that specific cavity is dictated by the bar in the middle, i.e. work per cycle from the horizontal surface. It is also found that the majority portion of work per cycle for the entire seal is coming from the inter-fin cavities, which is consistent with the simple seal model. In addition, contribution from the right rim (the surface area between the 2nd and the 3rd fins, as indicated by bar 24 and 25 between the two green rectangle groups) could be prominent such as for the 3ND forward TWM case. On the other
hand it provides minimal work compared with the inter-fin cavities for the 3ND backward TWM case.

More importantly it is clear that the inter-fin cavities located on the right and left hand side of the seal structure are behaving differently in a fundamental way. Based on the modal analysis, the inter-fin cavities on the right are pivoting around a virtual point somewhere to the left of seal while the cavities on the left are pivoting around a point to the right. This consequently makes the above mentioned cavities LPS and HPS respectively. A neutral support position is the point on the seal separating LPS and HPS cavities. At this specific point the structure is only moving sideways while pivoting around itself.

The behavior of these two types of cavities is already examined in the parametric studies, and again the governing principles are proven to be valid here. The transition from LPS to HPS cavities are colored from green, yellow to red. The yellow group itself evidently shows such a transition as well, and leans slightly towards HPS behavior.

The overall stability characteristics, the details of work contributions for all three designs and further conclusions on the root cause for instability are explained in Section 6.4.

![Figure 5.19 Detailed surface division for brush seal design](image-url)
Figure 5.20 Brush seal design 3ND backward TWM stability contributions

Figure 5.21 Brush seal design 3ND forward TWM stability contributions
6 DISCUSSION

6.1 Historical Failures and Related Experimental Results

The literature review has already been conducted in Section 1.2. In this section a comprehensive review of the related historical failure cases, experiments and observations is put forward, in the hope of providing some context and finding a common basis to compare with the performed numerical analysis. Further, it would be beneficial to try to explain some of the historical failure cases using principles found in the current study.

Alford (1964) first reported that labyrinth seals suffered HCF due to aeroelastic instability. These incidents appeared to be free from resonance between the rotor and stator. And both the rotor and the stator element of the seal are found to be susceptible to such instabilities.

From experience, the support side of the seal proved to be of significance on the overall stability. Alford pointed out that all stationary seals that had failed were supported on the entrance high pressure side, and no known stationary seals had failed if supported on the discharge low pressure side. The same rules apply to the rotating member as well. Such an HPS configuration (in reference of rotating seal) is shown in Figure 6.1.

![Figure 6.1 Rotor supported at entrance side; from Alford (1964)](image)

Ehrich (1968) showed that one design modification for a flight propulsion gas turbine with slightly enlarged labyrinth seal disk diameter resulted in a number of flutter failures (of the rotor elements) while the original smaller diameter design was running well. Experiments were conducted on both designs and one case of instability was observed on the large diameter design, which indicated a two nodal diameter forward traveling wave mode. The mechanical frequency spectrum of both seals is shown in Figure 6.2. Frequency value for 1ND was not shown due to the difficulty in measuring this mode.

From Ehrich’s experience, the most unstable mode is in fact the mode with the smallest mechanical frequency (without consideration of the first mode). A larger diameter disk directly caused the smaller frequencies for all the modes, in essence a shift downward. Reviewing Ehrich’s report retrospectively, at first glance the
cause of failure could be that it is a low pressure support seal (not mentioned in the paper though) and the mechanical frequency is falling below the acoustic frequency far enough to render the seal unstable.

Armstrong in his follow-up discussion of Alford's (1967) paper disclosed a similar example of seal instability but in fact a mechanical-acoustic coupled resonance. Fatigue cracking was found on the stator of a compressor discharge labyrinth seal during an aero-engine development project. For the investigation a representative static rig was setup with realistic environmental temperatures and pressures. The seal configuration is shown in Figure 6.3.

The stator was found to vibrate in a 6ND TWM. Later the instability was attributed to acoustically coupled vibration, i.e. a resonance between the acoustic waves in the annular cavity and the mechanical waves in the seal structure, oscillating in a mode with the exact same number of nodal diameters.

The problem was later solved by inserting radial baffles across the annular air cavity with equal distance as well as a more securely supported stator member. It appears that the traveling wave of the fluid could be stopped by the inserted baffles and a strengthened stator would contribute to smaller structural deflections, consequently smaller pressure perturbations.

Figure 6.2 Spectrum of seal natural frequencies; from Ehrich (1968)
Lewis (1978) documented aeroelastic instability-induced HCF as encountered by the F100 labyrinth air seals. Such seal configurations are seen in Figure 6.4, where the seal member is rotating and the land is stationary.

The problem was found during the brief transient period when the engine accelerated to maximum speed. At this time the left side of the seal cylinder surface would snap lose, thus making it supported on the right hand side, or low pressure side. Mechanical coincidence was excluded and the instability was found to be a 5ND TWM, yet the TWM direction is not specified in the paper.
The stability issue is fixed by increasing the upstream first fin tip (knife edge) clearance. The increased leakage mass flow as a result of the enlarged first clearance is compensated by subsequently decreasing the last tip clearance. The aeroelastic instability is claimed to be eliminated by such modifications.

According to the simple seal model parametric study in the current thesis, an increase in the first fin tip clearance for the LPS case results in an increased stability at lower and higher mechanical frequencies. The information available in Lewis’ report is however insufficient for making a sound comparison. Besides, both the stationary and rotating members in the examined F100 air seal appear rather thin. The related mode shape might not be similar to conventional rotor seal disks and hence the stability characteristics could be totally different.

Abbott (1980) reported that some unexpected aeroelastic problems were found on the inner seals of certain designs. Such failed inner seal fin parts are illustrated in Figure 6.5. The blackened areas denote oxidized crack regions (O - fatigue origin; 1, 2, 3 – crack progression). Recall in Figure 1.1 the inner seal is supported on the high pressure side.

Later a static rig is designed with realistic flow conditions. A range of pressure values are supplied to simulate actual engine operation environments. The inner seal is found to start vibrating over a large range of pressure ratios. The identified instability is caused by a 3ND TWM which is in fact the lowest frequency mode for the seal. The TWM direction is unknown for the failures in the real engine. For the test rig the TWM direction does not matter since this is a static rig in the first place. There would be no swirling flow and both forward and backward TWMS are thus treated as the same.

Abbott then designed an experiment to add six weights equally spaced on the disk (Figure 6.6) to artifically alter the mechanical frequency of the 3ND failure mode.
As the weights are slowly increased, the 3ND mechanical frequency could be decreased, hence gradually making it equal and then lower than the 3ND acoustic frequency. Results indicated that when the seal frequency is actually equal or lower than the acoustic one, the structure would seize to be excited by the flow.

In di Mare et al's publication (2010) the numerical analysis is deemed capable of predicting the aeroelastic behavior as found in testing for the two examined seal configurations (3-fin seal with stiffener and 4-fin seal without stiffener). The most unstable mode from experiments is not specified yet from the numerical results it is the 2ND backward TWM of the 4-fin configuration.

Based on Figure 1.9, Figure 1.10 and Table 4 (acoustic and natural frequencies for 1ND to 5ND, both 3-fin and 4-fin configurations) the authors indicated that Abbott’s criteria did not readily apply to this specific geometry. In Figure 1.10 the work contributions from the inter-fin cavities do appear rather sporadic and unexplainable. The reason could be that the inter-fin cavities are not entirely consisting of LPS cavities. For example, the cavity to the left is very likely an HPS cavity. Work contributions from each of the inter-fin cavities could be presented individually in order to apply Abbott’s criteria.

It should also be noted that in Abbott’s criteria the crossing position of the stability boundary is not directly located at the coincidence between the mechanical and acoustic frequencies. For an LPS seal the stability boundary is found for a mechanical frequency below the acoustic frequency and for an HPS seal above
the acoustic frequency. This is shown by Abbott and the same results are obtained in the current publication as well. Therefore if the mechanical frequency is slightly lower than the acoustic one, the seal could still be stable.

<table>
<thead>
<tr>
<th>Cavity 1</th>
<th>4-fin configuration</th>
</tr>
</thead>
<tbody>
<tr>
<td>448.6</td>
<td>370.1</td>
</tr>
<tr>
<td>464.4</td>
<td>365.1</td>
</tr>
<tr>
<td>440.3</td>
<td>358.8</td>
</tr>
<tr>
<td>Natural frequency (Hz)</td>
<td>483.5</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Cavity 2</th>
<th>3-fin configuration</th>
</tr>
</thead>
<tbody>
<tr>
<td>439.7</td>
<td>366.1</td>
</tr>
<tr>
<td>440.0</td>
<td>364.8</td>
</tr>
<tr>
<td>Natural frequency (Hz)</td>
<td>393.3</td>
</tr>
</tbody>
</table>

Table 4 Mechanical and acoustic frequencies; di Mare (2010)

Combining the information from Figure 1.9 and Table 4, it appears that the 4-fin design’s instability is probably caused by low mechanical frequencies in general, as compared with related acoustic frequencies. The 3-fin design turns out superior to the 4-fin counterpart. Besides having a different number of fins and inter-fin cavities, the 3-fin setup is also having a stiffening ring on the disk. This drastically increased the overall mechanical frequencies and is very likely the main positive contributor to the perceived stable design.

6.2 Nodal Diameter Zero (Umbrella Mode)

It has been well established that the flutter mechanism of rotor seals is a three-dimensional effect, in which two motions are primarily involved. One is the opening and closing of the seal teeth during oscillation in the radial direction. The other is the flexural vibration of seal surfaces in TWMS coupled with air movement in the annular cavities in the circumferential direction. The 0ND case is studied in order to check if axisymmetric analysis is sufficient with its greatly reduced cost in computation. In addition this helps to isolate the effect of the opening and closing of seal teeth apart from the circumferential influence.

Compared with the baseline 4ND case as well as other NDs, the 0ND “umbrella mode” shares the major features of the stability characteristics (Figure 6.7). The only noticeable yet crucial difference is the work on the inter-fin cavities. First of all for the 2\textsuperscript{nd} cavity with frequency ratio lower than one, the work per cycle on the disk is always stabilizing. Even for the 1\textsuperscript{st} cavity in the same frequency range, the curve is bulging towards negative work whereas for the other non-zero NDs it is leaning to the stability boundary.

This 0ND investigation shows the need for 3D modeling in the flutter calculation of labyrinth seals. An axisymmetric model is insufficient as it does not allow the fluid waves to travel in the circumferential direction. It is also obvious that 0ND flexural vibration would not easily cause aeroelastic instabilities since the work contributions from inter-fin cavities are almost always stabilizing.
6.3 Acoustic Resonance and Coupled Vibration

One interesting phenomenon discovered during the “frequency sweeping” for plotting the stability characteristics is the acoustic resonance. It seems that simply by using a URANS solver with a conventional turbulence model acoustic resonance could already be captured, although the accuracy of which could not be assessed. Normally when addressing acoustic issues using CFD techniques, an LES model should be more appropriate. Yet that comes with much more intensive computations.

During the CFX runs around certain simulated frequencies (the alleged acoustic resonance frequencies) the reversed-flow warnings are flagged in the monitor, some causing blockage of the flow up to one quarter of the area of an inlet. The unsteady pressure magnitude is plotted for one cross section in Figure 6.8. In this single slice (circumferential plane) the resonance is seen as a standing wave. The maximum amplitude (anti-node) is in the upstream flow channel near the entrance to the labyrinth seals. For a complete annular space including the circumferential direction, the acoustic mode is in fact a TWM, with the same number of NDs and the same frequency as the mechanical TWM, moving in the same direction.

A monitor point under the seal cantilevered beam in the upstream cavity is further analyzed. The nodal force amplitude as experienced on this node and the corresponding phase lag between the nodal force and nodal displacement are plotted for the complete frequency range in Figure 6.9. It is shown that the resonance frequency is around 600 Hz, where the nodal force amplitude reaches its peak value and the phase turns zero.
Such an alleged acoustic resonance appears both beneficial and undermining to the stability of seals because of the sign switch of the phase lag near resonance. However, a coupled acoustic and mechanical resonance phenomenon should always be avoided even if the work is stabilizing. In reality engines are not just running at design points but rather through a range of different conditions. Since the acoustic resonance phenomenon is dictated by the temperature, a stabilizing acoustic resonance work value could easily shift towards the opposite side when the operating condition changes.
6.4 Discussion of Realistic Designs

A detailed discussion of the realistic designs and the application of Abbott’s criteria are presented in this section. Figure 6.10 presents an overview of the ultimate stability characteristics (as calculated in percentage LogDec) including all the analyzed nodal diameters, for the original labyrinth design, the modified labyrinth design and the brush seal design. The predicted behaviors of all three designs agree well with experimental indications in terms of both critical ND identification and relative levels of instability.

![Figure 6.10 Realistic designs overall stability characteristic](image)

Previously the seal surfaces have been subdivided into a series of elementary parts to display their individual work contributions (Figure 5.20 and Figure 5.21). Here the parts are regrouped for simplicity, constituting the major features of the seal. They are named the inter-fin cavities (from 1 to 6), the right rim, the disk upstream surfaces and the disk downstream surfaces, as indicated in Figure 6.11 for the brush seal design. The special black symbol shows the approximate position of the brush seal.

The aforementioned surfaces primarily fall into two categories according to their distinctive aeroelastic behaviors. They are the inter-fin cavity surfaces (2 to 6) and other surfaces. The normalized aerodynamic work per cycle is plotted against the analyzed NDs in Figure 6.12 for the inter-fin cavity surfaces; Figure 6.13 shows the work for other surfaces.
Going through the inter-fin cavities from upstream to downstream, the seal virtually shows a transition from LPS to HPS cavities for all NDs. As mentioned before, according to the modal analysis Cavity 2 and 3 are supported on the low pressure side and Cavity 5 and 6 are on the other hand both HPS cavities. Cavity 4 serves as a bridge between the two groups while slightly leaning towards HPS behavior. The LPS and HPS cavities are then easily identified and separated in Figure 6.12. It could be seen that all LPS cavity work is negative, hence stable, whereas all HPS cavities including the transitional Cavity 4 are receiving positive work from the fluid, hence unstable. The curves to some extent are forming a dumbbell shape pattern, indicating that at mid-range NDs the absolute value of the work per cycle is relatively larger.

For the other surfaces, no clear patterns could be found. The upstream and right rim surfaces are contributing the most work. Together with Cavity 1, the three surfaces are behaving in a similar manner, although rather unpredictable across NDs. The downstream cavity does not seem to belong to this category. It looks similar to those HPS work contributions from the previous figure, but with smaller amplitude.

The stability characteristic for Cavity 1 is again interesting for discussion. It is supported on the low pressure side, but does not follow Abbott’s criteria. Instead it exhibits the same traits as the right rim and upstream surfaces. In this case it is believed that both Cavity 1 and the right rim are acting like “bridging cavities” between upstream and the traditional inter-fin cavities. As a result of radial deflection due to thermal expansion and centrifugal forces, the tip clearance for Cavity 1 is very large compared to the others, therefore making the “bridging cavities” behave more like upstream surfaces than inter-fin cavities.
Figure 6.12 Brush seal design stability contributions: inter-fin cavities

Figure 6.13 Brush seal design stability contributions: other surfaces
Back to Figure 6.10, for the brush seal design the most unstable mode is -3ND, i.e., the 3ND backward TWM. The root cause for the perceived instability is explained as follows. Since the brush seal design has an inherent majority of HPS inter-fin cavities, the net outcome of work contribution from the inter-fin cavity group is unstable in the first place. Fortunately the unconventional surfaces, especially the upstream and right rim, are balancing out the destabilizing work. However at -3ND and -4ND the contribution from the above two surfaces is almost non-existent, or even slightly unstable. This eventually makes the -3ND mode the most flutter-prone mode and the nearby NDs undesirable as well.

### 6.5 Extension to Abbott’s Criteria

Abbott’s criteria were originally proposed to describe the stability characteristic of the entire seal structure as a whole based on its support location and attitude. Now it has been shown that the same criteria apply to individual inter-fin cavities as well. Therefore when checking labyrinth seal stability, it is recommended that work contribution from each cavity is examined separately, especially for seals with a mixture of LPS and HPS cavities.

![Figure 6.14 Brush seal design frequency ratio vs. NDs](image)

Applying Abbott’s criteria to the brush seal design, the mechanical-to-acoustic frequency ratios for all “enclosed cavity surfaces” are plotted in Figure 6.14 over the investigated range of low-order NDs. Strong correlation could be found when referencing the respective work contributions in Figure 6.12. First it could be observed that all frequency ratios are above one. According to the criteria, LPS cavities should be stable and HPS cavities unstable, which is in fact the case. Even the work per cycle magnitude is to some degree well predicted. When
frequency ratios are above one, larger ratios would result in smaller work magnitude. This is reflected by the two “valleys” in Figure 6.14 correlating to the dumbbell shape in Figure 6.12. In addition, a lower “valley” at backward TWMs makes the “radius” of the left dumbbell “sphere” larger.

For the other two realistic designs, i.e. the original labyrinth design and the modified labyrinth design, the same criteria apply. The difference between the three designs is caused by their respective LPS and HPS cavity work distribution as well as the seemingly erratic work contribution from the right rim and upstream surfaces.

6.6 Influence of Initial Fin Clearance on Realistic Design

Lewis et al (1978) documented the importance of the first or initial fin clearance on rotor seal stability and claimed that flutter could be eliminated by controlling such clearances. For the current study the comparison between the original labyrinth design and the modified labyrinth design serves as a prime example for analyzing the influence of the initial fin clearance. Figure 6.15 shows the surface division for the two labyrinth seal designs. Figure 6.16 and Figure 6.17 present the work contribution from the two surface groups respectively. In the figure legends ORG denotes the original design and MFD the modified design.

For the most part the increase in gap clearance (in modified labyrinth design over Cavity 2) has led to a decrease in work magnitude for most surfaces in both surface groups. For certain NDs the work contribution from Cavity 2 and 3 even shifted to the unstable (positive work) side. It is also noticed that for -4ND and -3
ND the work from the right rim and the upstream surface have become more stabilizing. In general curves from most surfaces have become more “flat”. The change in clearance virtually affects every part of the seal surface, from upstream to downstream. These intricate changes collectively resulted in the different overall stability characteristics as seen in Figure 6.10.

Figure 6.16 Two labyrinth design stability contributions: inter-fin cavities

Figure 6.17 Two labyrinth design stability contributions: other surfaces
6.7 Preliminary Design Guidelines

Based on historical criteria, experiments, analysis, together with the knowledge acquired through the current study, the following design guidelines are proposed:

Eliminate HPS cavities. HPS cavities could be stable for frequencies smaller than the acoustic frequency. However for a disk-like structure, the frequency goes up exponentially for higher modes, thus making the crossing of the stability boundary eventually inevitable. LPS cavities on the other hand are more favorable as long as larger-than-one frequency ratios with good safety margin are ensured.

From the mechanical design point of view, the support location could be positioned to make sure all cavities are LPS cavities, or at least a majority of them are. The lowest frequency ratio should be checked and maintained above one. If the frequency ratio requirement is not met, the disk could be stiffened or a smaller disk diameter could be employed to get sufficiently high mechanical frequencies.

In addition, acoustic resonance should always be avoided. Acoustic resonance frequencies can be determined by performing FE based acoustic calculations that are available today in most FE solvers.

Finally, aeromechanical analysis like the one demonstrated in the current study can be conducted to provide detailed information for each design. It has been shown that the effects of small design modifications can be accurately captured by such calculations whereas simple criteria might not be able to differentiate.
7 CONCLUSIONS

Using time-marching CFD techniques with morphing meshes, extensive high fidelity flutter analysis has been performed to assess the aeroelastic stability of rotor seals.

First a number of parameters as identified by historical studies were varied to see their effects on the perceived instability. These parameters include the support side of the seal, mechanical frequency, nodal diameter, traveling wave direction, pressure ratio, first (upstream) fin clearance, and seal deflection amplitude. The calculations were done on a simplified seal model yet the core features of labyrinth seals were well maintained. Results from the parametric study agreed with the key historical findings while at the same time proposed some new interesting ideas.

Next using the same method three realistic gas turbine labyrinth seal designs were evaluated. Such case studies not only reaffirmed the basic principles of rotor seal flutter found during the parametric study but showed promising agreement with experimental indications as well. Finally, preliminary design guidelines were put forward.

7.1 Future Work

In light of a successful application of time-accurate CFD techniques to labyrinth seal flutter problems, more parametric studies could be done regarding the geometrical configurations of the seal. Detailed sealing fin features should be studied, including but not limited to, the number of fins, their relative axial placement, fin height, thickness, and individual fin tip clearances. The clearances should be of great concern since they ultimately determine the leakage mass flow, which in turn affects the engine performance. An increase of the upstream fin tip clearance has been shown to affect the stability in a significant way, both in previous experiments and in the current study. Optimization of the tip clearance for each fin could lead to a balanced design in terms of stability and engine performance.

Despite the fact that the computational cost of labyrinth seal flutter calculations could be managed within current industry standards, efforts could still be made on streamlining the process. Most numerical parameters were chosen conservatively in this initial analysis including inner loop coefficients, sampling frequency, total simulated vibration cycles etc. After a sensitivity study of these parameters the computation time could likely be further reduced.

Even though time-marching CFD could be used in detailed design analysis, the numerous routine design iterations require a faster approach, such as linear harmonic methods. Current codes could be modified to account for the morphing of cyclic boundaries during seal flutter calculation. Yet above all, a single flutter parameter as used in blade flutter preliminary design is very much favored and more effort could be spent on analytical methodology development.
8 REFERENCES

Abbott, D. R., 1980
“Advances in Labyrinth Seal Aeroelastic Instability Prediction and Prevention”
ASME Paper No. 80-GT-151

Alford, J. S., 1964
“Protection of Labyrinth Seals from Flexural Vibration”

Alford, J. S., 1967
“Protecting Turbomachinery from Unstable and Oscillatory Flows”

Alford, J. S., 1975
“Nature, Causes, and Prevention of Labyrinth Air Seal Failures”

Carta, F. O., 1988
“Aeroelastic Coupling – An Elementary Approach”

Di Mare, L., Imregun, M., Green, J. S., Sayma, A. I., 2010
“A Numerical Study of Labyrinth Seal Flutter”

Ehrich, F., 1968
“Aeroelastic Instability in Labyrinth Seals”

“Aeroelastic Instability in F100 Labyrinth Air Seals”
AIAA Paper No. 78-1087

Phibel, R., di Mare, L., Green, J. S., Imregun, M., 2009
“Numerical Investigation of Labyrinth Seal Aeroelastic Stability”
ASME Paper No. GT2009-60017

“Aeroelastic Instabilities in Labyrinth Air Seal Systems”
ASME Paper No. 84-GT-169

Verdon, J. M., 1987
“Linearized Unsteady Aerodynamic Theory”