LARGE-DIAMETER SPIRAL GROOVE FACE SEAL DEVELOPMENT

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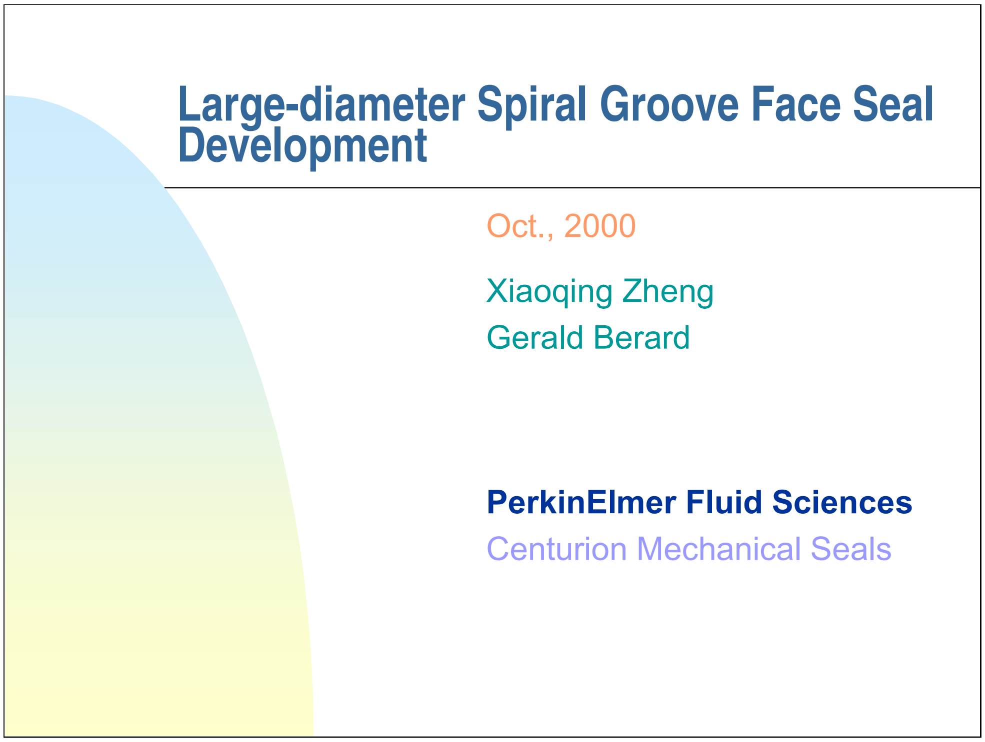

Large-diameter Spiral Groove Face Seal Development

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This presentation reports our recent development of film-riding face seals for large diameter gas turbine engines.

A new design tool, incorporating a commercial finite element program ADINA, has been developed for advanced analysis of film-riding face seals. This code is capable to model transient fluid-structure interaction inside a general seal configuration, enable designer to predict seal responses to speed, temperature and pressure changes in a time-dependent manner. Therefore, through evaluation of influences of different seal parameters, the application envelopes of conventional spiral groove face seal can be extended to larger diameter, higher speed range.

For applications that seal surface coning due to high speed and large diameter is too much for a conventional spiral groove seal to handle, a new double spiral groove design has been developed with significantly increased angular film stiffness. Axial and angular stability are crucial for successful operation of large diameter seals. Like the original double spiral groove design, the seal face consists of a pair of spiral groove seal sections, but the new design features an outward pumping groove section in the outer region. Both inner and outer grooves are fed through one set of deep middle feeding grooves that are connected to high-pressure gas through restricted orifices. The new design simplifies the seal stator ring, while resulting in a more robust concept less dependent on the thermal properties of materials. The feeding holes that lead high pressure into the middle feeding grooves are designed to have restrictive effects on feeding groove pressure when film thickness is large. Additionally, this greatly improves the film stiffness in large film gap regions.

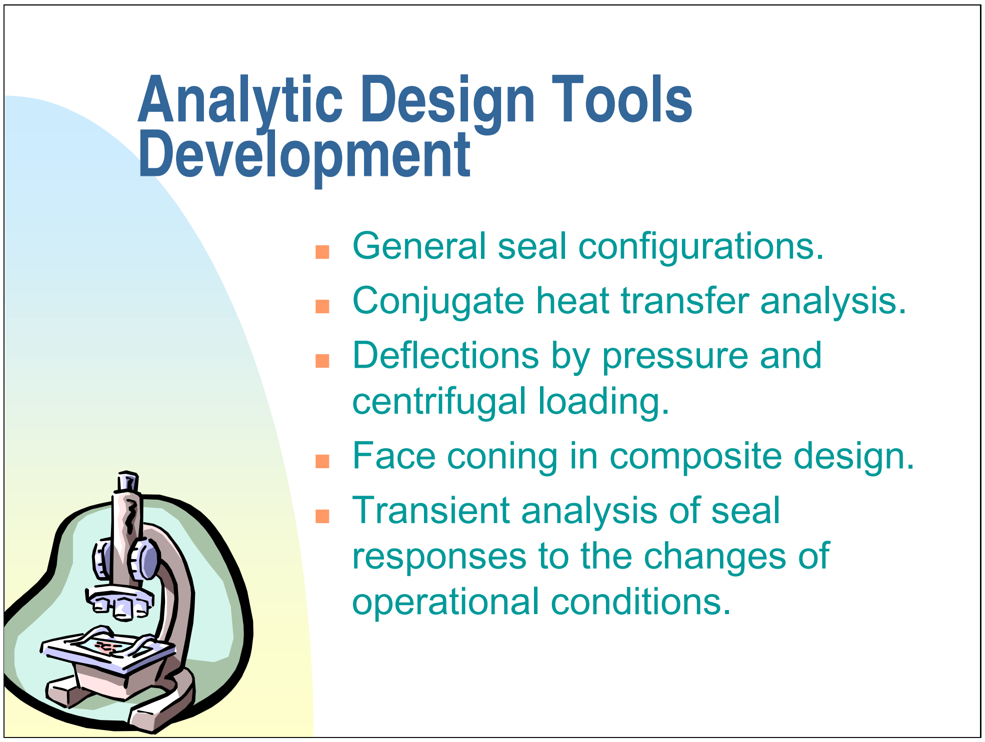
A computer program has been developed to analyze and design the new double-spiral groove seal. ADINA was used to analyze the orifice restriction factor of the feeding holes. Through calculating pressure drops at different mass flow rates in various sizes of feeding holes, an empiric formula is obtained from the computational results to relate the pressure drop ratio to flow parameters. The simple formula resulted from heavy, extensive computation is plugged into the seal design code to obtain fast solution.

Introduction

- Non-contacting, film-riding face seals for large diameter gas turbine engines.
- Low leakage, low wear.
- High speed.
- High temperature.
- High angular deflection.

Non-contacting, film-riding face seals have been used successfully for industrial applications ever since their introduction in 1969. Extremely low leakage and wear characterize non-contacting face seals. Because of that, there have been continuous efforts made by investigators in aerospace to develop non-contacting face seals for large diameter gas turbine engines, where the potential payoff is very high. As it turns out, the application of non-contacting face seals in gas turbine engines is much more demanding than in industrial applications

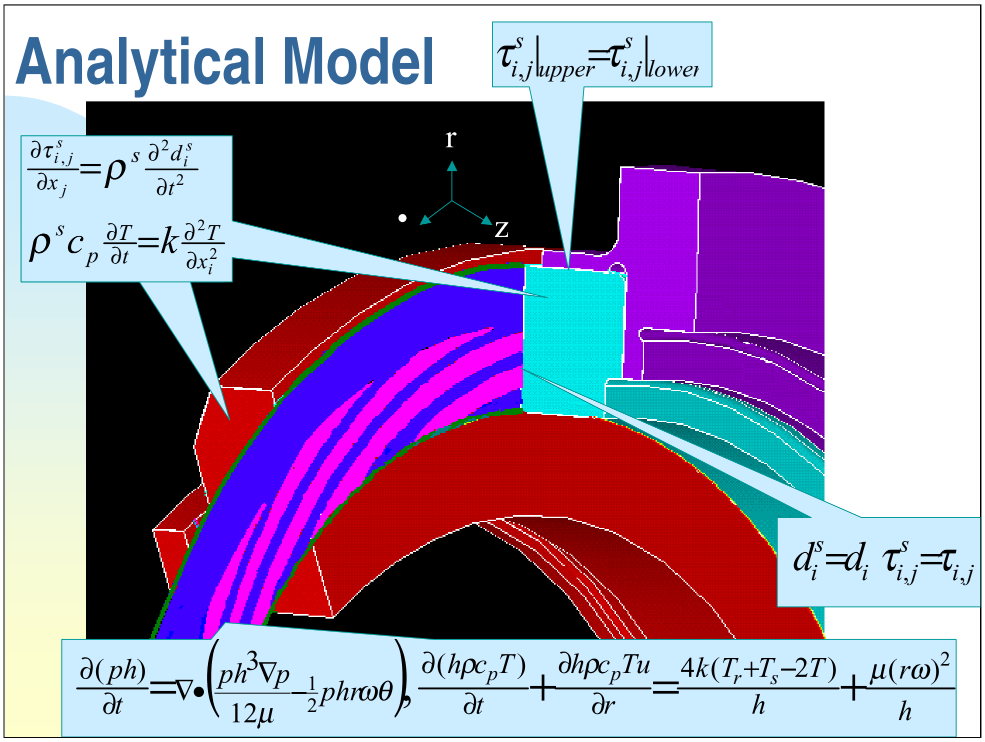
There are two major difficulties associated with using face seals for high rotational speed and large shaft diameter turbomachines. First, controlling the flatness of the seal faces is very difficult because of the size. Second the seal faces of both the rotor and stator can cone in either inward or outward direction due to the large thermal and pressure effects. A negative deflection causing a divergent flow path can be disastrous for a standard hydrodynamic face seal, since it tends to cut off the flow of gas into the region between the faces. With standard hydrodynamic face seals the deflection is expected to be much larger than the film thickness that the face seal runs on. Large positive coning can also result in failure for large diameter face seals because the resulting weak film stiffness increases the chance of face contact



Analytic Design Tools Development

- General seal configurations.
- Conjugate heat transfer analysis.
- Deflections by pressure and centrifugal loading.
- Face coning in composite design.
- Transient analysis of seal responses to the changes of operational conditions.

Tight R&D budget prevents systematic evaluation of large diameter spiral groove face seal designs in laboratory. More and more product development relies on theoretical analysis. Rig test is only a validation tool, other than a development vehicle as it used to be.



Analytical Model

$\tau^{s}_{i,j}\big|_{upper}=\tau^{s}_{i,j}\big|_{lowe}$

$$\frac{\partial\tau^{s}_{i,j}}{\partial x_j}=\rho^{s}\frac{\partial^2 d^{s}_i}{\partial t^2}$$

$$\rho^{s}c_p\frac{\partial T}{\partial t}=k\frac{\partial^2 T}{\partial x_i^2}$$

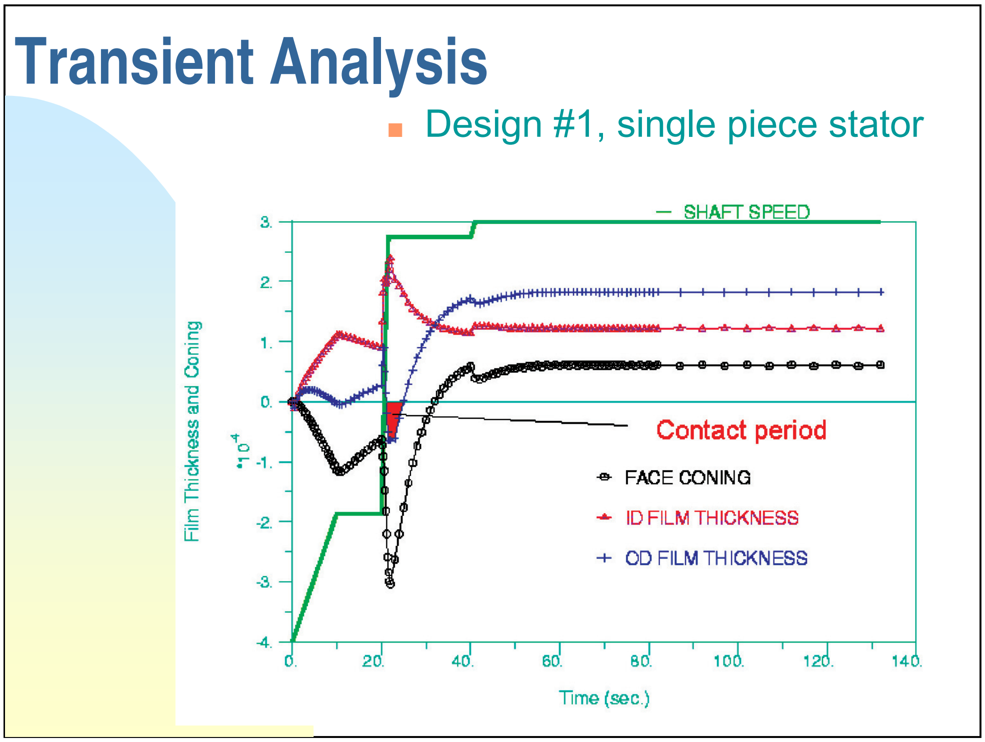
$d^{s}_i=d_i \quad \tau^{s}_{i,j}=\tau_{i,j}$

$$\frac{\partial(ph)}{\partial t}=\nabla\bullet\left(\frac{ph^3\nabla p}{12\mu}-\frac{1}{2}phr\omega\theta\right),\quad \frac{\partial(h\rho c_p T)}{\partial t}+\frac{\partial h\rho c_p T u}{\partial r}=\frac{4k(T_r+T_s-2T)}{h}+\frac{\mu(r\omega)^2}{h}$$

The seal equation, the Reynolds equation, is solved in the (r, •) plane to obtain pressure and leakage. It is inserted into a commercial finite element program, ADINA, which is capable of transient dynamic analysis of structural deflections due to thermal stress, pressure, and centrifugal force in a full three-dimensional way. But In most cases, the energy equation and solid structural equations are solved in (r,z) plane as axisymmetrical problems. Therefore the whole system of equations is quasi-three dimensional. The pressures from seal equation are circumferentially averaged before they are passed to structural analysis.

Application Examples

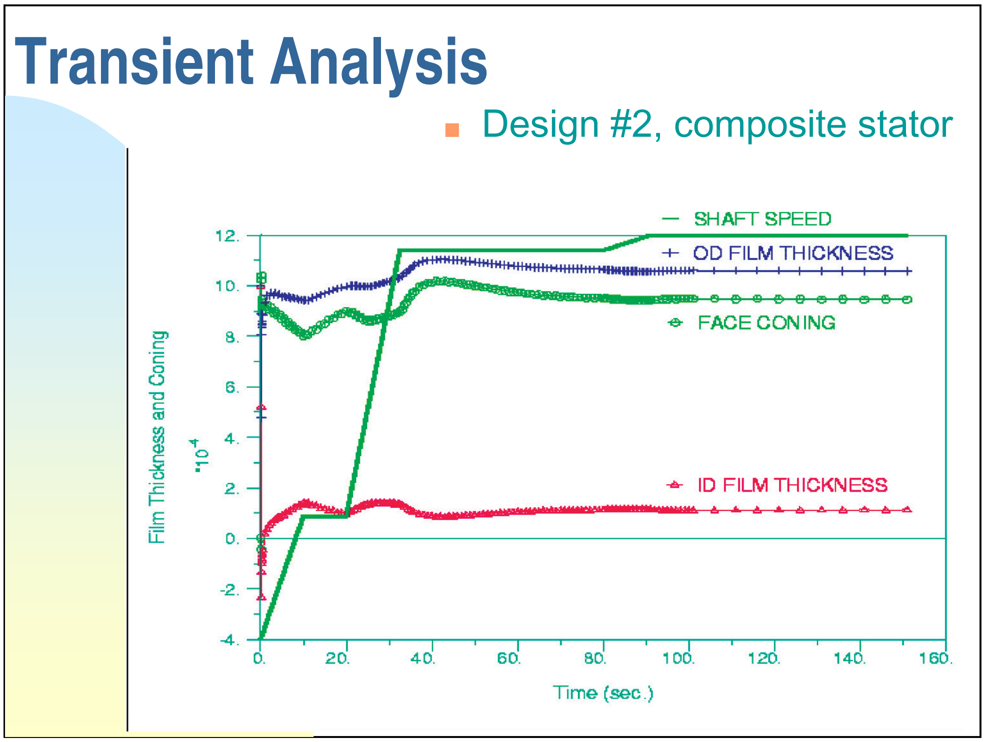
- 6” diameter, over 19,000 rpm
- 5” diameter, over 14,000 rpm



Transient Analysis

- Design #1, single piece stator

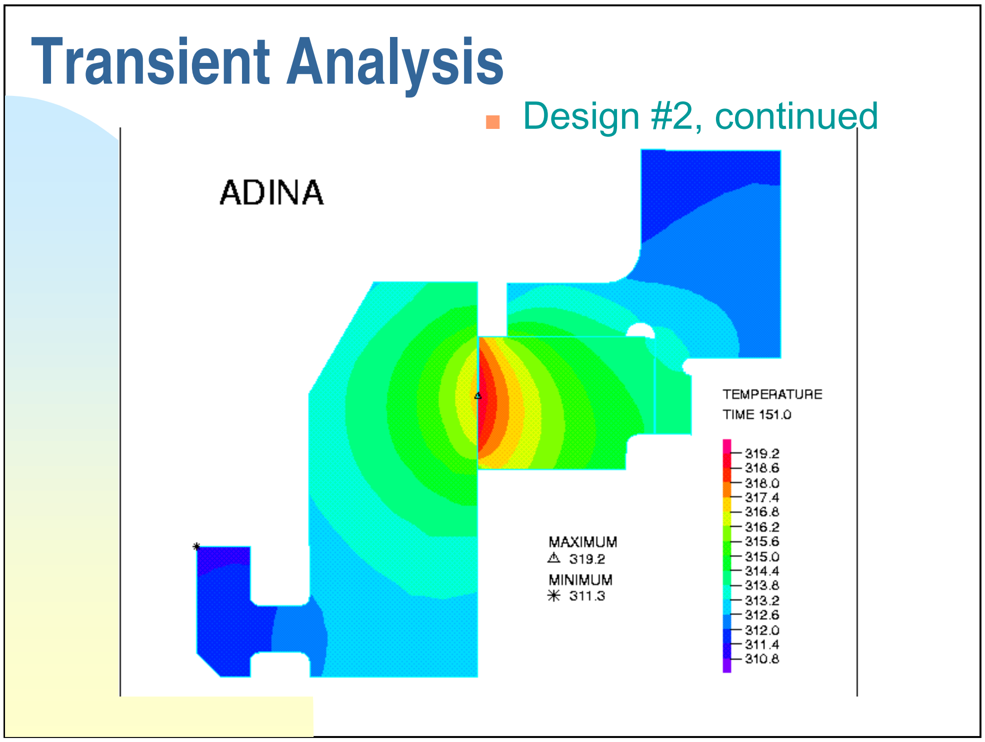
The seal is OD pressurized, with hot oil in the OD side and cold air in the ID side. The seal is designed to pump low-pressure air into high-pressure oil side. The above image shows the history of shaft speed, seal film thickness at ID and OD, as well as face coning. It is interesting to find out that the seal experiences momentary OD contact during the shaft acceleration, in contrast to usual ID contact for most seals in transient phase. Careful examination of results reveals that this phenomenon is due to the faster temperature increase in OD than that in the ID. As we know that the heating from the seal face rubbing makes the seal faces cone positively, leading to ID contact. But for this seal, the heating from oil in the OD overcomes the seal face heating and leads to OD contact.



Transient Analysis

- Design #2, composite stator

A revise composite stator design saves axial space and also provides favorable face coning for outward-pumping face seal



Transient Analysis

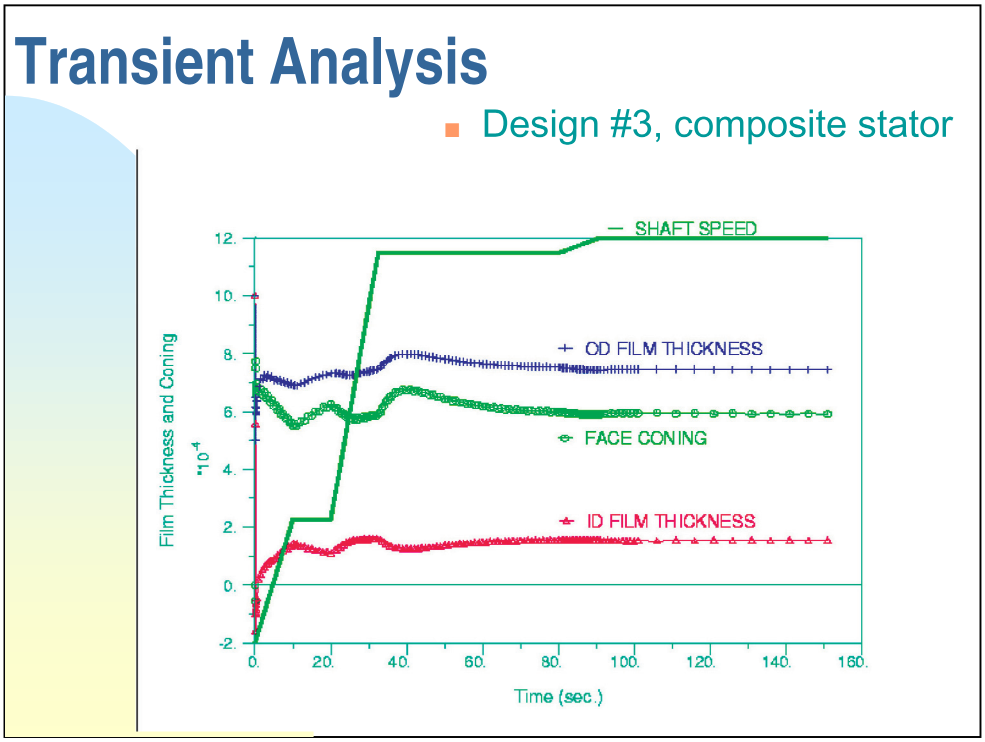
- Design #2, continued

Temperature contours at final steady state

Transient Analysis

- Design #2, continued

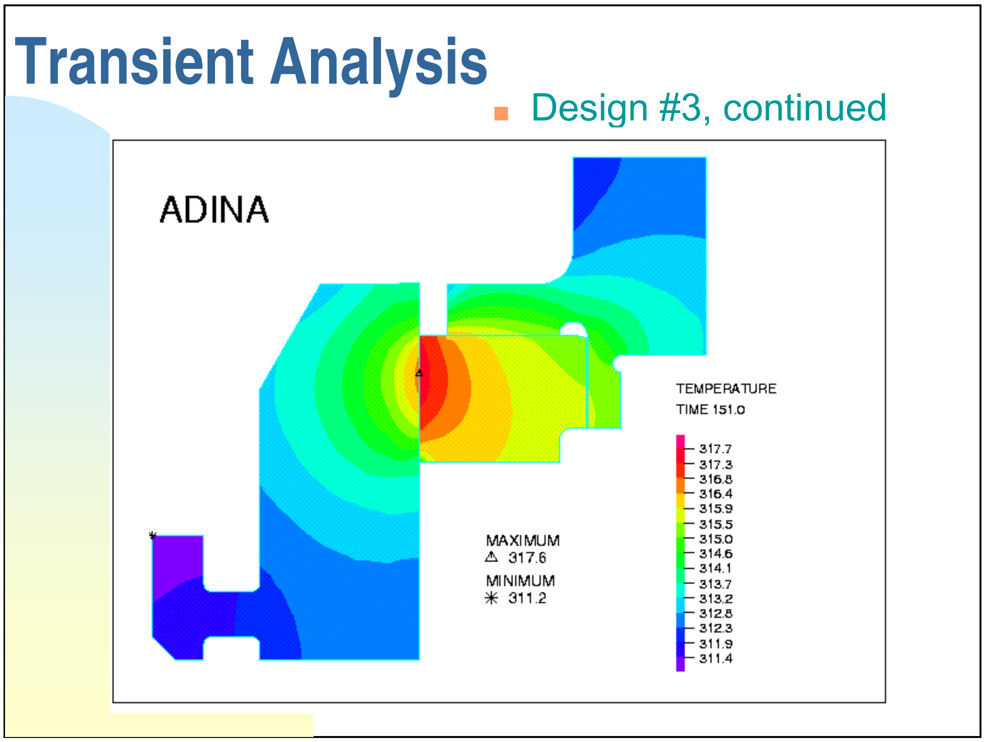
This animation shows a magnified view of seal deflection due to thermal and pressure effects. Colors are shown for temperature profiles.



Transient Analysis

- Design #3, composite stator

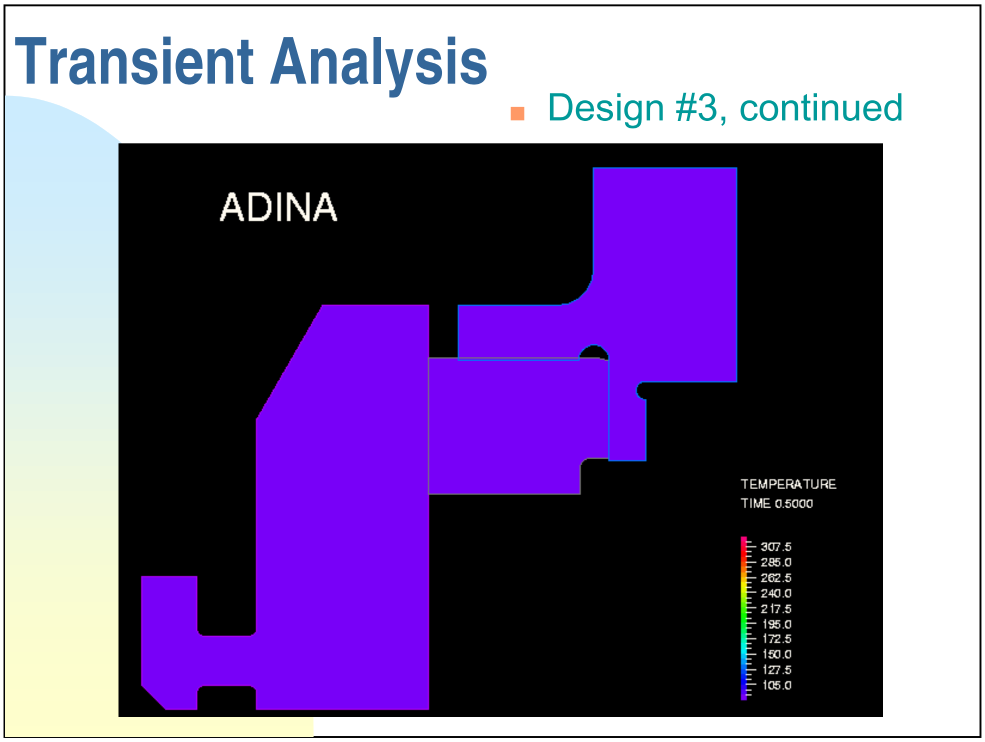
A further revised design reduces face coning provide more uniform gas film.



Transient Analysis

- Design #3, continued

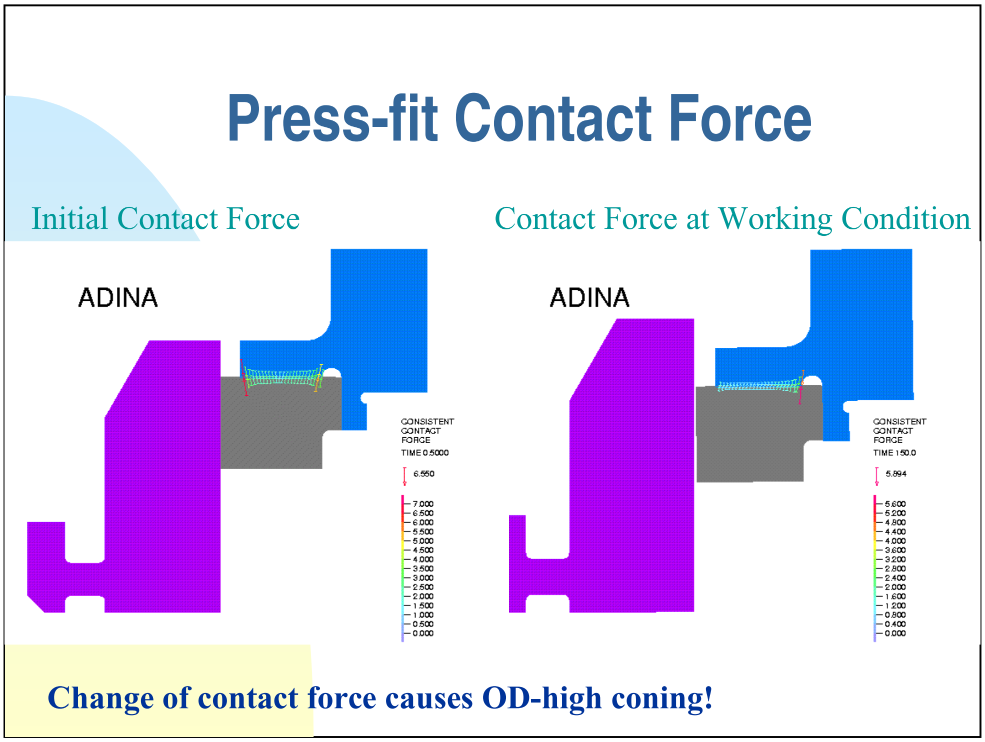
Final temperature contours at steady state.



Transient Analysis

- Design #3, continued

Animation shows the magnified seal deflection during the transient. Colors are shown for temperature contours.



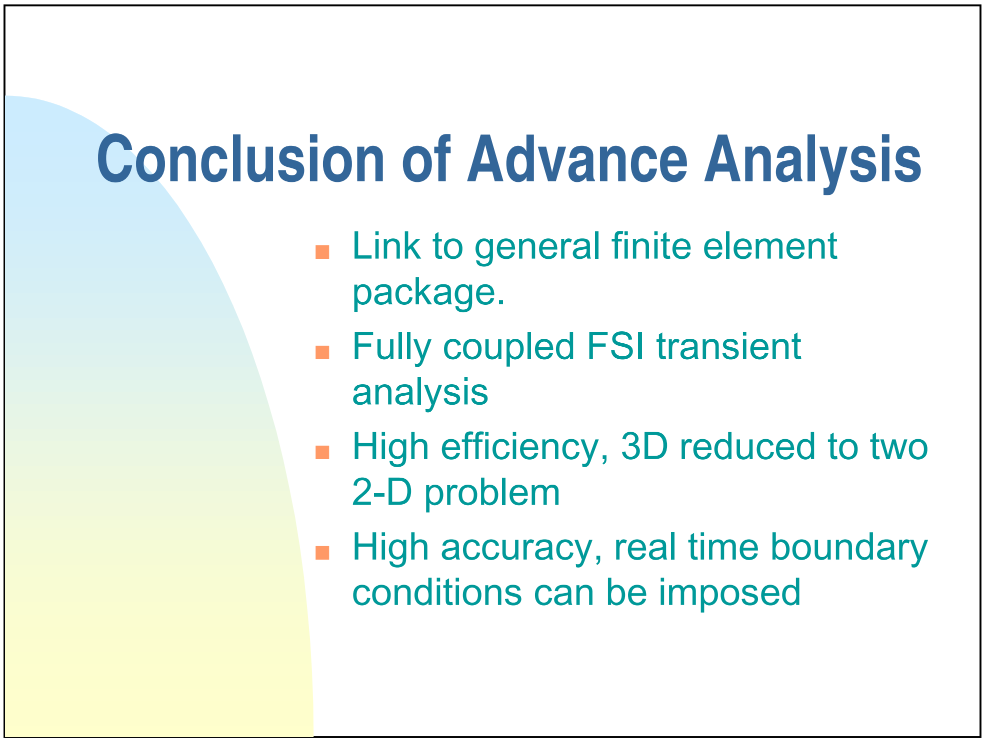
Press-fit Contact Force

Initial Contact Force

Contact Force at Working Condition

Change of contact force causes OD-high coning!

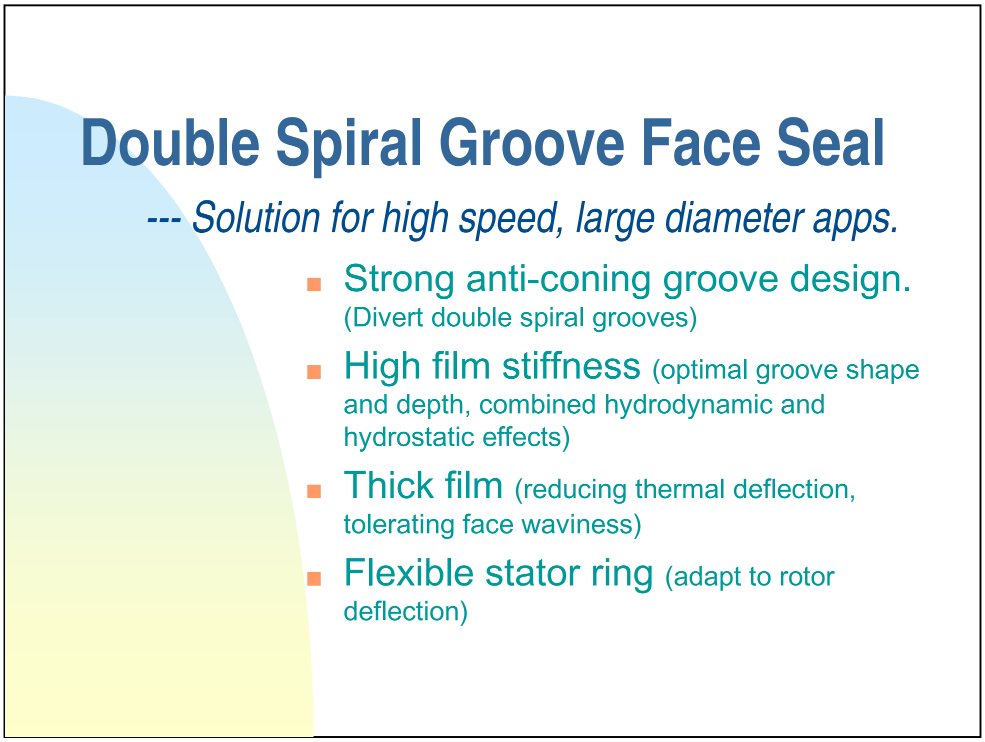
Thermal relief of contact stress in composite stator causes the stator face to cone in OD-high, which is opposite to usual thermal face coning. The amount of face coning is dependent on material differences and initial interferential fit. Those parameters can be used to achieve the desirable face coning.



Conclusion of Advance Analysis

- Link to general finite element package.
- Fully coupled FSI transient analysis
- High efficiency, 3D reduced to two 2-D problem
- High accuracy, real time boundary conditions can be imposed

The new design tool, incorporating a commercial finite element program ADINA, is capable to model transient fluid-structure interaction inside a general seal configuration, enabling designer to predict seal responses to speed, temperature and pressure changes in a time-dependent manner. Therefore, through evaluation of influences of different seal parameters, the application envelopes of conventional spiral groove face seal can be extended to larger diameter, higher speed range.

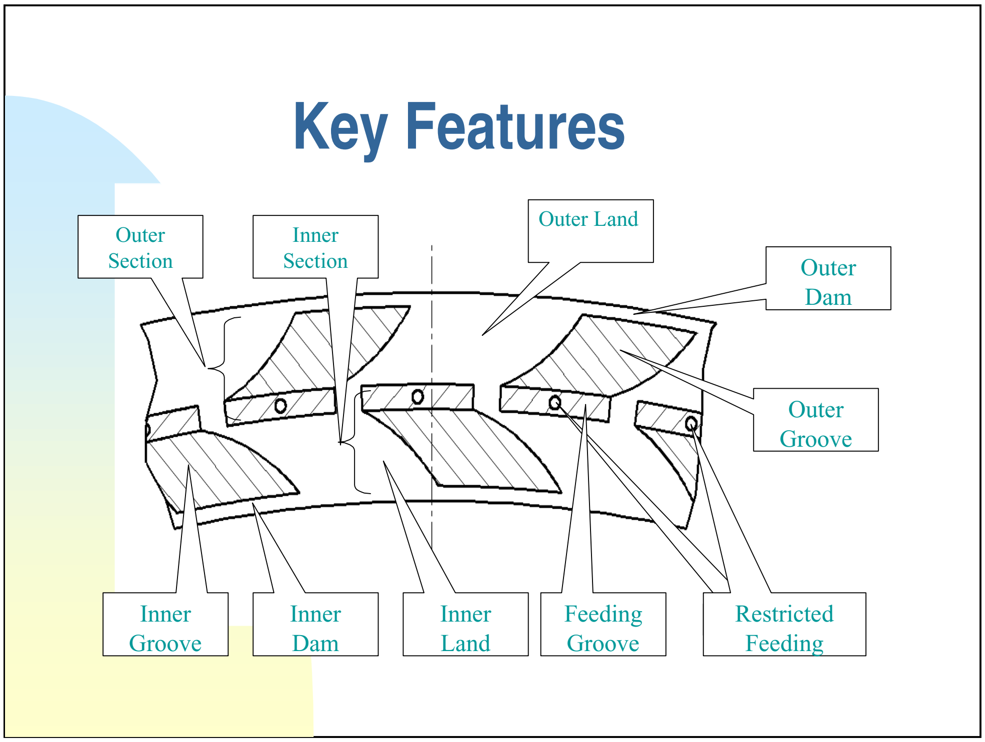


Double Spiral Groove Face Seal

--- Solution for high speed, large diameter apps.

- Strong anti-coning groove design. (Divert double spiral grooves)
- High film stiffness (optimal groove shape and depth, combined hydrodynamic and hydrostatic effects)
- Thick film (reducing thermal deflection, tolerating face waviness)
- Flexible stator ring (adapt to rotor deflection)

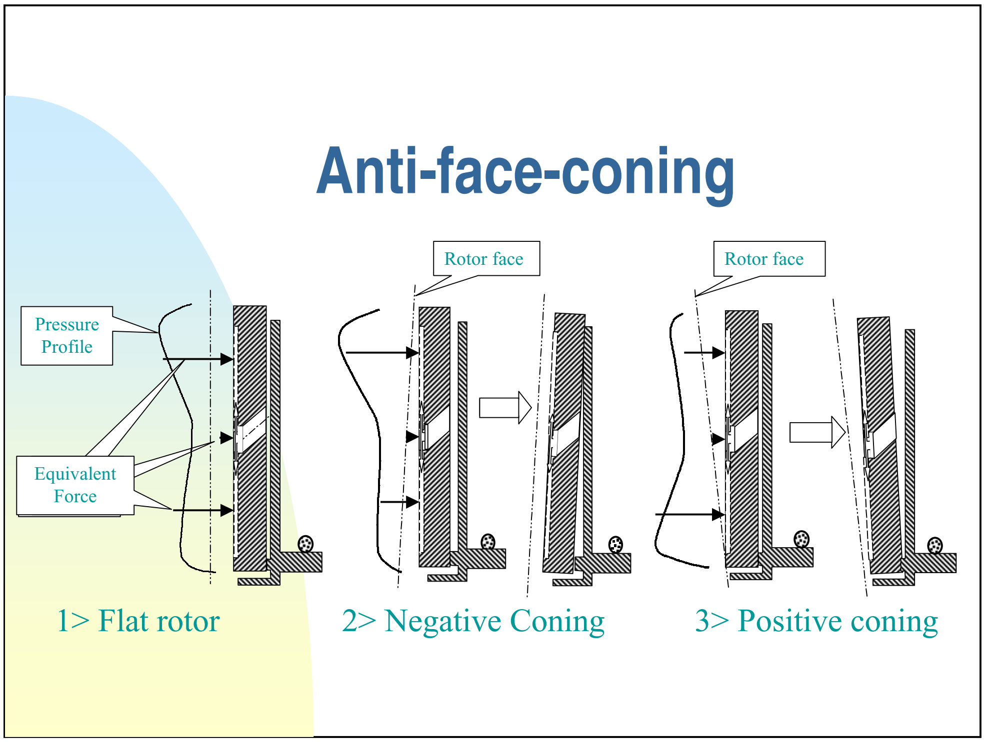
In case that conventional spiral groove face seals cannot meet operational requirements in large turbine engines, a new double-spiral groove face seal has been designed. Two sets of seal sections and single center feed groove are used for the seal face. This configuration is able to provide higher film stiffness giving the stator more power to adapt to the deflection of the rotor.



Key Features

Two basic strategies were used to achieve the high film stiffness, high coning restorability of the face seal. First, the seal face was redesigned to have strong anti-coning capability. A unique feature of the new seal is that the film will not be pinched under any condition. Secondly, orifice restrictive effects of the feeding holes are consciously used to enhance film stiffness whenever the film thickness in one or both seal sections is too large to render the hydrodynamic force effective.

T the seal face consists of two sets of seal sections and a set of either segmented or connected deep grooves. Each seal section contains a hydrodynamic section, marked with an alternating groove and land pattern, and a dam section near the face edges. System fluid or gas, which is allowed to leak in a small amount, is fed into the middle feed grooves of the face seal through restricted feeding holes, and pumped inward and outward simultaneously by specially designed grooves on the stator and/or rotor face. This allows the seal to work through harsh conditions of severe face deflection. Since the fluid enters from the center, face coning will never cut off fluid from getting into the seal face.



Anti-face-coning

1> Flat rotor

2> Negative Coning

3> Positive coning

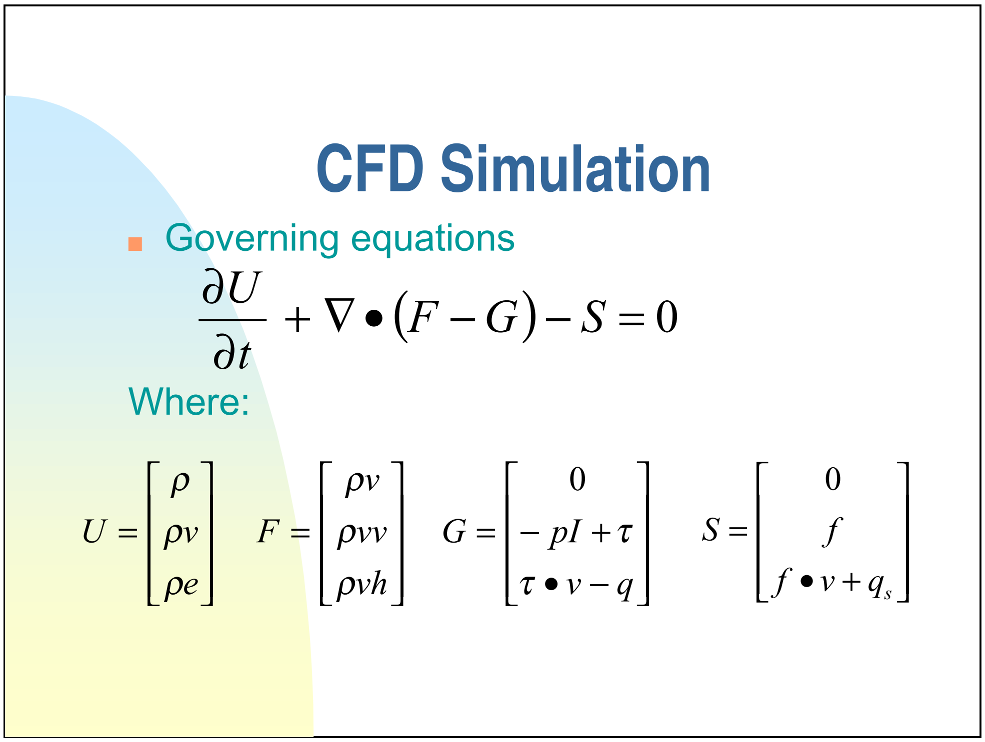
When the rotor face deflection causes negative coning, the outer seal section is working in a convergent film (refer to the flow direction). That makes the groove work more effectively to create higher pressure in the hydrodynamic section. Therefore, the outer seal section generates more positive moment to open up the clearance at outer diameter. Meanwhile, the inner seal section is working at a divergent film. That reduces the hydrodynamic effects of the grooves. Less pressure, and therefore less negative moment, is generated by the inner seal section. The net increase of positive moment causes the stator ring to cone positively and form a uniform film thickness.

When the rotor face deflection causes positive coning effect, the outer seal section is working in a divergent film. That makes the groove work less effectively to create a high-pressure zone in the hydrodynamic section. Therefore, the outer seal section generates less positive moment. Meanwhile, the inner seal section is working at a convergent film, which increases the hydrodynamic effects of the grooves. Higher pressure, and therefore larger negative moment, is generated by the inner seal section to open up the clearance at inner diameter. The net increase of negative moment causes the stator ring to cone negatively and form a uniform film thickness.

Restrictive Orifice Design

- Purposes:
  - Control leakage
  - Extend the range of high film stiffness
  - Improve film stiffness
- Calculation of effectiveness
  - Empirical formula
  - Detailed CFD simulation
  - Integrated into double-spiral groove seal design code

The restricted orifice design is not only good at increasing coning film stiffness, but also effective to improve axial film stiffness. The pressure between seal faces is not only dependent on the hydrodynamic effects of spiral grooves, which is a function of film thickness, but is also affected by the hydrostatic effects of restricted orifices. The pressure in the feeding groove is strongly dependent on the flow amount through the feeding holes. As the film thickness increases, the pressure drop through the feeding hole increases. The opening force will drop as a result of lower pressure in the seal faces. At very thin film, the double-spiral grooves alone can generate enough film stiffness. The restricted feeding holes can be designed in such a way that it is most effective at relatively thick film, so that the seal has large film stiffness in a wide range of film thickness. In other words, once the seal faces open up, hydrodynamic effect from spiral groove diminishes gradually; the hydrostatic effect kicks in to continue the strong dependency of opening force on film thickness.



CFD Simulation

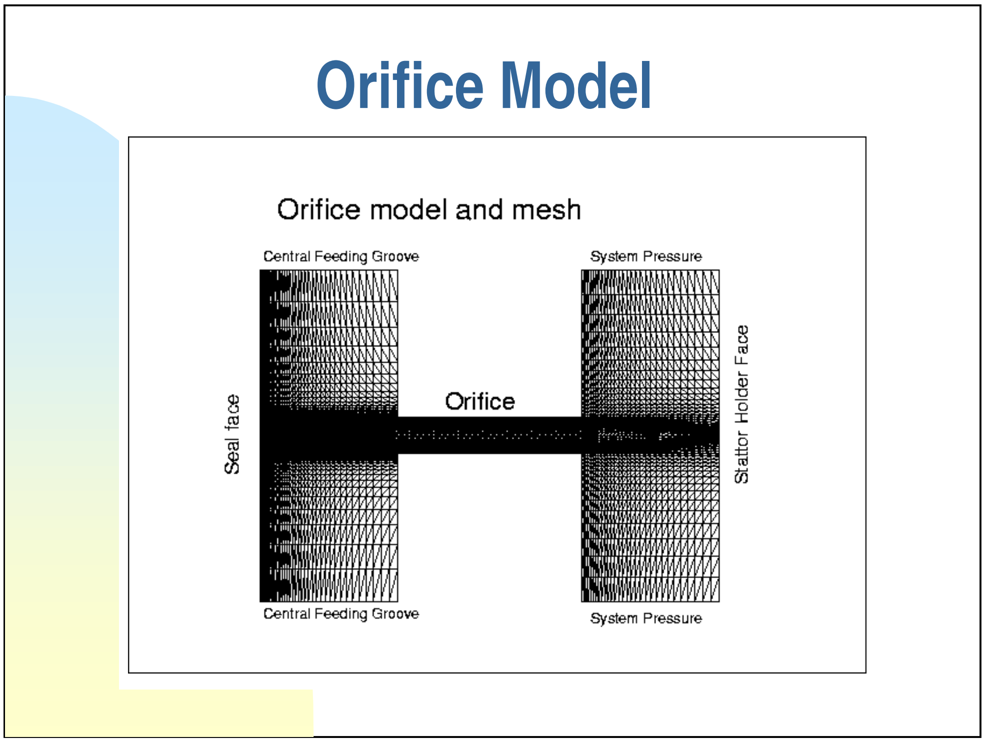
- Governing equations

$$\frac{\partial U}{\partial t} + \nabla \bullet (F - G) - S = 0$$

Where:

$$U = \begin{bmatrix} \rho \\ \rho v \\ \rho e \end{bmatrix} \quad F = \begin{bmatrix} \rho v \\ \rho v v \\ \rho v h \end{bmatrix} \quad G = \begin{bmatrix} 0 \\ -pI + \tau \\ \tau \bullet v - q \end{bmatrix} \quad S = \begin{bmatrix} 0 \\ f \\ f \bullet v + q_s \end{bmatrix}$$

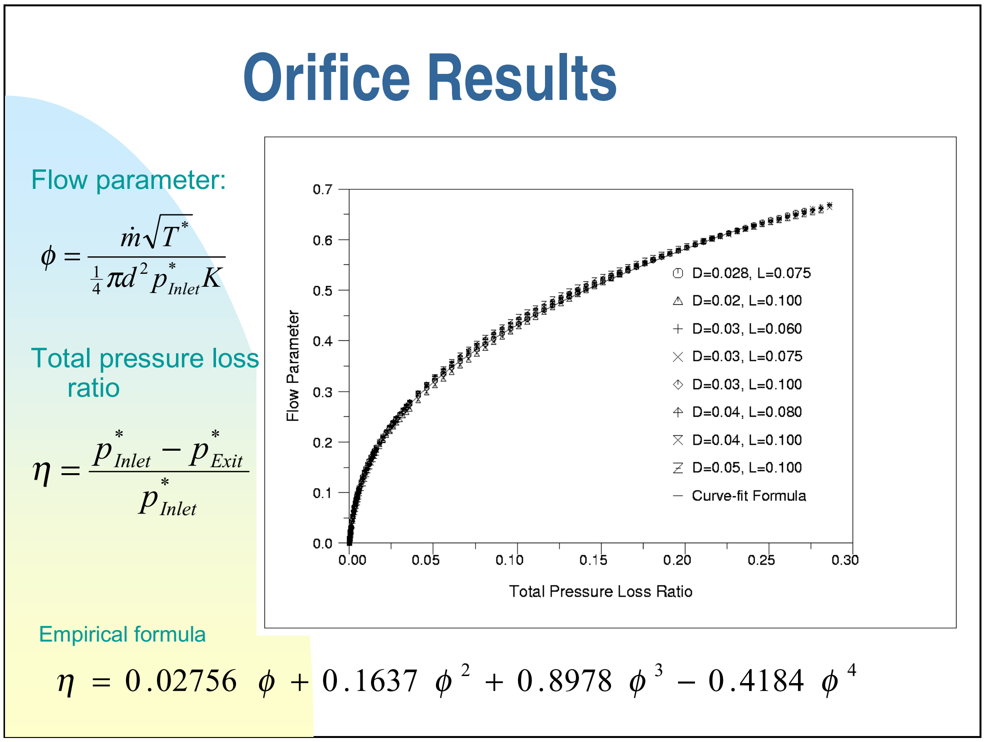
The orifice restrictive effects of the feeding holes play an important role in performance of the new seal. A simple formula for pressure drop over orifice can be found in current published literature, however a more accurate solution is required owing to the significance of pressure drop over the feeding hole at maximum operating condition. A CFD model was built to find the pressure drop as a function of flow rate.



Orifice Model

Orifice model and mesh

ADINA, a general fluid and solid finite element analysis program, is used to solve the problem. Because of the rotational surface at the flow exit, the flow is actually three-dimensional. Here each feeding hole and the feeding groove at the exit is modeled approximately as an axisymmetric case. Figure 6 shows the whole cut plane. But only half of the domain needs to be solved.



Orifice Results

Flow parameter:

$$\phi = \frac{\dot{m}\sqrt{T^*}}{\frac{1}{4}\pi d^2 p^*_{Inlet} K}$$

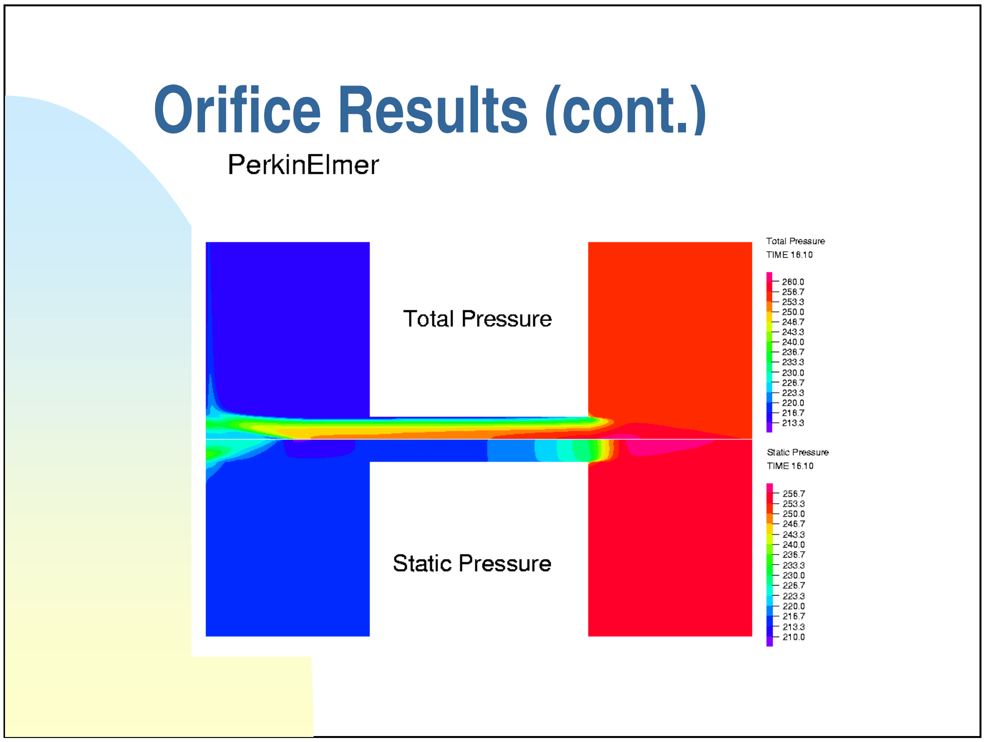
Total pressure loss ratio

$$\eta = \frac{p^*_{Inlet} - p^*_{Exit}}{p^*_{Inlet}}$$

Empirical formula

$$\eta = 0.02756\ \phi + 0.1637\ \phi^2 + 0.8978\ \phi^3 - 0.4184\ \phi^4$$

The purpose of this analysis is to find out the relationship between pressure drop and orifice flow rate for various orifice dimensions. Hopefully, their relationship can be expressed in a single formula and in terms of non-dimensional variables. Then the formula is plugged into the seal design code to obtain fast solutions. First, all the data for all possible choice of orifice dimensions are plotted together. In terms of flow parameter and pressure drop ratio, we found that they closely form a curve as shown in Figure 8. Therefore a curve-fitting program was used to approximate the data in a polynomial form.



Orifice Results (cont.)

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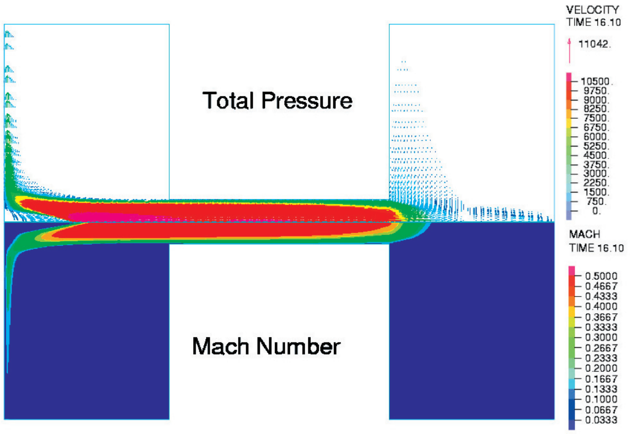
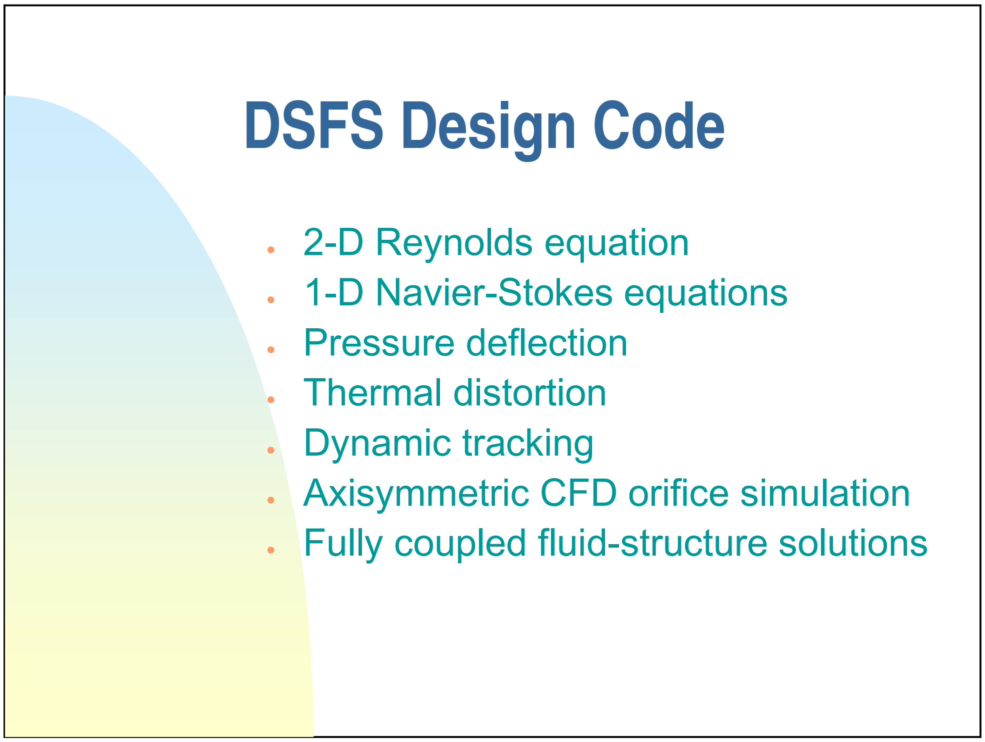
This plot shows the total pressure contours on the upper half plane, and static pressure contours on the lower half plane for the orifice of current design at high pressure conditions. It easily can be seen that there are major total-pressure changes near the entrance and in the region of stagnation point on the rotor seal face. At large pressure difference conditions, the air stream speeds through the orifice and keeps straight ahead until it hits the rotor seal face. Because of high momentum of the flow, the sudden expansion at the orifice exit does not cause the air stream to spread sideward and slow down. The core flow is only slowed down when it hits the rotor seal face and spreads outward, causing great total-pressure loss due to large shear stresses.

Orifice Results (Cont.)

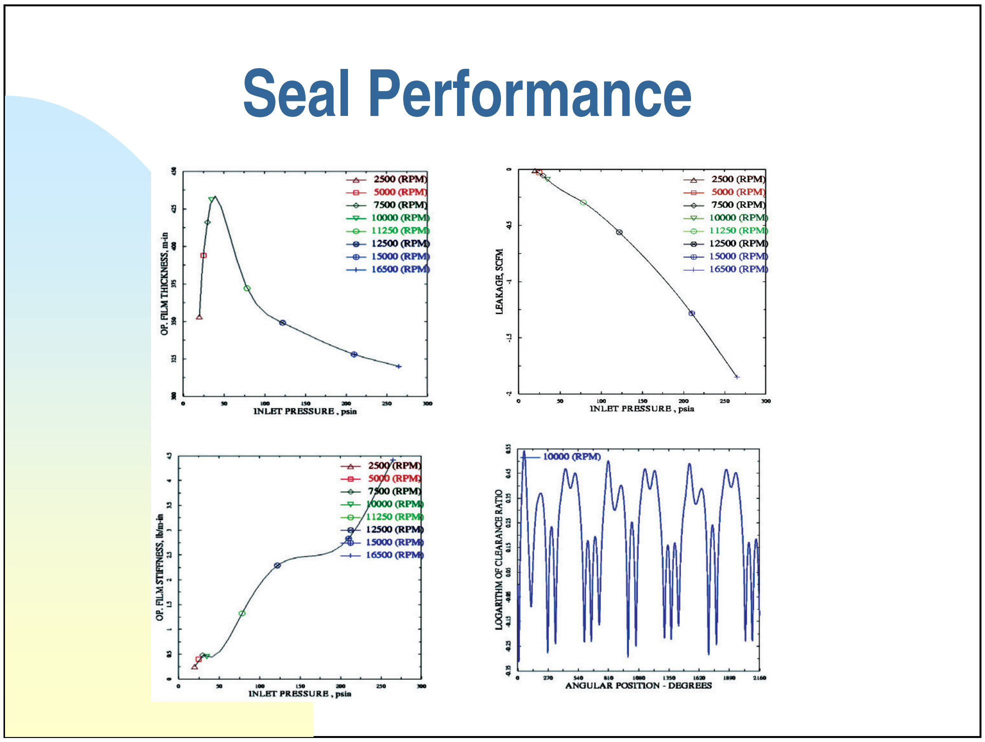




DSFS Design Code

- 2-D Reynolds equation
- 1-D Navier-Stokes equations
- Pressure deflection
- Thermal distortion
- Dynamic tracking
- Axisymmetric CFD orifice simulation
- Fully coupled fluid-structure solutions

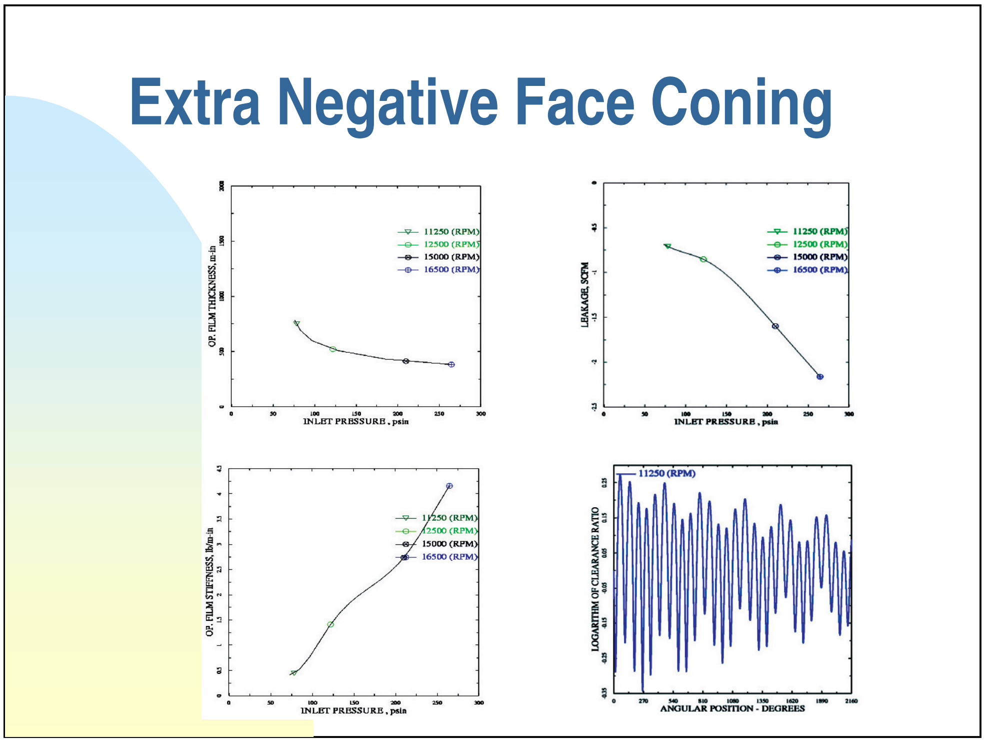
The computer code for double-spiral groove face seal analysis and design is based on a well-calibrated gas seal design code for conventional spiral groove seals, which was developed by James Gardner a decade ago, and has been enhanced greatly by Prit Basu and Zack Williams. The first author added an integrated graphics package to it and built it into a web-based application program for ease of access within the local intranet. More advanced iterative solution methods are used to improve the efficiency and the ability to cope with the new geometric configuration was implemented by the first author.



Seal Performance

With help of the design code developed for the new configurations, we are able to design the seal with outstanding performance at various rotor face coning conditions. First of all, the new seal was found quite insensitive to material properties. That is the most desirable characteristics since the choice of high temperature and tribological compatible materials is very limited. Second, the new seal works very well for all expected rotor face coning.

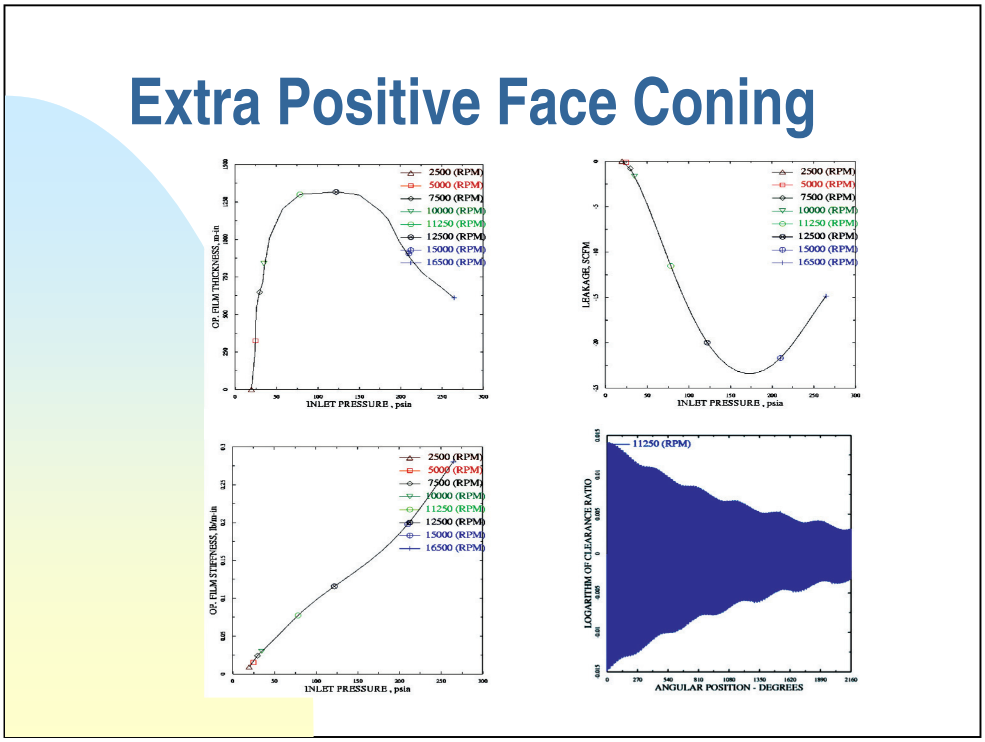
The strong capability of anti-coning of the new seal enables it to work under conditions of severe positive and negative rotor face coning. The seal can tolerate 0.010 inches of positive and 0.004 inches of negative face coning for all speeds higher than 2500 rpm. For high-pressure and high-speed conditions, the seal can deal with rotor face coning more than 0.010 inches of negative face coning. Negative face coning is usually deadly for single-spiral groove face seal. But for the divert double-spiral groove seal, it is no longer a problem.



Extra Negative Face Coning

If the rotor face has extra coning from either transient effects or some other unforeseen reasons, the seal is still able to work well.

If the rotor face has an extra negative coning of negative 0.010 inches (about 0.6 degrees), there exists OD contact for operating pressure difference of less than 50 psi. Extra rotor face coning of 0.010 inches is just too much for the stator ring to react at low speed and low pressure. But for higher pressure and speed, the seal works quite well.



Extra Positive Face Coning

Positive face coning is usually easier to deal with for OD-pressurized seal. The new seal advances to a new height in this aspect. As shown in the following plots, the seal works under all speed conditions larger than 2500 rpm with extra rotor face coning of positive 0.010 inches.

With sufficient margins, the seal can operate successfully under extra rotor face coning conditions ranging from –0.004 to 0.004 inches.

Summary for Double-spiral Groove Face Seal

- The seal:

  Excellent face coning recovery capability.

  Simple and the cost-effective.

  High film stiffness, robust.
- The computer code.

  Pressure deflection.

  Thermal distortion.

  Hydrodynamic effects of spiral grooves.

  Restrictive orifice effects.