BRUSH SEAL ARRANGEMENT FOR THE RS-68 TURBOPUMP SET

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The nature of the RS-68 turbopumps requires that the hydrogen seals separating the pump from the turbine must have extremely low levels of leakage and be contained in small packages. Conventional seal technologies are not able to reasonably satisfy such design requirements. A review of experimental measurements and analysis publications suggests that brush seals are well suited for the design requirements. Brush seals are shown to have less leakage than conventional labyrinth and damper seals and have no adverse affects on the rotordynamics of the machine. The bulk-flow analysis presented by Hendricks et al. is used as a guideline to create a spreadsheet that provides mass flow through the seal and heat generated by the rubbing contact of the bristles on the shaft. The analysis is anchored to published data for LN2 and LH2 leakage tests. Finally, the analysis is used to design seals for both applications. It is observed that the most important analysis parameter is the thickness of the bristle pack and its relationship to seal clearance, lay angle and pressure drop.

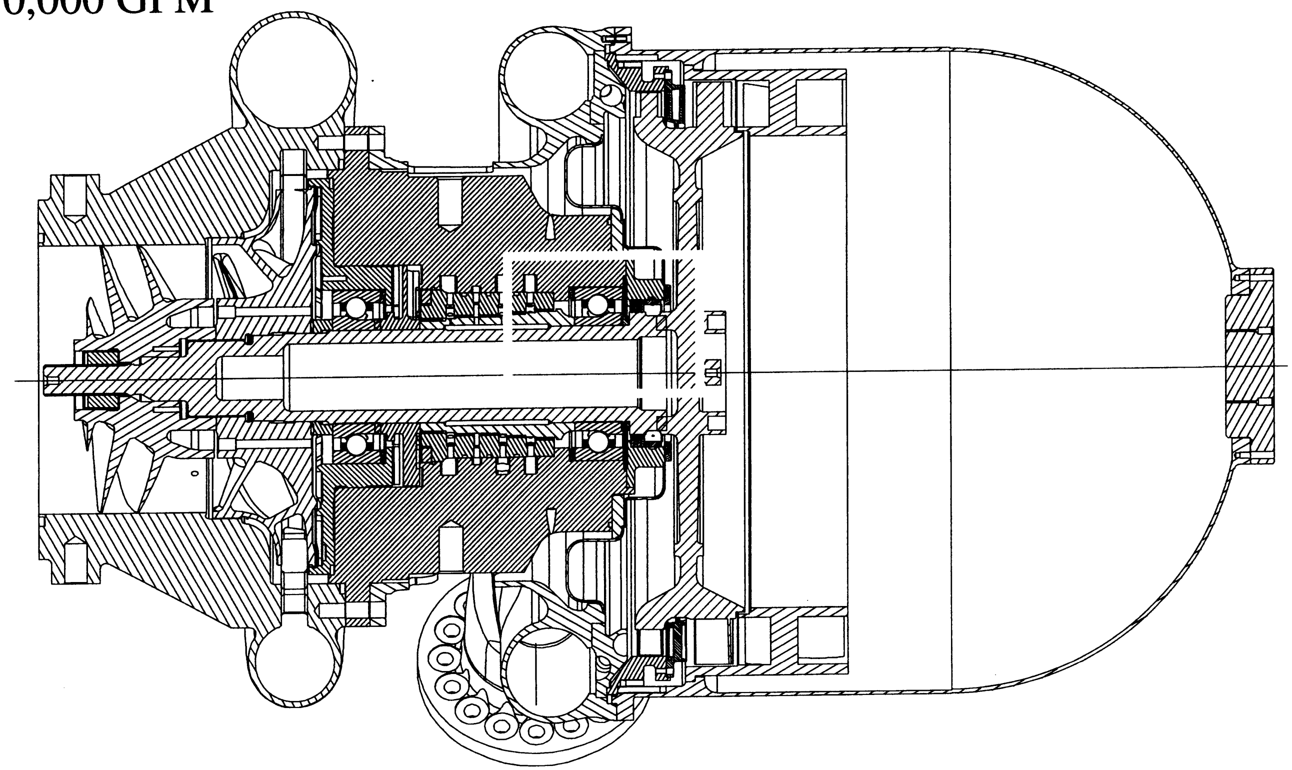
Agenda

- Introduction to seal applications
- Concept trade
- Analysis methodology
- Design methodology
- Final design description
- Recommendations

RS-68 Oxygen Turbopump

<u>Pump</u>
8,700 RPM
24,000 PSI Discharge
10,000 GPM

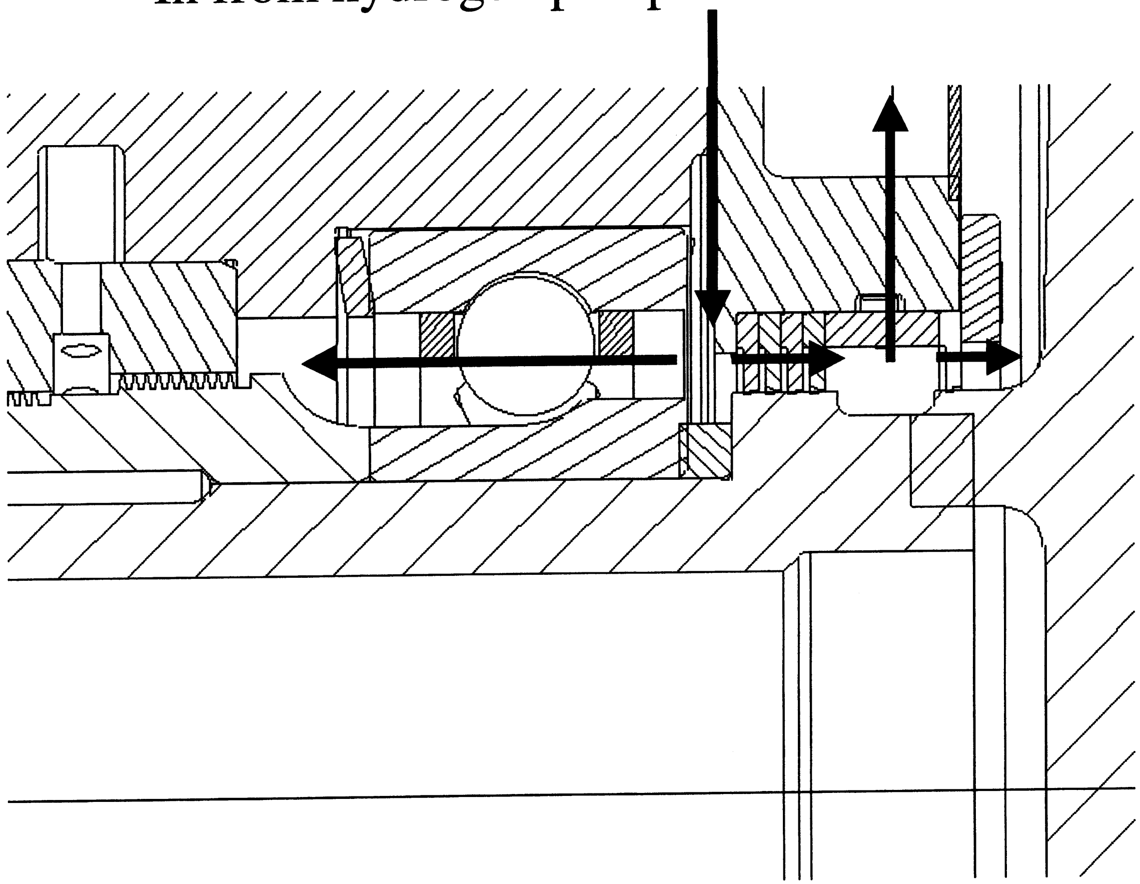
<u>Turbine</u>
1,400 R
715 PSI Inlet
19 lbm/sec



Slide 3 – RS-68 Oxygen Turbopump

This first slide is a picture of the RS-68 liquid oxygen turbopump. The LOX is pumped by the inducer and impeller located on the left end of the turbopump. The shaft rotates at 8,700 RPM producing a 24,000 PSIA pump discharge pressure and 10,000 GPM of flow. The driving turbine is on the right end of the turbopump. The turbine inlet temperature and pressure is 1,400° R and 715 PSIA respectively with a mass flow rate of 19 lbm/sec. The yellow box highlights the seal application under discussion.

RS-68 Oxygen Turbopump (Hydrogen Flow Path)



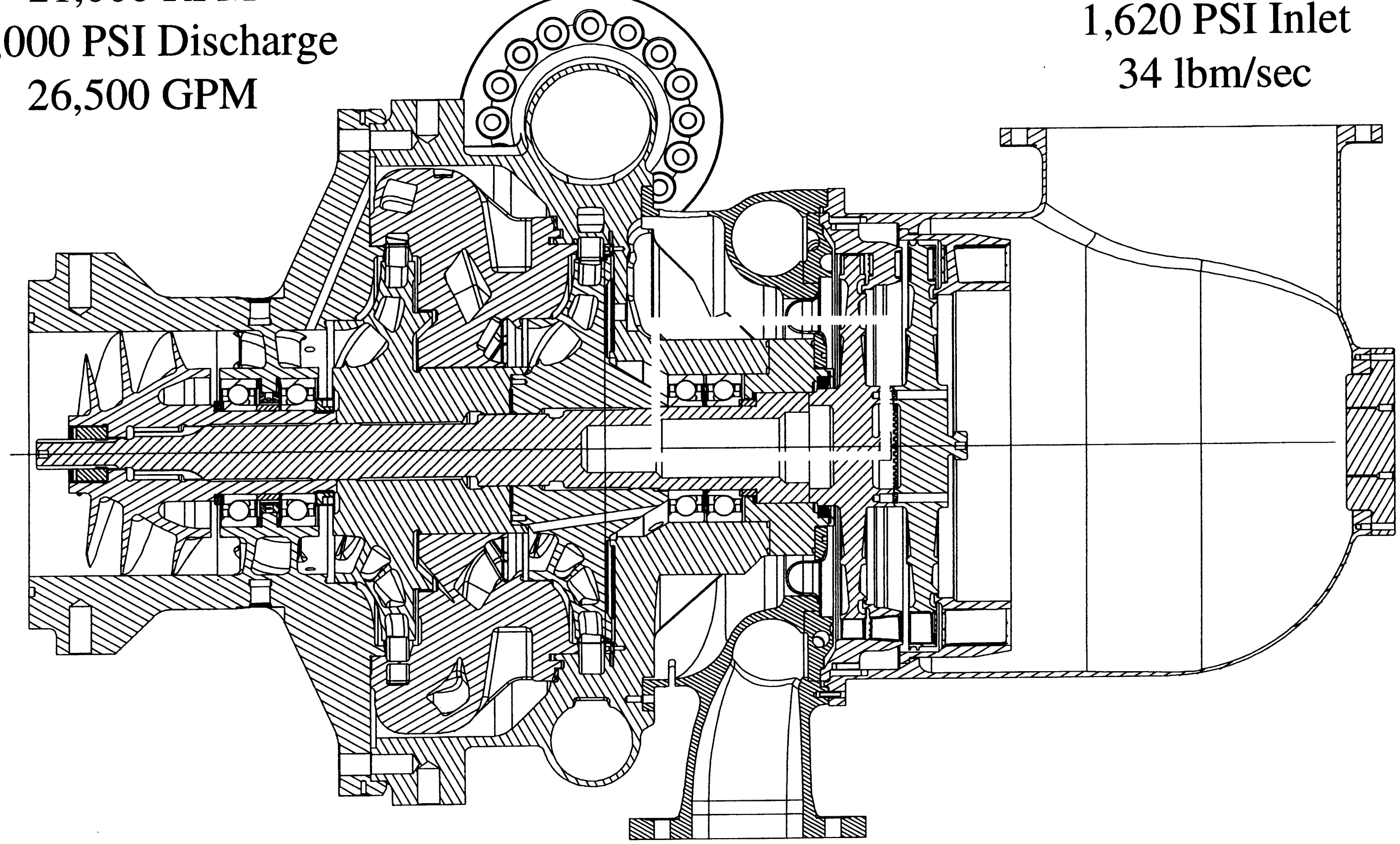
Slide 4 – RS-68 Oxygen Turbopump (Hydrogen Flow Path)

This picture shows the previously highlighted region of the oxygen turbopump. The hydrogen is introduced from the liquid hydrogen turbopump pump discharge and flows across the bearing (to provide cooling) and through the seal package where the majority of the flow is sent overboard and the remaining flow is used to prevent turbine gases from progressing towards the pump end of the turbopump. The amount of hydrogen that leaks into the turbine cavity must be minimized to prevent the turbine disk from cooling on the back side which will cause deformation of the disk.

RS-68 Hydrogen (Fuel) Turbopump

Pump
21,000 RPM
24,000 PSI Discharge
26,500 GPM

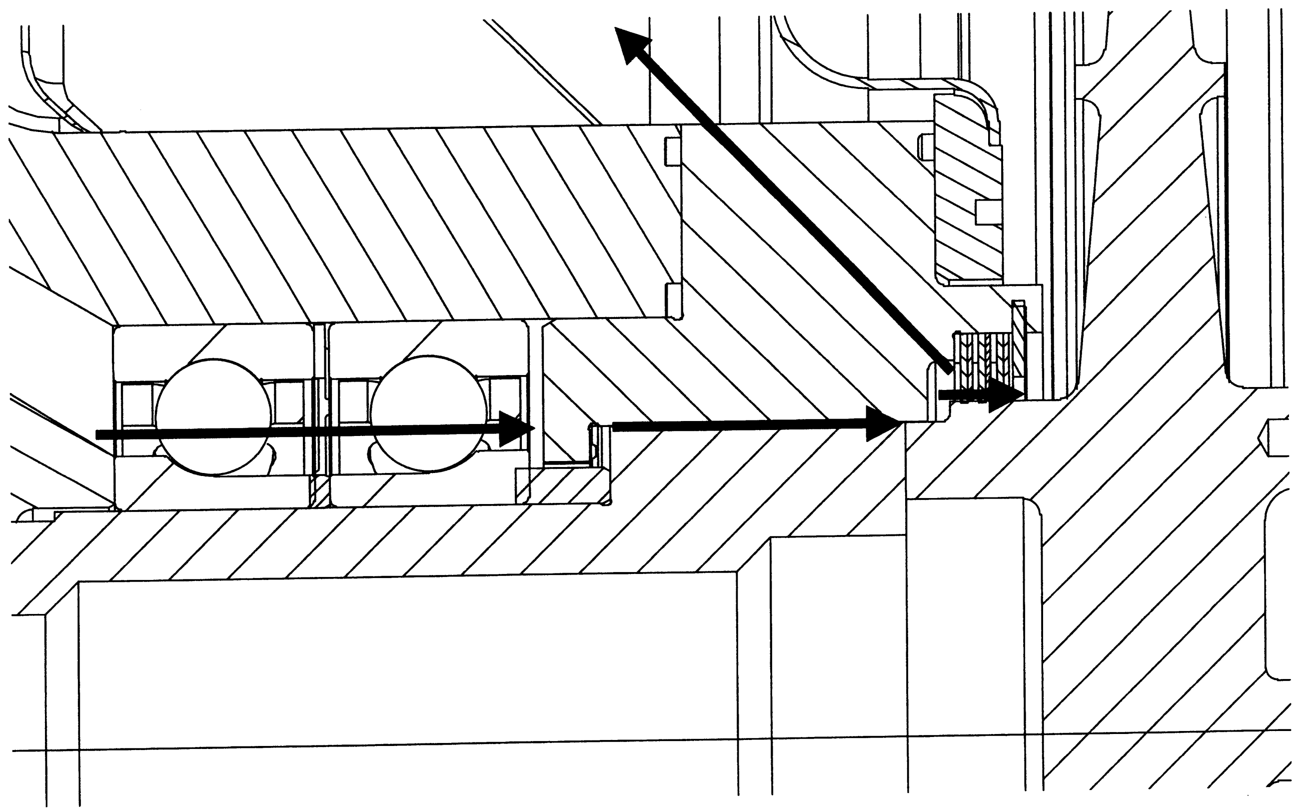
Turbine
1,400 R
1,620 PSI Inlet
34 lbm/sec



Slide 5 – RS-68 Hydrogen (Fuel) Turbopump

This is a picture of the RS-68 liquid hydrogen turbopump. The LH2 is pumped by the inducer and impeller located on the left end of the turbopump. The shaft rotates at 21,000 RPM producing a 24,000 PSIA pump discharge pressure and 26,500 GPM of flow. The driving turbine is on the right end of the turbopump. The turbine inlet temperature and pressure is 1,400° R and 1,620 PSIA respectively with a mass flow rate of 34 lbm/sec. The yellow box highlights the seal application under discussion.

RS-68 Hydrogen (Fuel) Turbopump (Hydrogen Flow Path)



Slide 6 – RS-68 Hydrogen (Fuel) Turbopump (Hydrogen Flow Path)

This picture shows the previously highlighted region of the hydrogen turbopump. The hydrogen passes through the ball bearings to provide cooling and then passes through a damper seal to add stability to the rotor system. Finally, a small amount is passed into the turbine cavity to prevent turbine gasses from progressing towards the pump end. The remaining hydrogen flow is dumped overboard. Again, it is important to minimize the leakage into the turbine cavity to prevent the turbine disk from deforming.

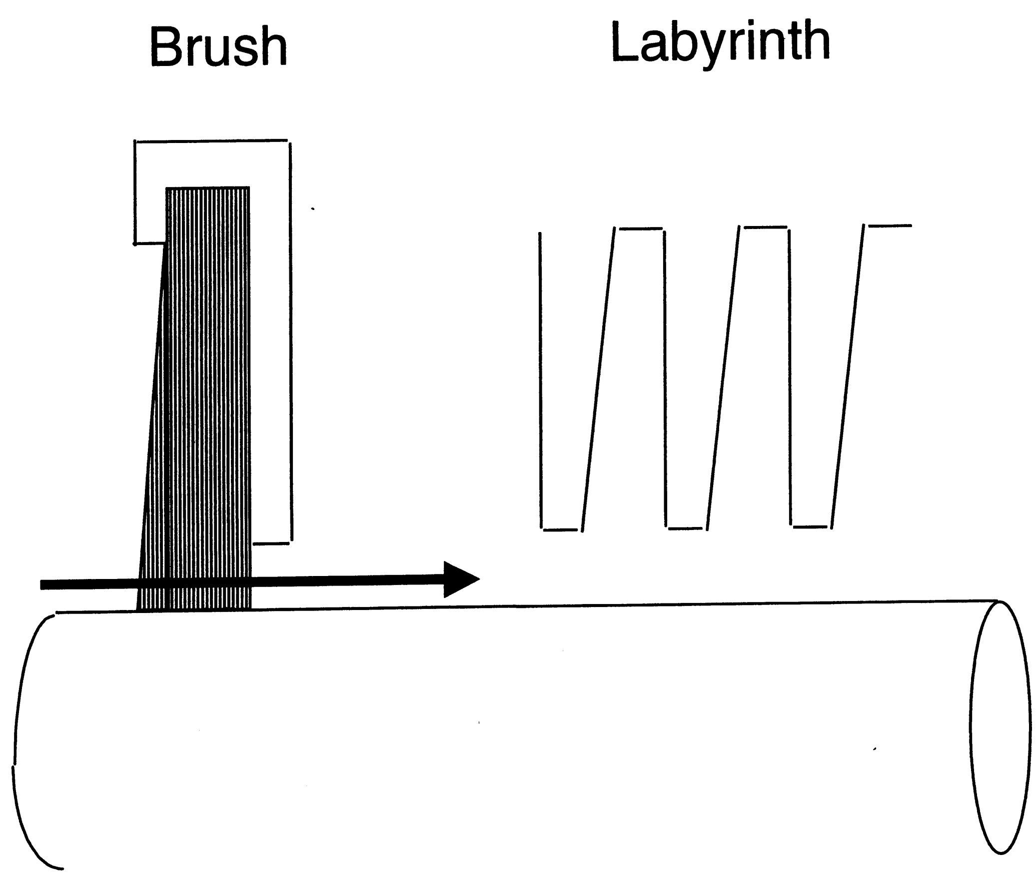
Seal Requirements for the RS-68 Turbopumps

- Hydrogen cooled turbine end ball bearing
- Low leakage into turbine cavity (less than 0.15 lbm/sec)
- Low cost, robust design
- *Liquid Oxygen Turbopump*
  - Approximately 250 psi ΔP
- *Liquid Hydrogen Turbopump*
  - Approximately 50 psi ΔP
  - Restricted axial package size

Slide 7 – Seal Requirements for the RS-68 Turbopumps

In both turbopumps, the turbine end ball bearings are cooled by liquid hydrogen. The same liquid hydrogen is passed into the turbine cavity to prevent turbine gasses from flowing towards the pump end. As mentioned before, the leakage into the turbine cavity must be minimal (less than 0.15 lbm/sec) to prevent the turbine disk from deforming. Also, the seal design used must be low cost and reliable to meet expendable vehicle cost requirements. The liquid oxygen pump requires a seal pressure drop of 250 psi. The liquid hydrogen pump requires a seal pressure drop of 50 psi and minimal axial length.

Two Annular Seal Types



Slide 8 – Two Annular Seal Types

An example of a brush seal and a labyrinth seal are shown here. The fluid flow is designated by the red arrow. As seen in the picture, the brush seal actually contacts the rotor whereas the labyrinth seal does not.

Labyrinth Seal Design for the RS-68 Turbopumps

- *Liquid Oxygen Turbopump*
  - Labyrinth seal sized for pressure drop
  - Radial clearance of 0.002 to 0.0025 inches required
  - Predicted leakage too high at 0.30 lbm/sec
  - Design clearance unreasonable due to bearing deadband
- *Liquid Hydrogen Turbopump*
  - Face riding carbon seal with labyrinth seal to control leakage
  - Added length of shaft to accommodate seal package detrimental to rotordynamic stability margins

Slide 9 – Labyrinth Seal Design for the RS-68 Turbopumps

Initially, a labyrinth seal is sized for the liquid oxygen pump application. In order to get minimum leakage, the seal clearance is set at 0.002 to 0.0025 inches radially. Since the bearing dead-band alone is about 0.001 inches, this design is not feasible and will likely result in a rub. Additionally, the leakage rate is still too high (0.30 lbm/sec) to meet the design requirements. For the liquid hydrogen turbopump, a face riding carbon seal with a labyrinth seal to control leakage is designed. The largest difficulty in using the proposed design is the axial length required for such a device. The longer shaft necessary for such a seal package is detrimental to rotordynamic stability margins.

Brush Seals Investigated (Advantages)

- Low leakage for axial length
- Application to turbopump allows for further reduction in leakage
  - Turbopump radial tolerance permits lower fence height
  - Proximity to bearing package reduces potential for fence rub
- No rotordynamic instability generated from circumferential flow
  - Brush seal behaves like a swirl brake (Conner and Childs, 1993)

Slide 10 – Brush Seals Investigated (Advantages)

Brush seals have two distinct advantages over labyrinth seals, low leakage and rotordynamic stability. The low leakage is enhanced in turbopump applications because the tight radial tolerance allows for a lower fence height. In addition, the seals are located close to bearing packages so that shaft movements are minimal, thus reducing the potential for fence rub. Conner and Childs (1993) report on tests to identify the rotordynamic force coefficients of brush seals. The authors report that the seal did not exhibit any cross-coupled stiffness. Further, it is noted that the brush seal actually behaves like a swirl brake, removing the circumferential energy from the fluid that generates cross-coupled stiffness in labyrinth seals.

Brush Seals Investigated (Disadvantages)

- **Increased temperatures due to friction between bristle and rotor surfaces**
  - **Liquid hydrogen good coolant**
  - **Minimum interference of -0.002 inches reduces normal force of bristles on rotor**
- **Rubbing contact wears bristle material and rotor coating**
  - **Design life of expendable turbopump around 800 seconds**

Slide 11 – Brush Seals Investigated (Disadvantages)

Because brush seals are a contacting annular seal, heat generation and material wear are problems that must be dealt with. In the cryogenic turbopump application, the cool liquid hydrogen passing through the seal removes the heat generated by the bristles rubbing on the shaft. Also, minimizing the interference helps to reduce the magnitude of the normal force between the bristle and the shaft, thus minimizing heat generation. The minimal interference also reduces the possibility of bristle wear and particle contamination of the turbine. Finally, since the RS-68 is designed for ELV application, the design life is only around 800 seconds so that the seal does not have to last for a long duration.

Brush Seal Literature Reviewed

- Analysis Literature
  - Hendricks et al. - "A Bulk Flow Model of a Brush Seal System"
  - Chupp et al. - "Simple Leakage Flow Model for Brush Seals"
  - Rhode and Pung - "Numerical Investigation of Brush Seal Bristle Lifting Phenomena"
- Test Literature
  - Proctor et al. - "Brush Seals for Cryogenic Applications"
  - Conner and Childs - "Rotordynamic Coefficient Test Results for a Four-Stage Brush Seal"

Slide 12 – Brush Seal Literature Reviewed

There are three main papers reviewed concerning the analysis of brush seals. The first two are similar in that the leakage is estimated using a bulk flow analysis. However, the first paper (Hendricks et al.) models the flow through the brush seal with more sophistication and is accompanied by test data. In the second analysis, Chupp et al. intend to provide a rough but quick tool for predicting brush seal performance by lumping all of the seal parameters into an effective seal thickness, but this application requires a more thorough analysis. The third paper by Rhode and Pung appears to be too complicated for this particular analysis and is not pursued. In addition to the analysis literature, two papers are used for experimental test information. Proctor et al. present test data which is used in the analysis of Hendricks et al. to validate the analytical model. Proctor et al. measure mass flow, pressure drop and temperature for several brush seal configurations and operating conditions. Conner and Childs, as mentioned earlier, identify the rotordynamic force coefficients of several brush seals and find that the seals are dynamically stable.

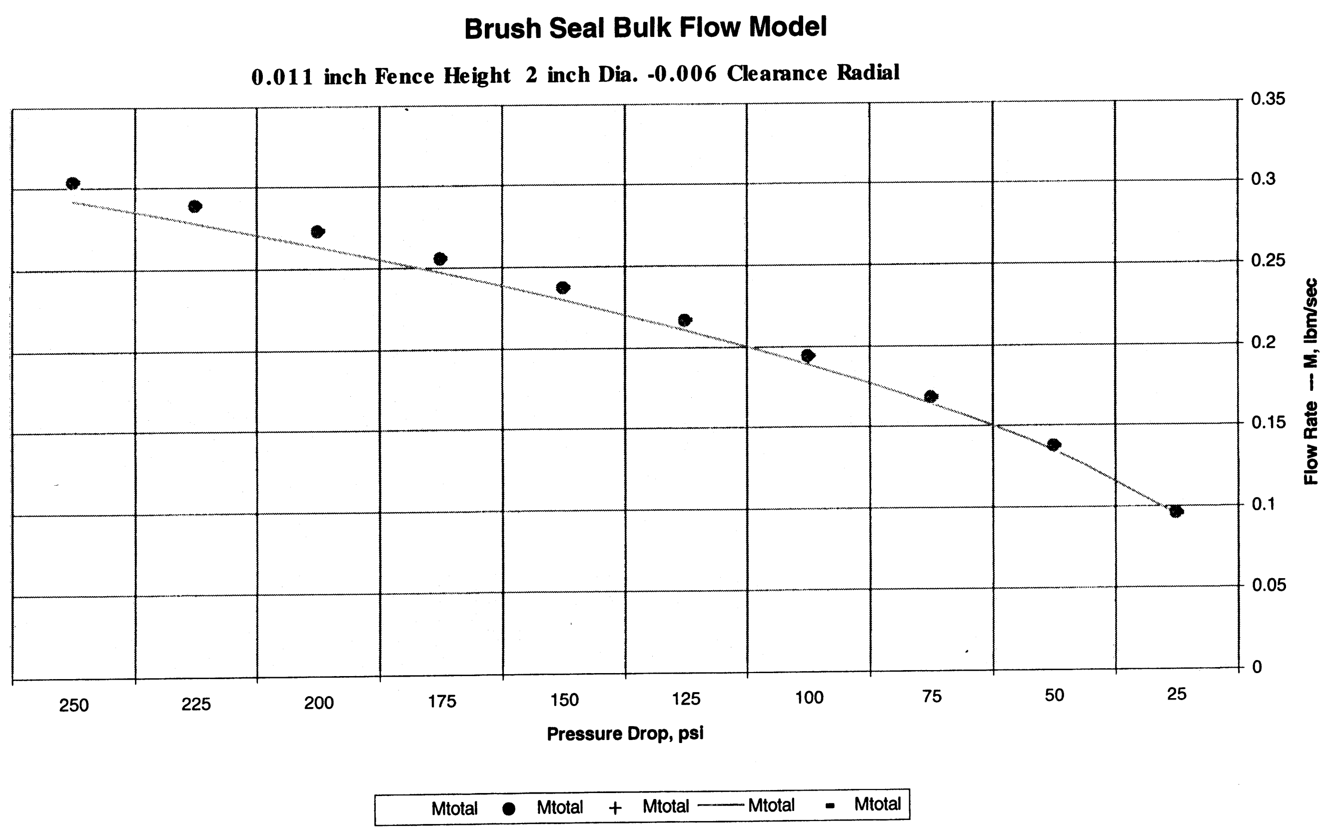
Brush Seal Bulk Flow Model Developed and Anchored

- Hendricks model used for final analysis
  - More thorough analysis method
  - Available NASA Lewis RC test data in hydrogen
  - Tested brush seal geometry closer to turbopump requirements
  - Provides leakage estimates for the three flow paths identified in brush seal
  - Heat generation estimated from bristle normal force and coefficient of friction
  - Anchored to NASA Lewis RC published test data

Slide 13 – Brush Seal Bulk Flow Model Developed and Anchored

The model presented by Hendricks et al. is used to build an Excel spreadsheet because of the thorough analysis and the already available test data. Also, the tested seals are similar in geometry to turbopump seal requirements. The analysis spreadsheet provides leakage estimates for three flow paths; flow between the bristles, flow between the bristles and the seal dam, and flow underneath the bristles. The model also estimates heat generation from the bristle normal force and the coefficient of friction. Before it is used as a design tool, the analysis is anchored to the published test data from NASA Lewis RC.

Bulk Flow Model Validation



Slide 14 – Bulk Flow Model Validation

This chart shows how well the analytical model created in Excel compares with the LN2 experimental test data for the fence height, shaft diameter and radial clearance.

Brush Seal Design Methodology

- Begin with parameters similar to tested seals (bristle lay-angle, bristle interference, fence height)
    - Decrease interference fit to reduce bristle wear and heat generation
    - Decrease fence height to decrease leakage

Slide 15 – Brush Seal Design Methodology

While developing the analysis method, it is learned that the analysis is most sensitive to the bristle pack thickness. Therefore, large variations in the interference or lay angle would likely render the analysis invalid. To avoid this, the seal design is intentionally made to be similar to the already tested seals used to anchor the analysis model. The interference is decreased (from 0.005 to 0.002 inches) to avoid the possibility of wear and the fence height is lowered to decrease the seal leakage. With only these slight changes, it is expected that the analysis is still valid.

Final Design Description

Shaft Diameter	**5.000 inches**
Bristle Radial Clearance	**-0.002 inches**
Seal Fence Height	**0.010 inches**
Bristle Lay Angle	**50°**

Slide 16 – Final Design Descriptions

The final brush seal design dimensions for the two pumps are as follows; with a shaft diameter of 5.000 inches, the bristle clearance is a negative 0.002 inches radial. The fence height is 0.010 inches from the shaft radial. The bristle lay angle is 50 degrees. All of these values are subject to manufacturing tolerances so the installed dimensions may vary somewhat.

Recommendations

- Investigate brush seal operation in liquid oxygen
- Investigate stability of clearance fit brush seals (generation of cross-coupled forces?)
- Develop eccentric operation model

Slide 17 – Recommendations

In the interest of using brush seals on the pump side of the oxygen turbopump, it is recommended that brush seal operation in oxygen be investigated. Perhaps it is useful to use a clearance fit brush seal on the oxygen side that would only generate heat (from friction) in the event of a rub. It is also recommended that the rotordynamic stability of clearance fit brush seals be investigated since a circumferential flow could possibly develop in the clearance portion of the seal. Finally, it is recommended that a model be developed to predict brush seal behavior for eccentric operation.