DYNAMICS OF FACE SEALS FOR HIGH SPEED TURBOMACHINERY

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ABSTRACT

Face seals in rocket engine fuel and oxidiser turbopumps have been the subject of intense investigation for over 25 years. Whilst advances have been made in the understanding of thin film lubrication between seal faces; valuable data has been produced on the friction and wear of material pairs in cryogenic environments; pioneering work has been done on the effect of lubricant phase change in seals; and many improvements have been made in mechanical seal design, relatively superficial attention has been given to the vibrational dynamics of face seals in high-speed turbomachinery.

BHR Group Ltd. (formerly BHRA) has recently completed the first stage of a study, commissioned by the European Space Agency, to investigate this area. This has involved the development of a two-dimensional adiabatic, turbulent lubrication model for thick gas film applications, the production of an integrated mathematical model of gas seal vibrational dynamics for thin film applications, implementation in software, the undertaking of an experimental programme to validate software against variations in operating conditions and design variables, and suggestions for improved seal design.

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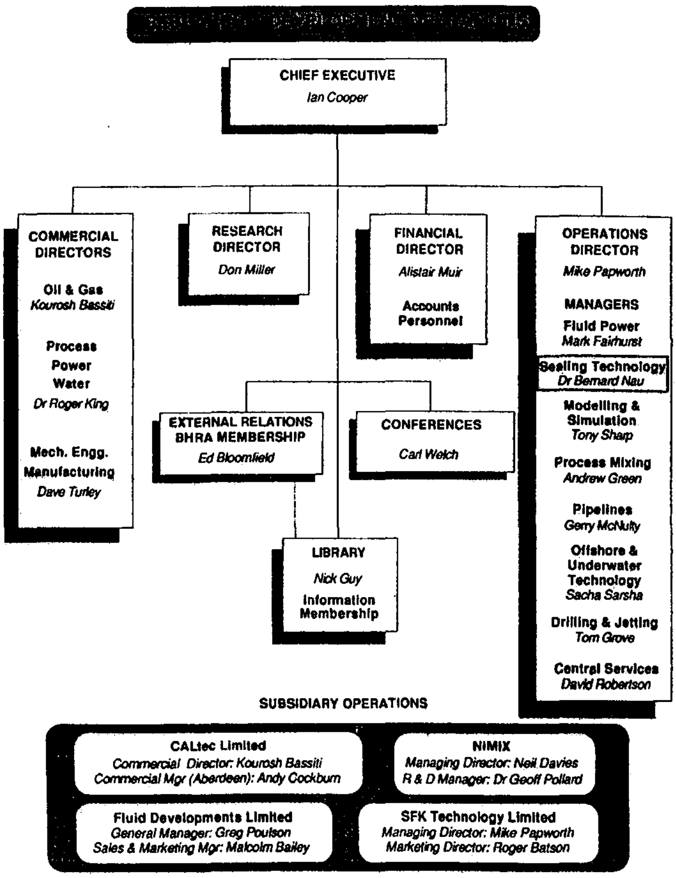

CALtec Limited - Oil & gas
NIMIX - Non-intrusive mixing equipment
Fluid Developments Limited - Abrasive jet cutting equipment
SFK Technology Limited - Software development

CURRENT PROJECTS

- Applications
  - rotary
  - reciprocating
  - static
- Technology
  - experimental
  - analytical
  - design studies
- Organisation
  - direct contract
  - consortium + Dept. of Trade and Industry
  - consortium + EEC
  - independent consortium

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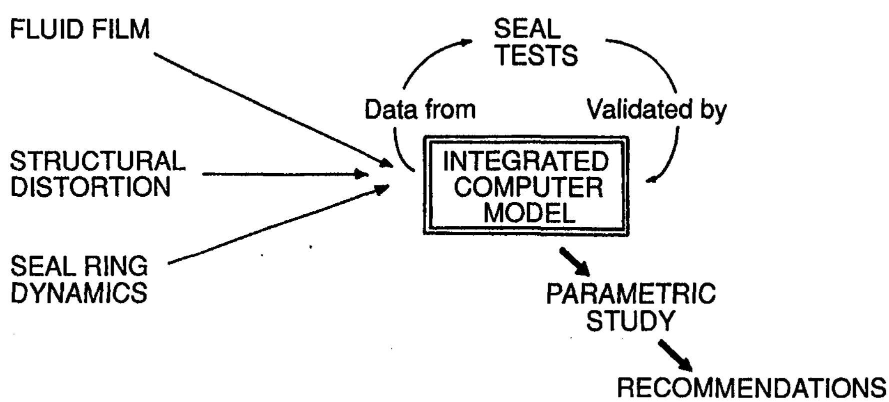
FLUID SEALING TECHNOLOGY

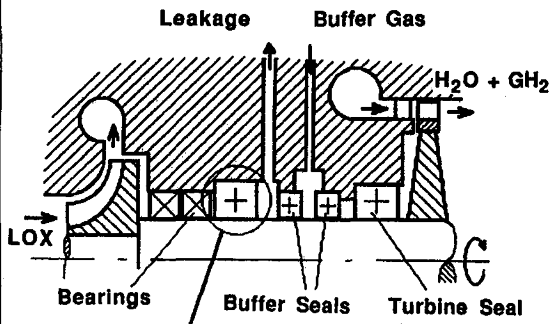
Independent facilities and expertise:-

- Seal Analysis:- thermal, mechanical, lubrication
- Seal testing:- oil, water, gas, cryogens, contaminants
- Pump Loops:- oil, water, slurry, water/air
- Site Measurements:- fixed, portable
- Design Audit:- analysis, critical review
- Rig Manufacture:- design, build, modification

PROJECT STRUCTURE



TURBOPUMP SEALING

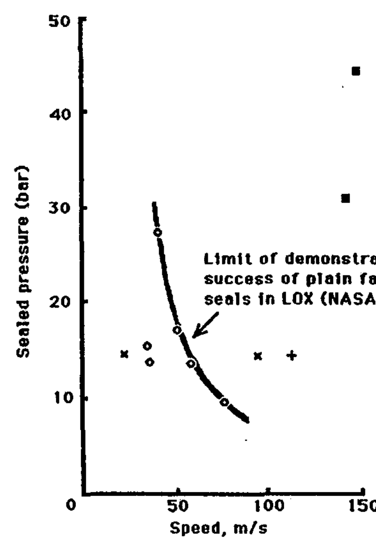


FACE SEAL OPTIONS:

- Plain
- Hydrostatic
- Self-acting

LITERATURE SURVEY

MAIN FINDINGS – SEALING PRACTICE



- ■ Self-acting face-seals in LOX (NASA evaluation)
- o Plain face-seals in LOX (demonstrated success)
- + Plain face seals in LH2 (Japanese test programme)
- × Plain face seals in GN2 (BHRG test programme)

LITERATURE SURVEY

MODELLING - FILM LUBRICATION ANALYSIS

Laminar, isothermal - Reynolds equation

- Liquid films - treatment of cavitation
- Gas films - grid design (adaptive, graded, etc.)
  - algorithm design (implicit, alternating, multigrid, 'interior co-location', etc.)

Turbulent lubrication - Hirs' bulk flow
- Constantinescu
- Ng & Pan

Non-isothermal (higher Mach number) compressible flow

- 1-D (radial) adiabatic model with radial taper and entrance effects - Zuk

Two-phase (boiling interface) films

- 1-D models
- Stability approached from consideration of equilibrium film thickness vs. opening force curves (i.e. not from dynamic analysis)

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LITERATURE SURVEY

MODELLING - DYNAMICS

- [K] and [C] matrices from fluid film analysis then dynamics as a separate problem
- Integrated analysis - fluid film forces and moments calculated at each timestep
- Excitation mechanisms
- Number of vibrational degrees-of-freedom
- Thermal and vibrational transients - 3 or 4 orders of magnitude difference in timescale - separate problem

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SCOPE OF MODEL

- Concentrate initially on DYNAMICS
- Gas seal assumed (2-D transient 2-phase prohibitive within commercial constraints)
- Transient structural distortions
- Full transient lubrication analysis
- Turbulent, adiabatic AND laminar, isothermal leakage flow
- Choked exit conditions catered for
- 4 vibrational degrees of freedom
- Mechanical damping

MATHEMATICAL MODEL

TURBULENT, ADIABATIC FLOW

Iterate round instantaneous equations

Radial and circumferential velocities:

$$u_r = -\frac{G_r h^2}{\eta}\left(\frac{\partial p}{\partial r} - I_r\right) \quad \text{where} \quad I_r = \frac{\rho u_\theta^2}{r}$$

and **(optional inertia)**

$$u_\theta = -\frac{G_\theta h^2}{\eta}\frac{\partial p}{r\partial \theta} + \frac{r\omega}{2}$$

G_r and G_θ from Hirs' bulk flow turbulent lubrication theory (or = 1/12, laminar)

Given in terms of Reynolds numbers "as seen by" rotor and stator (different)

Shear stresses from these Reynolds numbers

2-D adiabatic energy equation relates pressure to circumferential shear stress for density at current timestep

Timestepping

Substitute u_r and u_θ in continuity equation as 'knowns' and find density at next timestep

Use film thickness, h, at mid-timestep throughout procedure

MATHEMATICAL MODEL

LAMINAR, ISOTHERMAL FLOW

Compressible Reynolds equation with ideal gas assumption

$$\frac{\partial}{\partial r}\{ph^3\frac{\partial p}{\partial r}\} + \frac{\partial}{r\partial\theta}\{ph^3\frac{\partial p}{r\partial\theta}\} - 6\eta r\omega\frac{\partial}{r\partial\theta}\{ph\} = 12\eta\frac{\partial}{\partial t}\{ph\}$$

Time discretisation:

$$\frac{\partial}{\partial r}\{p_n h^3_{n+\frac{1}{2}}\frac{\partial p_{n+1}}{\partial r}\} + \frac{\partial}{r\partial\theta}\{\frac{h^3_{n+\frac{1}{2}}}{2}\frac{\partial p^2_n}{r\partial\theta}\} - 6\eta r\omega\frac{\partial}{r\partial\theta}\{p_n h_{n+\frac{1}{2}}\}$$
$$= 12\eta\left\{\frac{(p_{n+1}+p_n)}{2}\frac{\partial h}{\partial t}\Big|_{n+\frac{1}{2}} + h_{n+\frac{1}{2}}\frac{(p_{n+1}-p_n)}{\delta t}\right\}$$

No energy equation required

Velocities and shear stresses from pressure gradients

- **Check exit Mach number distribution for condition**

  $M < \frac{1}{\sqrt{\gamma}}$ **(isothermal flow)**

- **Check Reynolds number distribution for condition**

  $Re < Re_{critical}$ **(laminar flow)**

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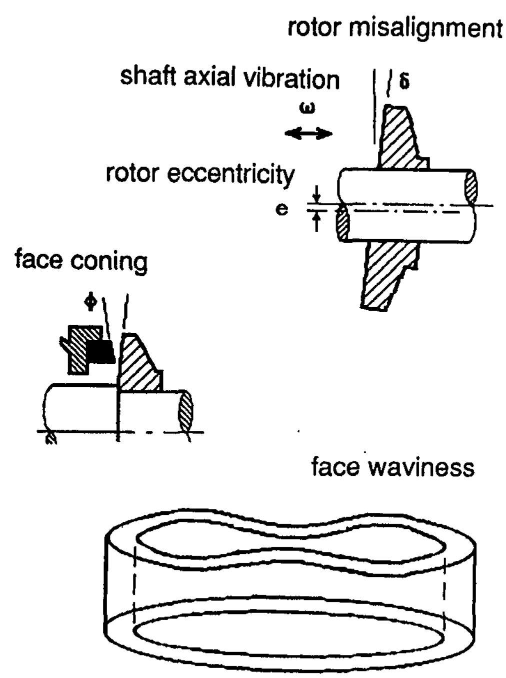
EXCITATION MECHANISMS



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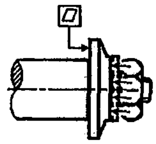
MATHEMATICAL MODELLING

FACE CONING

Sources:

- Pre-lapped taper and clamping forces



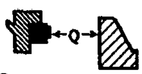
- Bellows load

- Rotor centrifugal inertia

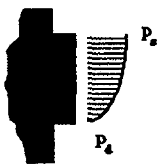
- Change in ambient temperature for seal ring assemblies

- Sealed pressure differential

- Interfacial heat generation



- Interfacial pressure distribution



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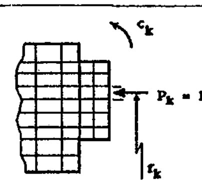
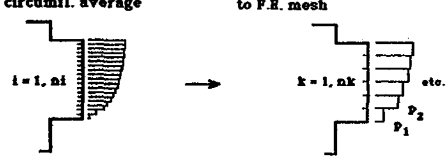
MATHEMATICAL MODELLING

INTERFACIAL PRESSURE CONING

1. F.E. Analysis



2. Pressure distn. over F.D. grid - circumfl. average → uniform pressure over annular rings corresponding to F.E. mesh



3. Use influence coefficients to calculate face coning

$$\Psi = \sum_k c_k \cdot P_k$$

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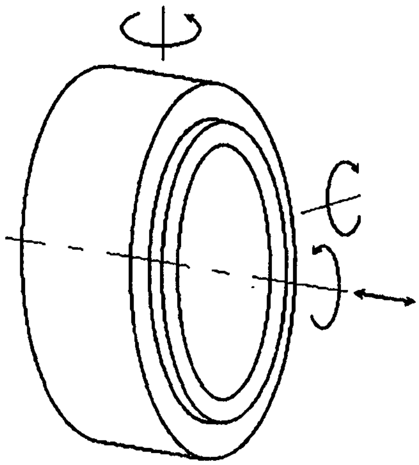
VIBRATIONAL DEGREES OF FREEDOM



MATHEMATICAL MODELLING

DYNAMICS

Equations of motion time-discretised by Newmark's method:

$$[M]^{-1}\{F\}_{n+1} = \frac{4}{\delta t^2}\left(\{X\}_{n+1} - \{X\}_n\right) - \frac{4}{\delta t}\{\dot{X}\}_n - \{\ddot{X}\}_n$$

$[M]^{-1}\{F\}_{n+1} \rightarrow \{\ddot{X}\}_{n+1}$

$$\{F\}_{n+1} = [K_L]\{X\}_{n+1} + \{F_{NL}\}_{n+1}$$

$[K_L]$: linear stiffness (bellows)

$\{F_{NL}\}_{n+1}$: all nonlinear forces (fluid film and mechanical dampers)

Collect terms in $\{X\}_{n+1}$ and solve:

$$\{X\}_{n+1} = \left[[I] - \frac{\delta t^2}{4}[M]^{-1}[K_L]\right]^{-1}\left\{\frac{\delta t^2}{4}\left\{[M]^{-1}\{F_{NL}\}_{n+1} + \{\ddot{X}\}_n\right\} + \delta t\{\dot{X}\}_n + \{X\}_n\right\}$$

Inertia and stiffness matrices, $[M]$ and $[K_L]$, are diagonal in the absence of lateral stiffness, so that inverses are trivial

MATHEMATICAL MODELLING

TIMESTEPPING

COMPUTATION	USES	TO PREDICT
Fluid film shape	$h_{n-1/2}$ $\dot{h}_{n-1/2}$ $\{X\}_n$ $\{\dot{X}\}_n$ Ψ_n	$h_{n+1/2}$ $\dot{h}_{n+1/2}$
Fluid film pressure distribution, p	p_n $h_{n+1/2}$ $\dot{h}_{n+1/2}$	p_{n+1}
Velocity and shear stress distributions	p_{n+1} $h_{n+1/2}$	$\boxed{u_{r_{n+1}}}$ $u_{\theta_{n+1}}$ $\tau_{r_{n+1}}$ $\boxed{\tau_{\theta_{n+1}}}$
Coning	p_{n+1}	Ψ_{n+1}
Fluid film forces and moments	p_{n+1} $\tau_{\theta_{n+1}}$ $\tau_{r_{n+1}}$	$\{F_{\text{Fluid film}}\}_{n+1}$
Mechanical damper forces and moments	$\{\dot{X}\}_n$ $\{\ddot{X}\}_n$ —Taylor Expansion→ $\{\dot{X}\}_{n+1}$	→ $\{F_{\text{Damper}}\}_{n+1}$; (both) → $\{F_{ML}\}_{n+1}$
Dynamics	$\{X\}_n$ $\{\dot{X}\}_n$ $\{\ddot{X}\}_n$ $\{F_{ML}\}_{n+1}$	$\boxed{\{X\}_{n+1}}$ $\{\dot{X}\}_{n+1}$ $\{\ddot{X}\}_{n+1}$

☐ = Output of fundamental importance

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MATHEMATICAL MODELLING

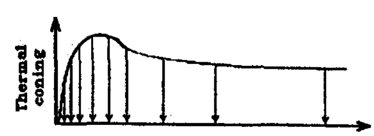
APPROACH TO THERMAL TRANSIENT CONING

PROBLEM

Thermal diffusion timescale 2-4 orders of magnitude slower than timescale of vibrational dynamics

SOLUTION

- Off-line thermal transient F.E. analysis provides coning as a function of time
- Coning-time cuve 'sampled' at user-specified intervals to provide quasi-steady coning for vibrational analysis



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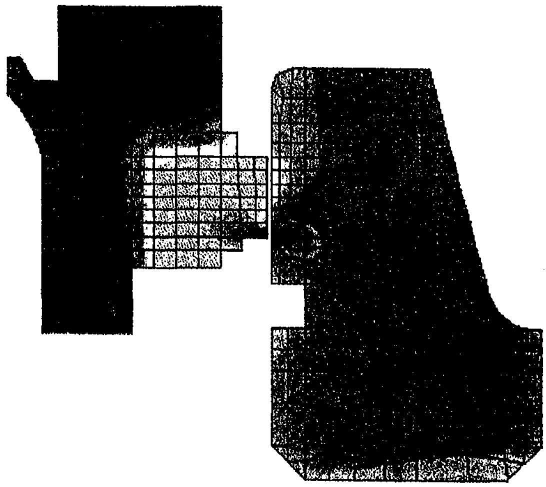

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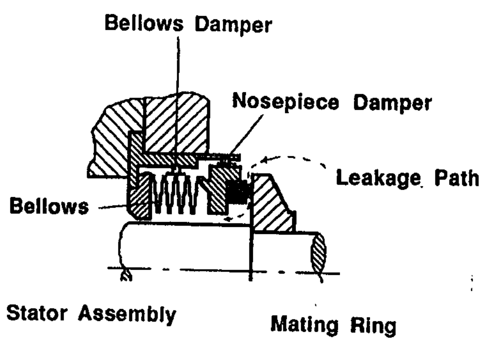
GENERIC SEAL TYPE



- Plain-faced balanced mechanical seals
- Bellows-type flexibly-mounted stator ring
- Rigidly-clamped rotor ring
- Coulomb friction vibration dampers

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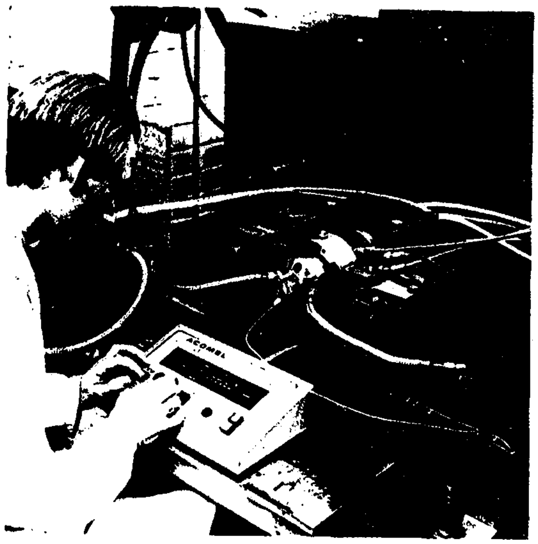

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TEST PROGRAMME

SCOPE

- 42 tests covering:
  - face coning
  - rotor eccentricity
  - rotor out-of-squareness
  - degree of damping

  } room temp, 14000 rpm, 5,10,15 bar
- Effects of high and low temperature investigated
- Tests at high speed (60,000 rpm)
- All tests on typical plain face seal, modified to suit required conditions (face diameter ≈ 30mm)

FACILITY

- Hot gas (to around 220°C) supply
- Cold gas supply (boiling to room temp)
- Liquid cryogen supply possible
- High pressure (rated to 20 bar) up to 14000 rpm
- High speed (60,000 rpm) at lower pressure

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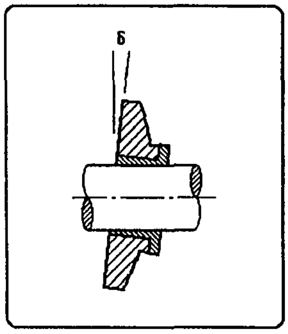
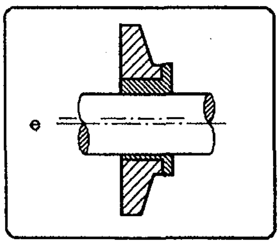
TEST PROGRAMME

MISALIGNMENT AND ECCENTRICITY

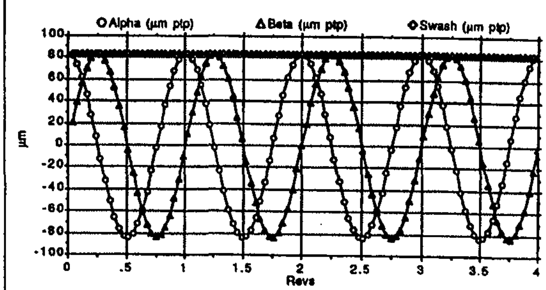
MISALIGNMENT



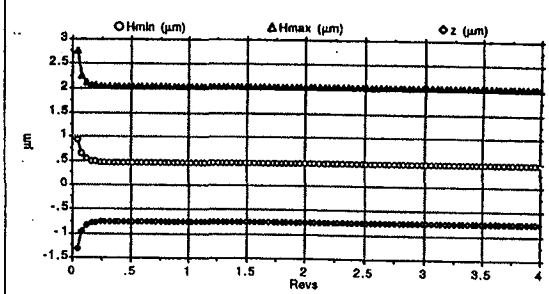
ECCENTRICITY



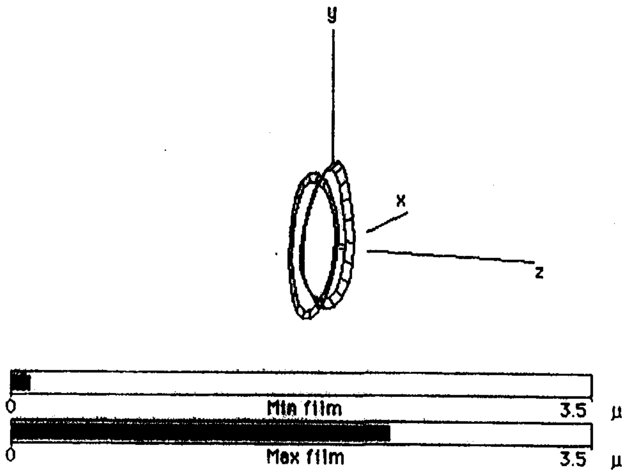
Accurate tracking of run-out



Stability reached quickly



ANIMATION SEQUENCE



AREAS SUITABLE FOR FURTHER DEVELOPMENT

- Liquid lubricant film
- Cavitating film
- Mechanical contact
- Circumferential EHD
- Different spring and secondary seal types
- Floating rotor types
- Ring seal geometries