ABOUT THE HIGH-SPEED MECHANICAL SEAL

Georgeta Gabriela Chindlea
University of Oradea, chindlea@uoradea.ro

Keywords: rotary seals, mechanical seals, aerospace.

Abstract: This paper is abouthigh-speed mechanical seal stability and the tendency to dynamic stability. The seals were test in different conditions of temperature, to pressure of 16 bar, and speed of 13500rpm. Also, it was a parametric variation of damping, face coning, eccentricity and out- of- squareness.

Introduction

There are a lot of turbo-pumps deliver fuel which present a difficult sealing problem. The high levels of acceleration and the ambient vibration (sometimes a fluid phase change within the seal), the very low fluid viscosities, make the full film lubrication very difficult. The sliding speeds may exceed 100 ms^{-1} so, the contact is very serious.The ambient temperatures can balance from cryogen conditions to 600 ^oC. The object of the experiment is to exam the behavior of the seal under different condition associated with dynamic instability.

Test conditions

The seal in study has been used in a turbopumps of moderate speed. How we can see, these consist of a bellows – mounted nonrotating seal face assembly and a rigidly-clamped rotating mating ring, with a hard chromium face.The sealing face was impregnated with carbon graphite insert.

The dynamic instability can be caused by:

- phase change within the seal;
- ambient vibration;
- alignment tolerance arising from manufacture and assembly

Analytical investigation suggested that hydrogen pump seals are likely to see vapor troughout. In any case, it is considered that the destabilizating effect of a phase change should be avoided by design.

Some of the experimental parameters investigated are:

- initial, pre-lapped face coning;
- mating ring face out-of-squareness;
- mating ring eccentricity in conjunction with face coning;
- Coulomb friction damping force.

We can see the values of the above presented in Table 1.

Table 1.

Test number	Shaft speed N [rot/min]	Temp. of sealed gas T_s [^oC]	Sealed pressure p_s [MPa]	α* [μm]	δ** [μm]	e*** [mm]	F_d**** [N]
1-3	13500	20	5, 10, 15	1,3	0	0	
4	13500	200	5	1,6	0	0	
5	13500	200	10	1,2	0	0	
6	13500	200	15	1,6	0	0	
7-9	13500	20	5, 10, 15	1,6	0	0	
10-12	13500	20	5, 10, 15	1,9	0	0	
13	13500	20	2	2,5	0	0	
14-16	13500	20	5, 10, 15	1,2	28	0	
17-19	13500	20	5, 10, 15	1,3	56	0	
20-22	13500	20	5, 10, 15	1,3	84	0	
23-25	13500	20	5, 10, 15	1,2	0	0,055	
26-28	13500	20	5, 10, 15	1,2	0	0,115	
29-31	13500	20	5, 10, 15	1,2	0	0,150	
32-34	13500	20	5, 10, 15	1,2	0	0	
35-37	13500	20	5, 10, 15	1,2	0	0	
38-40	13500	-150	5, 10, 15	1,2	0	0	

Nomenclature:

*α - initial pre-lapped taper on mating ring in terms of difference in resulting film thickness;

**δ - mating ring out of squareness on shaft in terms of resulting peak-to-peak axial runout at seal face;

***e - mating ring eccentricity on shaft;

****F_d – Coulomb frictional damping force.

The ambient conditions of testing (Table 1), were selected for practical convenience, to minimize instrumentation problems.

Fuel turbopumps operate at different speeds, fact reflected in the choice of the test speed(13500 rot/min)

To reduce load on the bearings, it was preferred for test a beak-to-beak design . The principal reason of this choice was that measurement of leakage and torque from one individual seals becomes very difficult.

Sealed gas pressure, sealed gas temperature, leakage chamber pressure, shaft speed, stator carbon insert temperature, seal torque, stator ring axial displacement are somes of measurements taken. Temperatures limitations restricted displacement probes to room temperature testes and low radial – clearance precluded displacement probes from eccentricity tests.

Seal frictional torque was calculated by subtracting from the total torque the previously measured windage and lip seal torque at test speed.

Some of the standard test procedure steps were:

- Seal face and mating ring preparation;
- Seal and mating ring assembly into rig;
- Start-up slow and then run-up to speed over about 30 secons;
- Monitoring the system;
- Testing, about an hour;
- Run-down in steps of 2000 rpm;
- Disassembly and examination of seal faces (visual and profilometry measurement)

The results consist of derived parameters: face temperature rise, friction coefficient, the degree of fluctuation in leak rate, vibration amplitude. A graphical presentation is exampled in fig.1

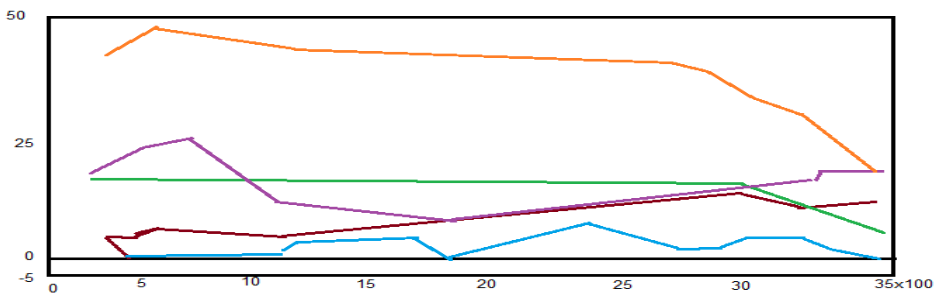


Fig.1

----lakerate ;--- face temperature rise (^{0}C);

--- speed (rot/min);--- 100x friction coefficient;

--- vibration amplitude; --- lakerate fluctuation (mg/hour)

A very condensed summary of test data is present in table 2

Table 2

Test	Δp_s [MPa]	N [rot/min]	F_d [N]	δ [μm]	e [mm]	α_f [μm]	α_m [μm]	Q [W]
1	0,413	13500	2,2	0	0	0,4	1,6	11
2	0,872	13500	2,2	0	0	0,4	1,6	15
3	1,432	13500	2,2	0	0	0,4	1,6	11
4	0,472	13500	1,3	0	0	0,1	1,4	46
5	0,633	13500	1,3	0	0	0,6	1,6	89
6	0,613	13500	1,3	0	0	0,1	1,4	47
7	0,571	13500	0,7	0	0	0,3	1,0	33
8	0,849	13500	0,7	0	0	0,3	1,0	34
9	1,226	13500	0,7	0	0	0,3	1,0	39
10	0,445	13500	2,2	0	0	0,2	1,6	?

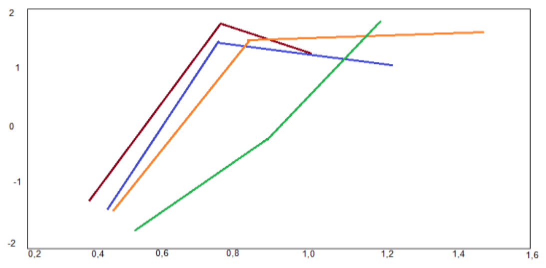
11	0,901	13500	2,2	0	0	0,4	1,6	?
12	1,187	13500	2,2	0	0	0,2	0,8	?
13	0,189	13500	2,2	0	0	1,6	0,9	21!
14	0,484	13500	2,2	28	0	0,4	1,6	8
15	0,902	13500	2,2	28	0	0,4	1,6	14
16	1,301	13500	2,2	28	0	0,4	1,6	4
17	0,399	13500	2,2	56	0	0,4	2,0	8
18	0,810	13500	2,2	56	0	0,7	2,0	6
19	1,036	13500	2,2	56	0	0,7	2,0	4
20	0,444	13500	2,2	84	0	0,8	1,8	6
21	0,784	13500	2,2	84	0	0,8	1,8	11
22	1,260	13500	2,2	84	0	0,8	1,8	5
23	0,334	13500	1,3	0	0,055	0,4	0,3	?
24	0,623	13500	1,3	0	0,055	0,4	0,3	17
25	0,801	13500	1,3	0	0,055	0,4	0,3	28
26	0,238	13500	1,3	0	0,115	0,2	0,4	23
27	0,605	13500	1,3	0	0,115	0,2	0,4	19
28	0,662	13500	1,3	0	0,115	0,2	0,4	22
29	0,387	13500	1,3	0	0,150	0,3	0,6	10
30	0,559	13500	1,3	0	0,150	0,3	0,6	17
31	0,722	13500	1,3	0	0,150	0,3	0,6	20
32	0,444	13500	1,3	0	0	0,4	1.2	?
33	0,669	13500	1,3	0	0	0,6	1,3	18
34	0,841	13500	1,3	0	0	0,6	1,3	22
35	0,409	13500	0	0	0	0,6	2,0	9
36	0,741	13500	0	0	0	0,6	2,0	35
37	1,025	13500	0	0	0	0,6	2,0	36
38	0,497	13500	0,6	0	0	0,3	1,6	107
39	0,471	13500	0	0	0	0,6	2,0	144
40	0,449	13500	0,6	0	0	0,3	1,6	55

The test temperature was about 200 ^{o}C (test: 4,5,6), -150^{o}C (test:38,39,40) and approximately 40^{o}C for the rest.

Conclusions

At first , in tests with moderate ring coning , was attained the stable equilibrium.

Leakage tended to increase with sealed differential pressure, but appeared independent of runout (fig. 2).



Sealed pressure differential

Fig.2

++++ medium; ++++ high; ++++ no runout; ++++ low.

The seal was tolerant of the combination of mating ring coning and eccentricity.

With changing the amplitude of damping no significant change in performance was noted (fig.3).

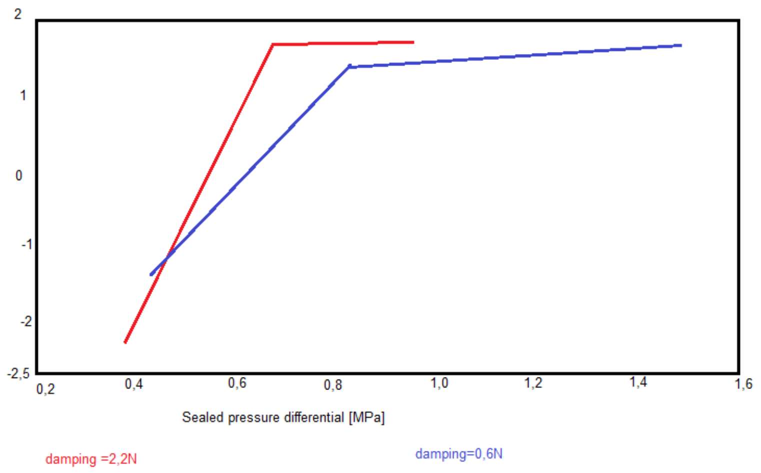


Fig.3

Reference:

1. Burcham,R. and Diamond,W. “High Pressure Hot Gas Self Acting Floating Ring Seal for Liquid Rocket Turbopumpe” NASA Cleveland, 1980
2. Hamm,R. and Shapiro,W. “Testing of Helium Buffered Releigh Step Floating Ring Seals” Lubrication Engineering, 43, 1986
3. Lefee,S. Hobson,R. “Turbopump Face Seal Technology” BHR Group Ltd Cranfield England.