Preliminary Test Results of a Non-Contacting Finger Seal on a Herringbone-Grooved Rotor

Margaret P. Proctor and Irebert R. Delgado
National Aeronautics and Space Administration
Glenn Research Center
Cleveland, Ohio 44135

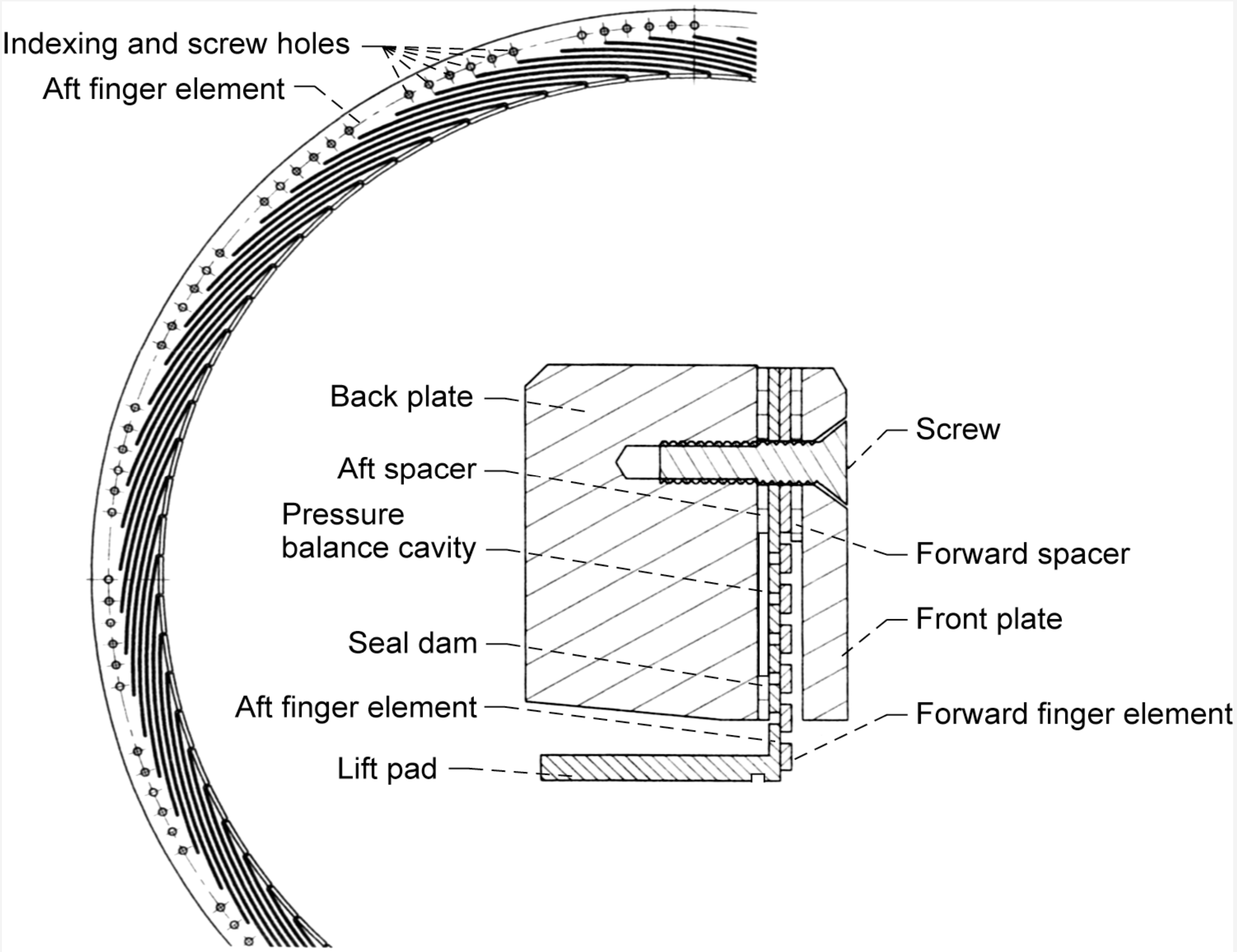
Abstract
Low leakage, non-contacting finger seals have potential to reduce gas turbine engine specific fuel consumption by 2 to 3 percent and to reduce direct operating costs by increasing the time between engine overhauls. A non-contacting finger seal with concentric lift-pads operating adjacent to a test rotor with herringbone grooves was statically tested at 300, 533, and 700 K inlet air temperatures at pressure differentials up to 576 kPa. Leakage flow factors were approximately 70 percent less than state-of-the-art labyrinth seals. Leakage rates are compared to first order predictions. Initial spin tests at 5000 rpm, 300 K inlet air temperature and pressure differentials to 241 kPa produced no measurable wear.

Preliminary Test Results of a Non-Contacting Finger Seal on a Herringbone-Grooved Rotor

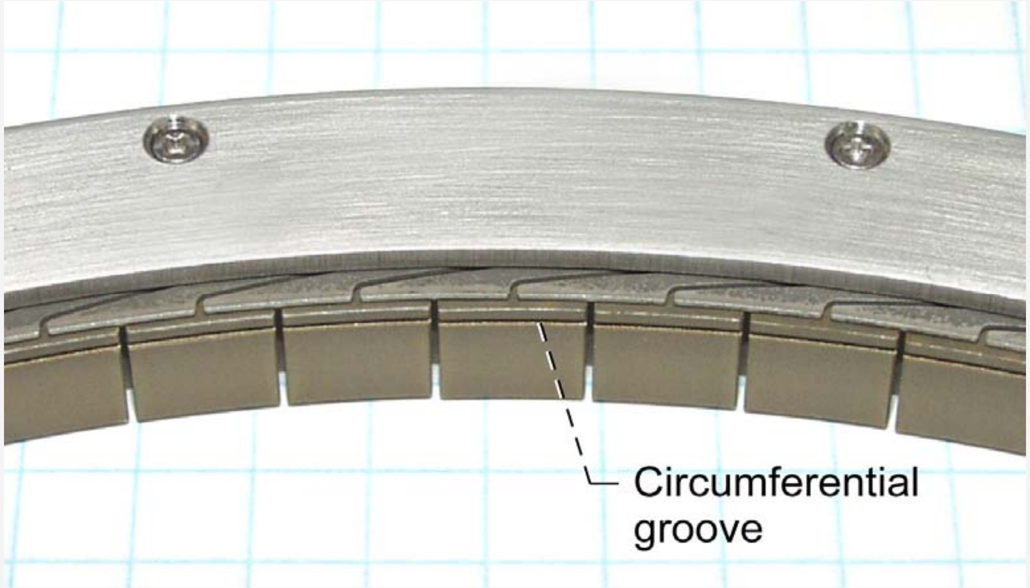
Margaret P. Proctor
and
Irebert R. Delgado
NASA Glenn Research Center
Cleveland, OH 44135

NASA Seals and Secondary Flows Symposium
Cleveland, OH
November 18, 2008

Baseline Non-Contacting Finger Seal



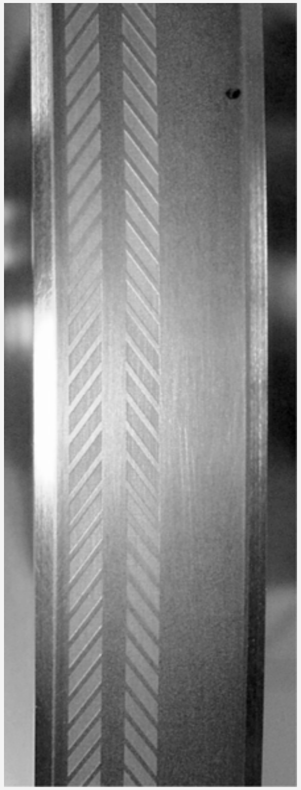
Non-Contacting Finger Seal—Pre-Test



- Haynes–188
- Temperatures up to 1089 K
- Radial clearance to rotor = 25.4 μm (0.001 in)
- Lift pads ride over herringbone grooves

Herringbone Grooves on Seal Test Rotor—Pre-Test

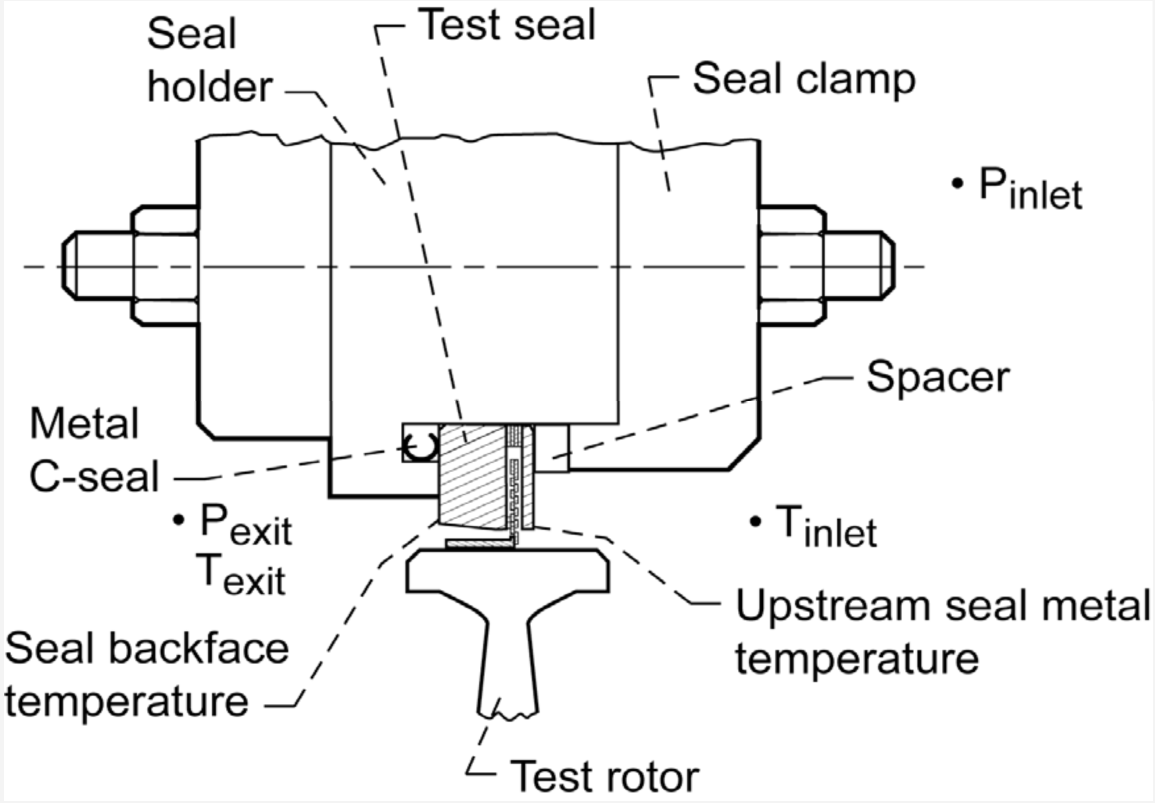
Rotation ↓



- Rotor O.D.: 21.6 cm (8.5 in)
- Grainex Mar-M–247 rotor
- Chrome carbide coating (HVOF)
- Surface finish: 0.2 µm (8 µin)
- 536 grooves (268 around circumference)
- Groove depth: 20 µm (0.0008 in)
- Groove ends:
  - Begin at middle of circumferential groove on lift pads
  - Extend past low pressure edge of lift pads

High-Temperature, High-Speed Turbine Seal Rig

Test Seal Configuration and Location of Research Measurements



Flow Factor

$$\phi = \frac{\dot{m}\sqrt{T_{avg}}}{P_u \times D_{seal}}, \quad \frac{\text{kg-}\sqrt{\text{K}}}{\text{MPa-m-s}}$$

$\dot{m}$ = air leakage flow rate, kg/s.

T_{avg} = average seal air inlet temperature, K.

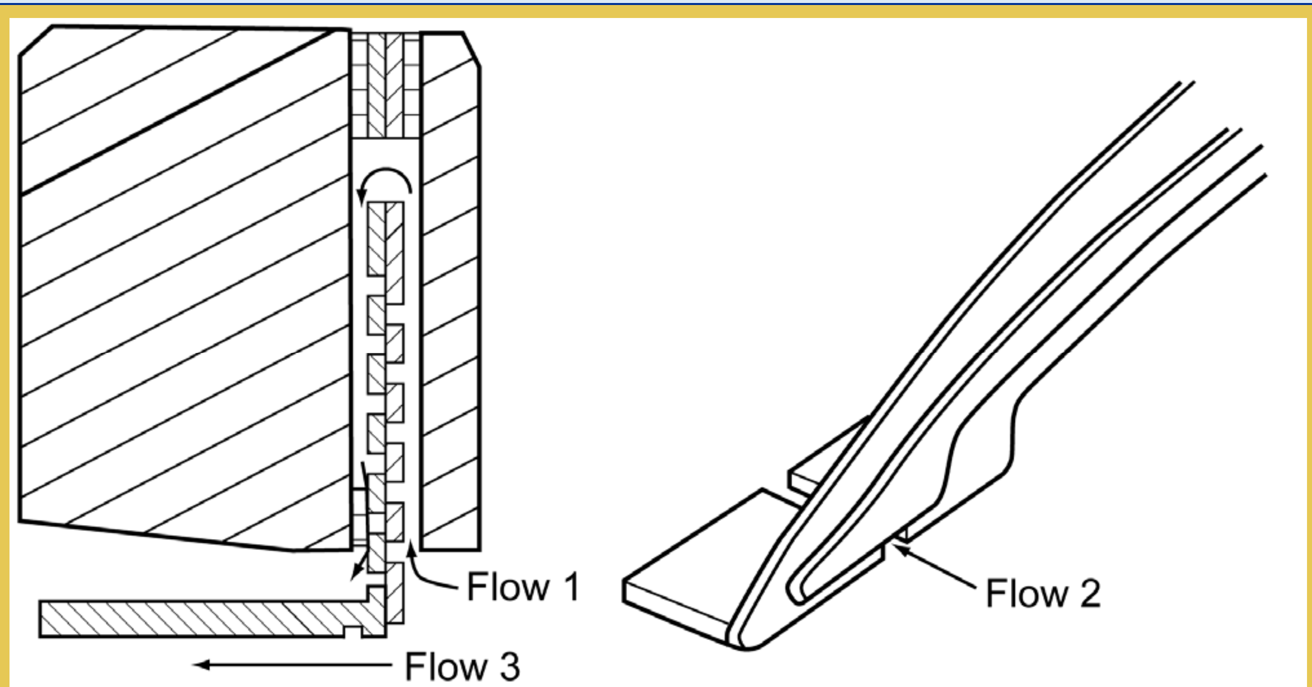
P_u = air pressure upstream of seal, MPa.

D_{seal} = outside diameter of the test rotor, m.

Test Procedure

Test	Shaft speed, rpm	Inlet air temperature, K	Pressure drop across seal, kPa	Comment
Initial static tests	0	300 533 700	0–517 kPa (or max.) – 0	
Shaft bound. Seal removed, visually inspected and reinstalled.				
Bind-up tests Part 1 Part 2	0	300 K	 69, 138, 207, 276, 345, 414, 483 0-276 in 13.8 increments, 483, 552, 576	Can shaft turn by hand? at 0 kPa at test pressure
Seal removed for visual inspection. Deposits on seal sampled. Seal ultrasonically cleaned and reinstalled.				
Repeat Static Tests	0	300 533 700	0 – max. – 0, 3 cycles	
Shaft bound. Seal removed, visually inspected, and reinstalled.				
Spin Test 1	0 5,000 0 0	300 K ↓	13.8 13.8, 34, 69, 103, 138,103, 69, 34, 13.8 13.8 0	25 min at 5000 rpm
Seal removed, visually inspected, and reinstalled.				
Spin Test 2	0 0 5,000 0	300 K ↓	0 – max. – 0, 1 cycle 13.8 13.8, 34, 69, 103, 138, 172, 206, 241, 206, 172, 138, 103, 69, 34, 13.8 13.8	68 min at 5000 rpm
Seal removed and visually inspected.				

Leakage Flow Model



Assumptions

- Isentropic flow
- Seal leakage area is sum of areas of each flow path
- Geometry is fixed
- Lift pads remain concentric to rotor
- Finger elements held tightly to each other and seal dam so there is no leakage between contacting areas
- Pressure in balance cavity equals seal inlet pressure

Leakage Flow Model

$$\dot{m} = \frac{P_u}{\sqrt{RT_u}} \cdot A\sqrt{\gamma} M \left(1 + \left(\frac{\gamma - 1}{2}\right) M^2\right)^{1/2 - \frac{\gamma}{\gamma - 1}}$$

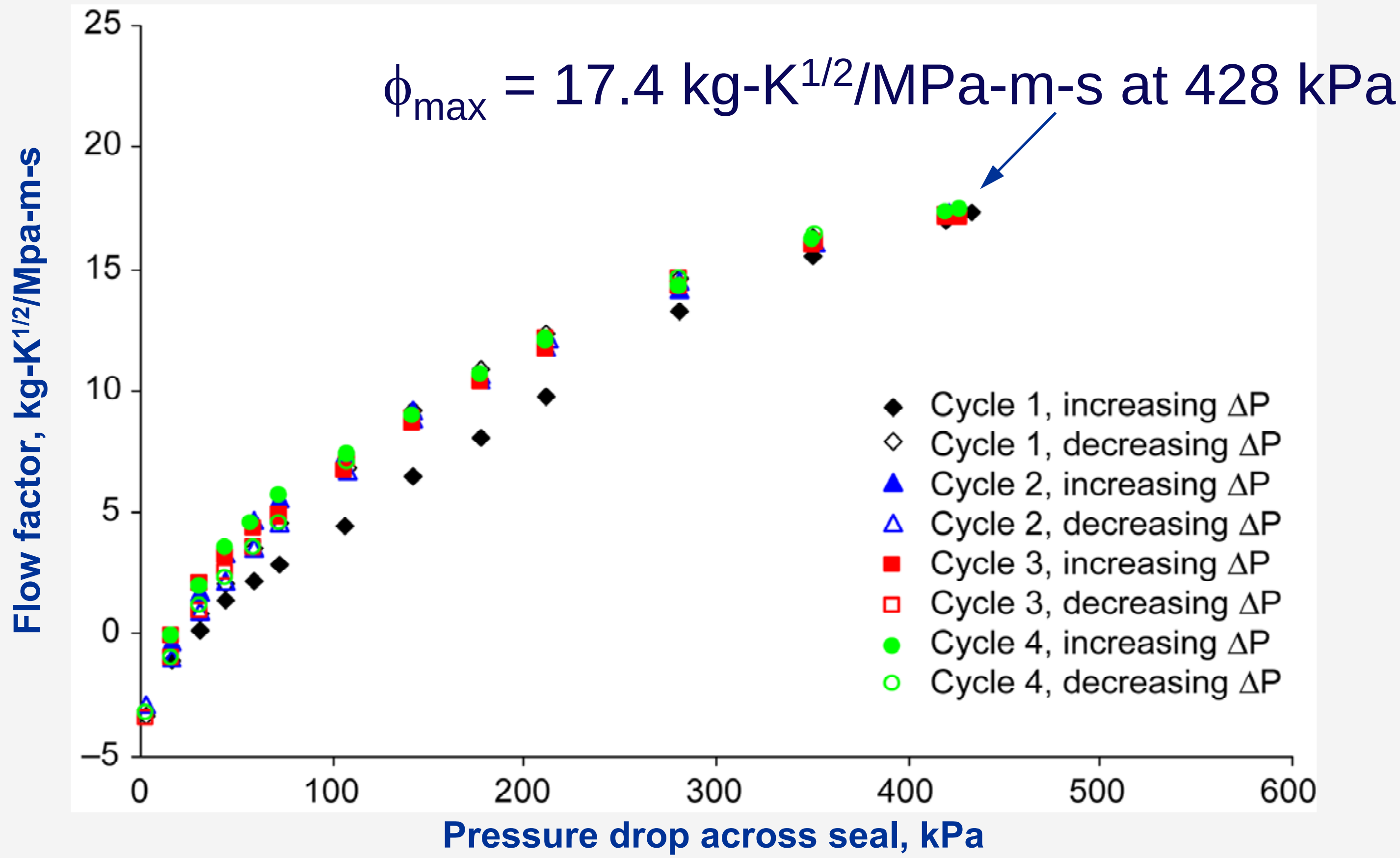
where

$$M = \left[\left(\left(\frac{P_u}{P}\right)^{\frac{\gamma - 1}{\gamma}} - 1\right)\frac{2}{\gamma - 1}\right]^{\frac{1}{2}}$$

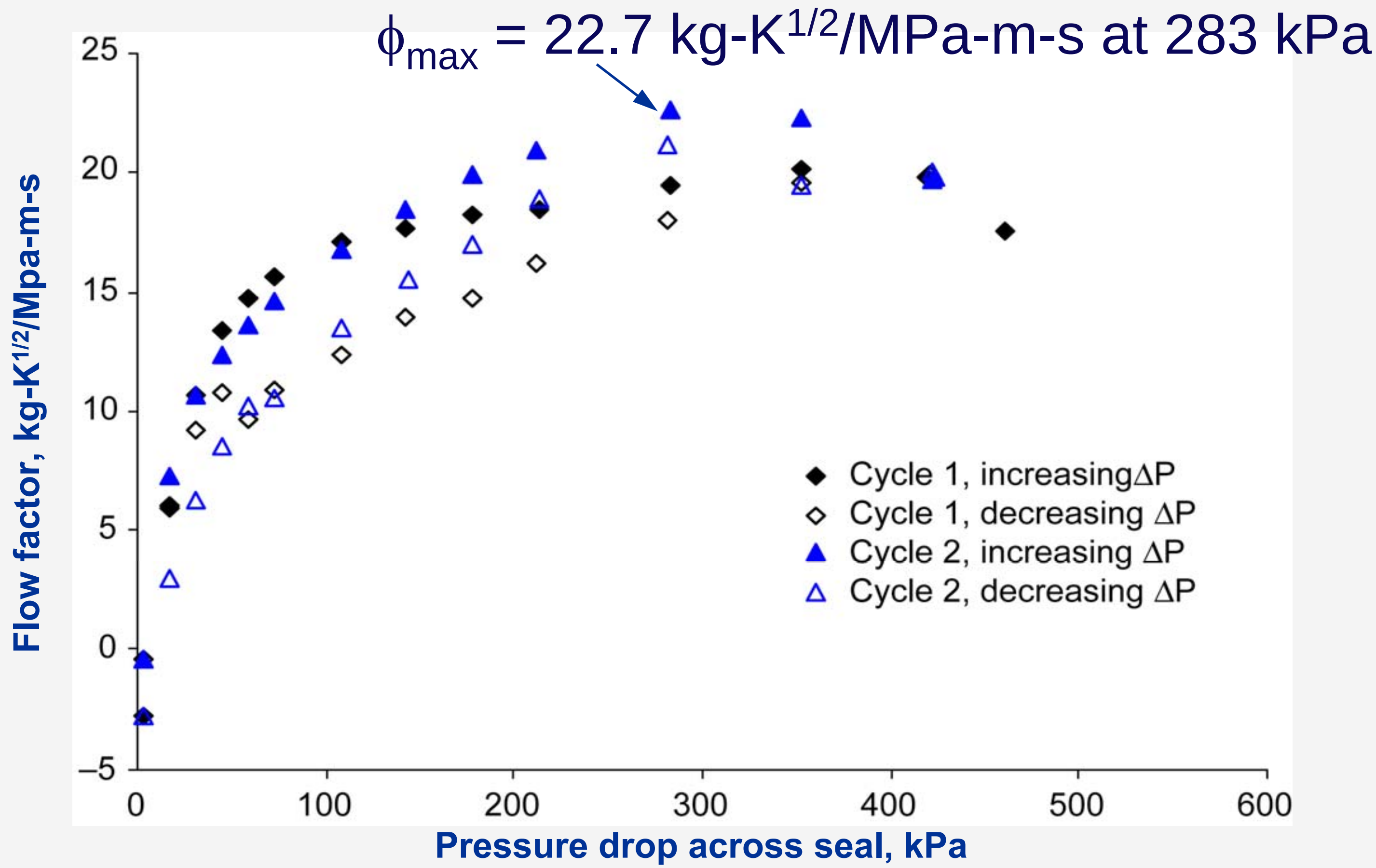
For air (γ = 1.4), when P/P_u ≤ 0.5283 the flow is choked

$$\dot{m} = \frac{P_u}{\sqrt{RT_u}} \cdot A \cdot (0.6847)$$

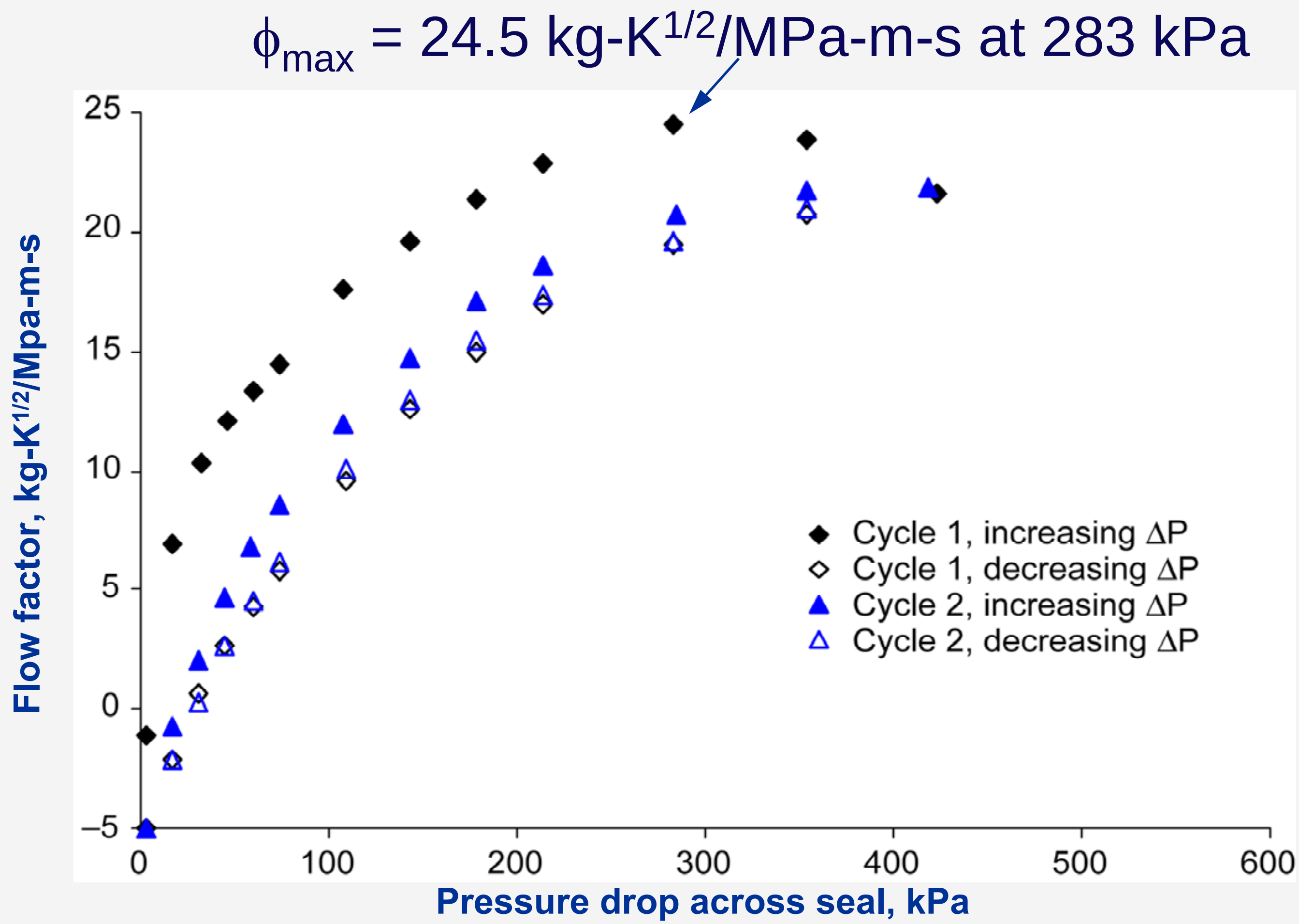
Initial Static Leakage Performance at 300 K
Radial Clearance = 25.4 µm



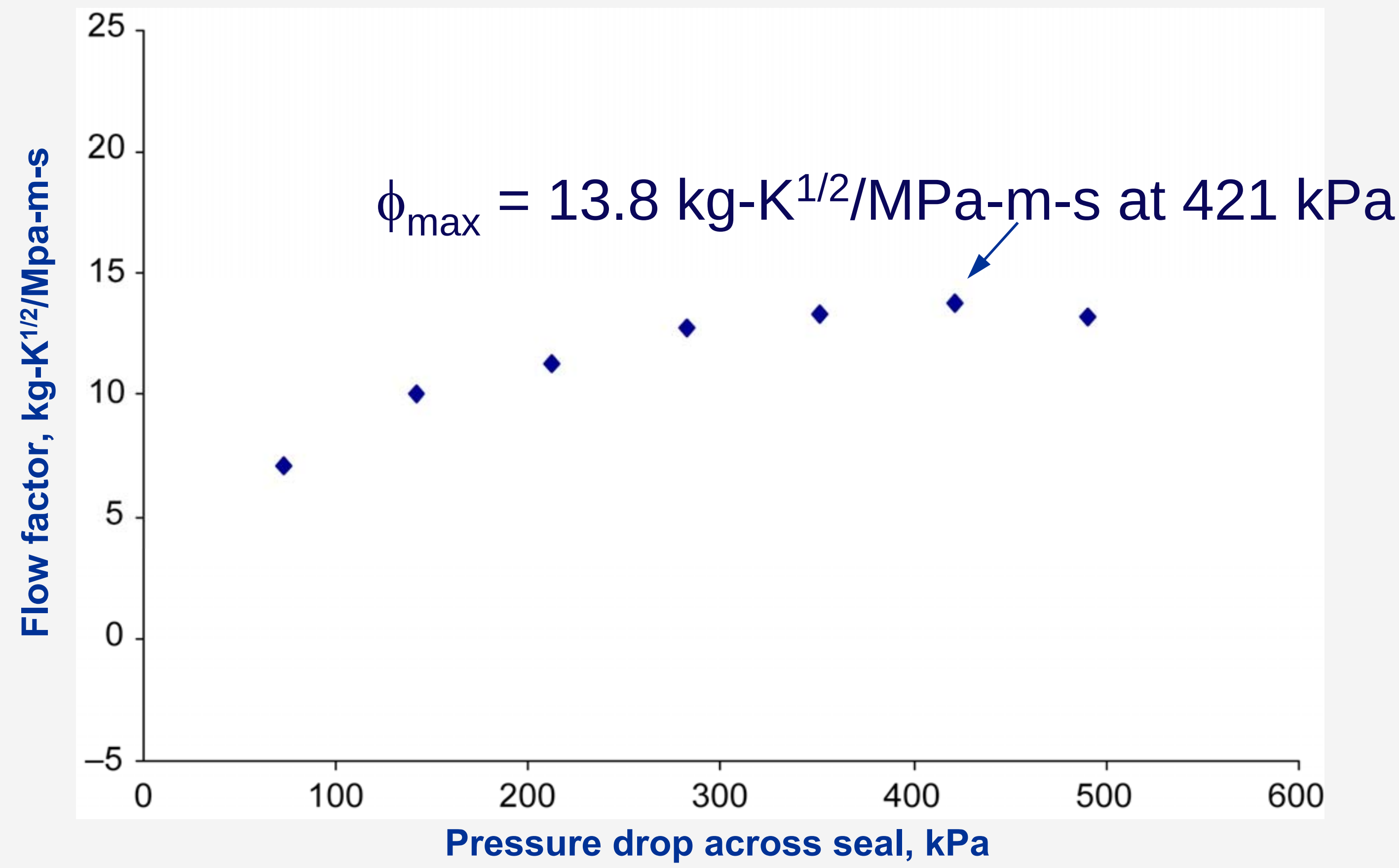
Initial Static Leakage Performance at 533 K
Radial Clearance = 48.3 µm



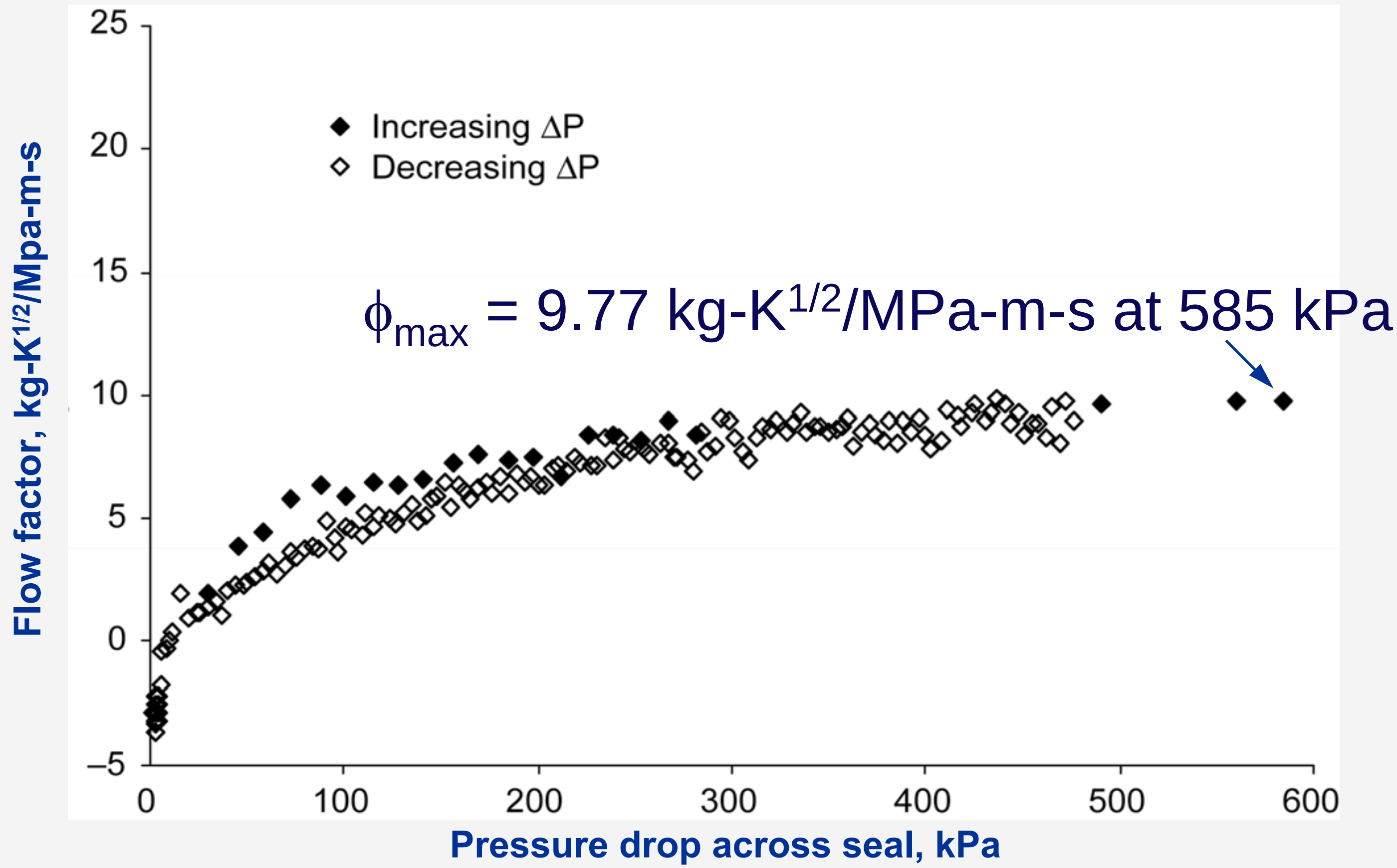
Initial Static Leakage Performance at 700 K
Radial Clearance = 61 µm



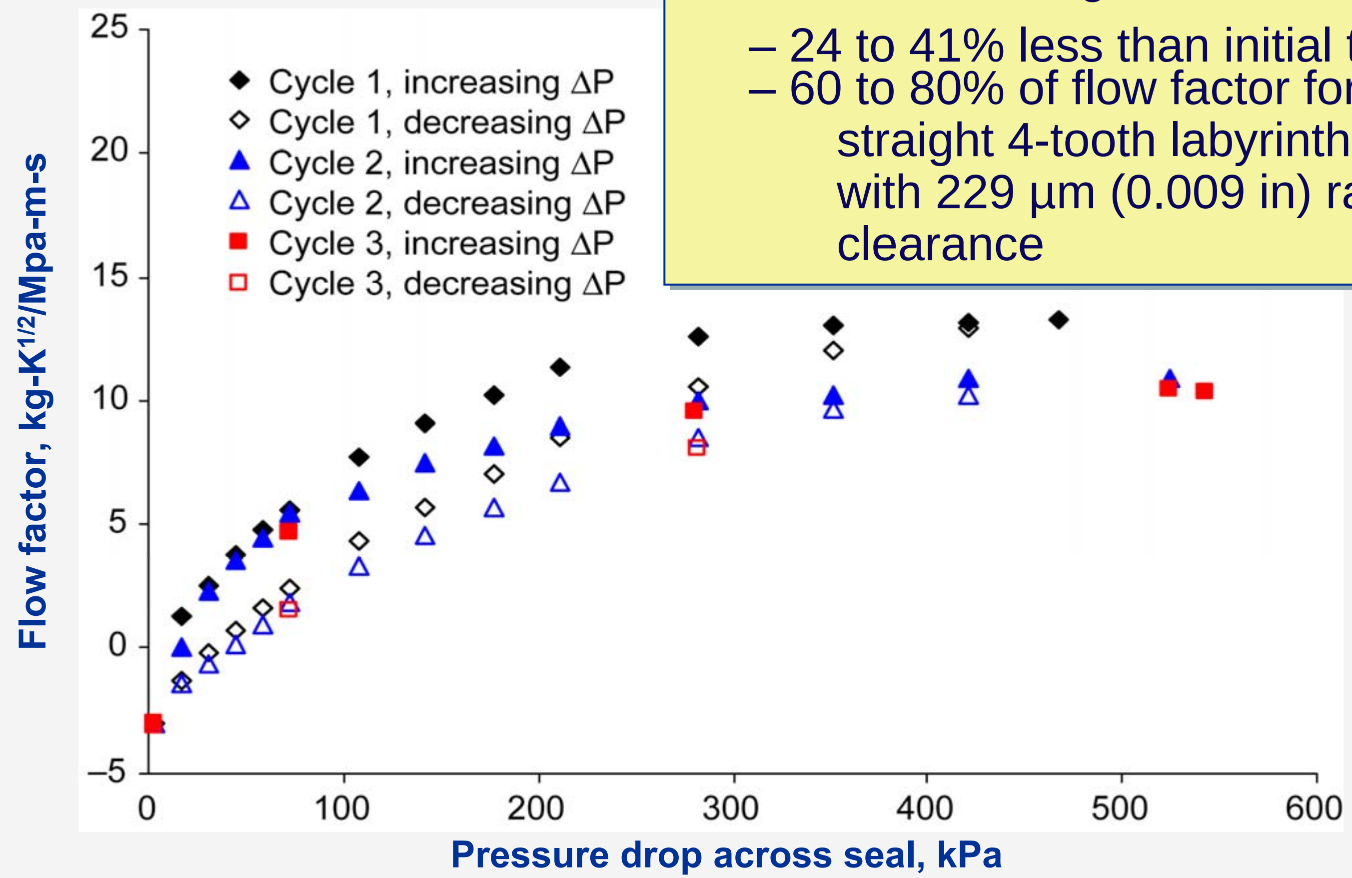
Bind-Up Test Part 1–Static Leakage Performance at 320 to 344 K



Bind-Up Test Part 2–Static Leakage Performance at 342 to 345 kPa



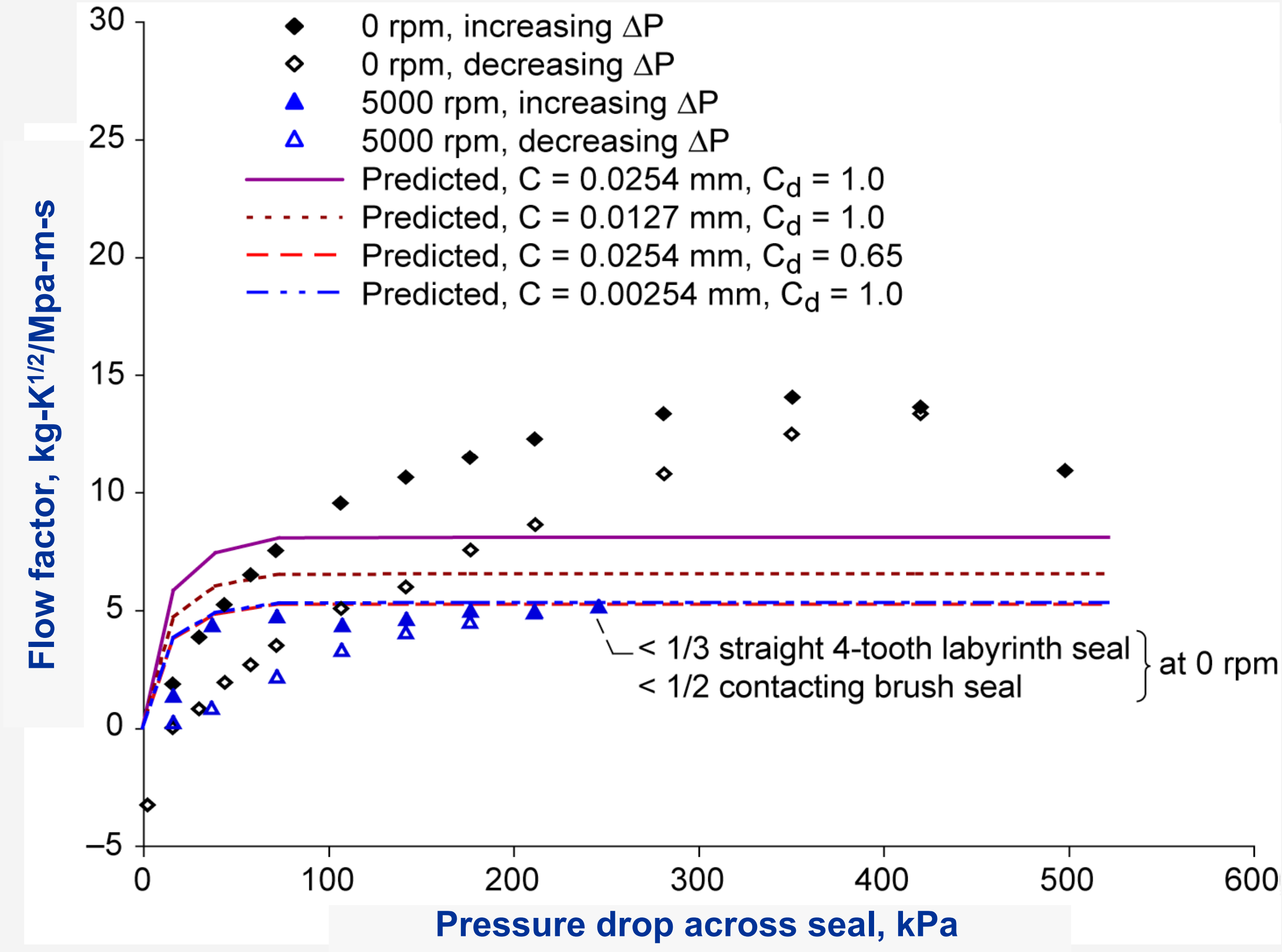
Repeat Static Test Leakage Performance at 300 K



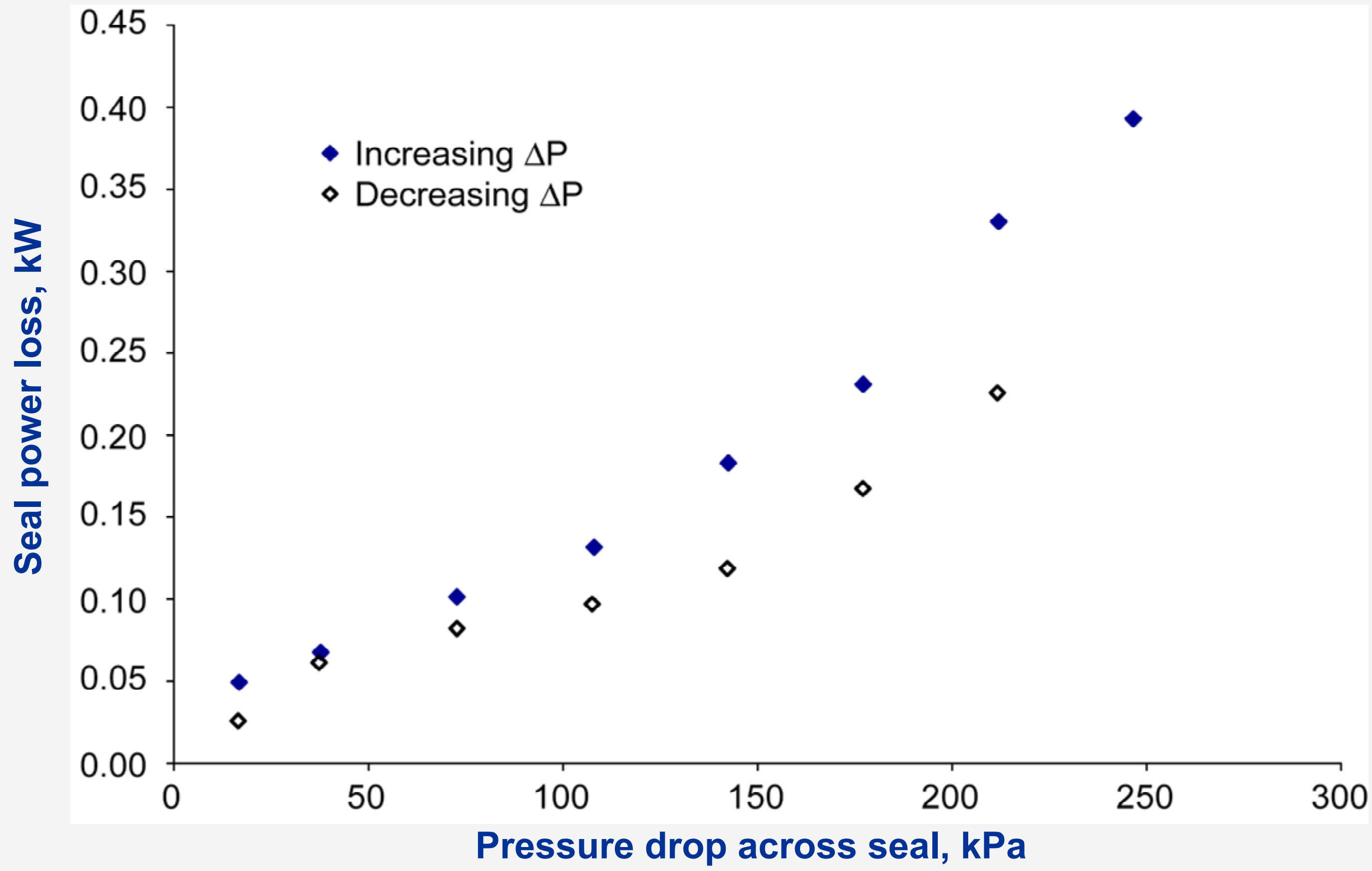
Maximum Flow Factor is:

- 10.3 to 13.2 kg-$K^{1/2}$/MPa-m-s
- 24 to 41% less than initial test
- 60 to 80% of flow factor for straight 4-tooth labyrinth seal with 229 µm (0.009 in) radial clearance

Second Spin Test Leakage Performance at 300 K



Second Spin Test Seal Power Loss at 300 K



Wear Results After Initial Spin Tests

Seal

- Visual inspection finds seal in good condition.
- No significant change in weight.
- Light burnishing on:
  – All low-pressure lift pads at I.D. near high-pressure edges.
  – High-pressure fingers around the finger "toe".
- All the fingers and lift pads are free to move.

Rotor

- Shiny wear track of uniform axial length around entire circumference has no perceptible depth by touch.
- Grooves were clean and free of debris.

- Burnishing is result of brief contact during start and stop of shaft rotation.
- There was no rapid or substantial rise in seal exit or back face temperature.

Non-contacting operation was achieved.

Conclusions

The Non-Contacting Finger Seal promises low leakage and long life capability.

1. No measurable wear after 93 minutes of rotation at 300 K and 5000 rpm.
2. Non-contacting operation was achieved at 5000 rpm and 14 to 241 kPa.
3. The measured flow factor at 5000 rpm and 241 kPa was
   <1/3 of the measured flow factor of a straight 4-tooth labyrinth seal and
   <1/2 of the measured flow factor of a contacting brush seal at static conditions.
4. Rotation is required to properly seat the seal and results in lower flow factors.
5. Non-contacting finger seal power loss is the same order of magnitude as brush and finger seals.

The simplified flow model is in reasonable agreement with data once flow chokes.

Further testing and analysis is needed to

- understand the nuances of this particular non-contacting finger seal design
- develop useful design methodologies and predictive tools.

Fluid-structural modeling is needed to

- understand bind-up observed at 276 kPa
- determine design modifications to achieve higher pressure capability.