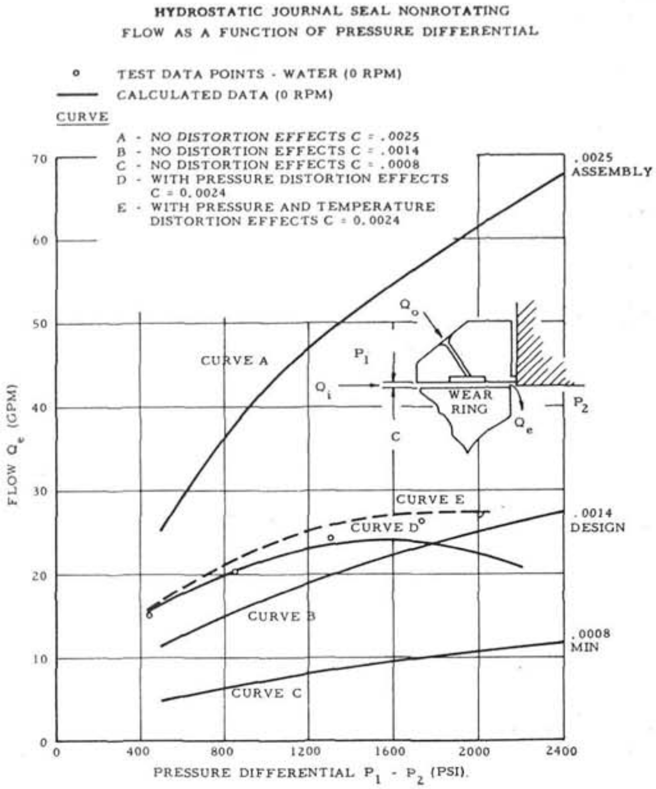

Fig. 17 Hydrostatic journal seal water flow data correlation with estimated flows

of the latter two curves is that the test seal had an initial radial clearance of 0.0024 in. and was analyzed to result in a running clearance of 0.0014 in. at a 2500-psi pressure differential and 30,000 rpm. The significance of the 0.0008-in. clearance curve is that, with design optimizing the seal cross section, it is expected that the seal flexibility could be increased, thereby resulting in a running clearance and thus a leakage flow comparable to those of a rigid seal with only 0.0008-in. clearance from an initial clearance of about 0.0025 in. In addition, because the seal is floating and is flexible, its ability to follow radial movements of the rotating wear ring without contact is excellent.

Also, since a close running clearance could be achieved even with relatively large buildup clearances, the radially flexible hydrostatic journal seal would lend itself to volume production without costly tight tolerance requirements. This is particularly important in commercial pumps.

Conclusions

1 The hydrostatic face seal and the straight labyrinth with Vespel SP-21 inserts limited the wear-ring flow during the tests in oxidizer to very low values at the required 4000-psi pressure differential. Both withstood intentional attempts at rubbing at 600-fps surface velocity (40,000 rpm) without causing fire or explosion and were in reusable condition.

2 A number of the other plastic inserts for labyrinths passed water tests but failed at the 4000-psi pressure differentials in N_2O_4.

3 Two straight labyrinths with inert inserts (Kynar and Kel-F) were satisfactorily tested in fuel at pressure differentials of 2250 psi. Both withstood rubbing at 820-fps surface velocity (40,000 rpm) without causing fire or explosion and were in reusable condition.

4 The stepped labyrinth with inert inserts did not appear to be practical for the high-pressure high-speed pump wear-ring application.

5 The hydrostatic journal seal passed the water tests and shows considerable promise. It was eliminated because the test program had funds sufficient to test only two concepts in oxidizer. Because of the radially flexible feature of this design, very low and constant leakage rates could be achieved over wide pressure differentials. It would appear that this seal could have good commercial potential.

6 All-metal hydrostatic seals would have the advantages of storage life and relative insensitivity to pressure. Properties of the plastic inserts will deteriorate with time and successive propellant exposures. Also the plastic inserts are pressure limited, i.e., at some pressure differential depending upon the material properties and the retention mechanism, they will be extruded. Use of the all-metal hydrostatic seals would eliminate these two problems.

Acknowledgment

The test evaluation program discussed in this paper was conducted by the Aerojet-General Corporation under an Air Force funded research and development contract. The contract was under the direction of Mr. C. W. Hawk, of Edwards Air Force Base. The authors wish to express their appreciation to Mr. Hawk and to the Aerojet-General Corporation for permission to publish these data. They also wish to acknowledge the contributions of Mr. W. N. Burkey in the test work, and the numerous valuable suggestions of Mr. J. J. Fleischmann in the paper preparation.

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DISCUSSION

J. E. Wolf[2]

The authors are to be congratulated for their efforts in advancing the state-of-the-art in the area of high-pressure wear ring seals. The writer's company has similar problems and it seems appropriate to point out some supporting work along the same lines.

As pointed out by the authors, a metal-to-metal rub in an oxidizer pump may be disastrous; consequently metal labyrinth-type seals must operate with relatively large clearances which adversely affect efficiency. In an effort to improve the performance, a Rocketdyne program was initiated in 1958 with the design and testing of a floating carbon wear ring. The carbon ring configuration consisted of a carbon ring supported in an aluminum housing. The carbon ring was free to float in a radial direction along with movements of the impeller and was retained from rotating by antirotation pins. The wear ring seal was tested on a pump and the improvement in efficiency, all due to reduced clearance, is shown in Fig. 18. Of equal importance was the improvement in cavitation performance shown in Fig. 19. Unfortunately, the ductility of the carbon ring was insufficient for the application and other materials were investigated. Currently the seal ring is made of Kel-F which works quite satisfactorily in LO_2 and in point of fact, the oxidizer pumps in the Apollo 7 booster engines and the oxidizer pump in the Apollo 7 second-stage engine were so equipped.

As the authors pointed out, even better performance can be expected using hydrostatic type seals. In 1964 an internally pressurized hydrostatic impeller seal similar to the authors' was designed and tested at Rocketdyne sealing kerosene at pressures

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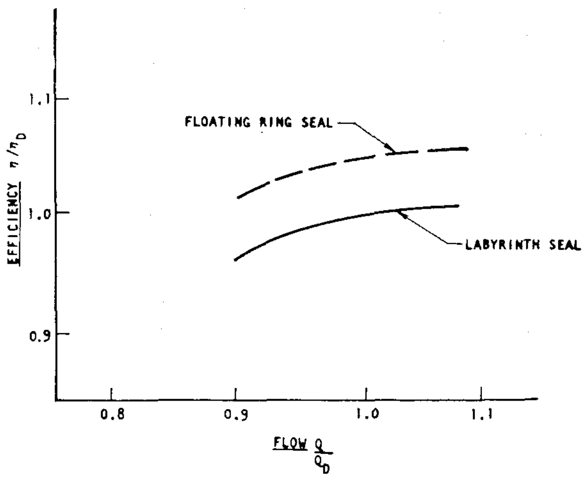

Fig. 18 Efficiency variation with seal type

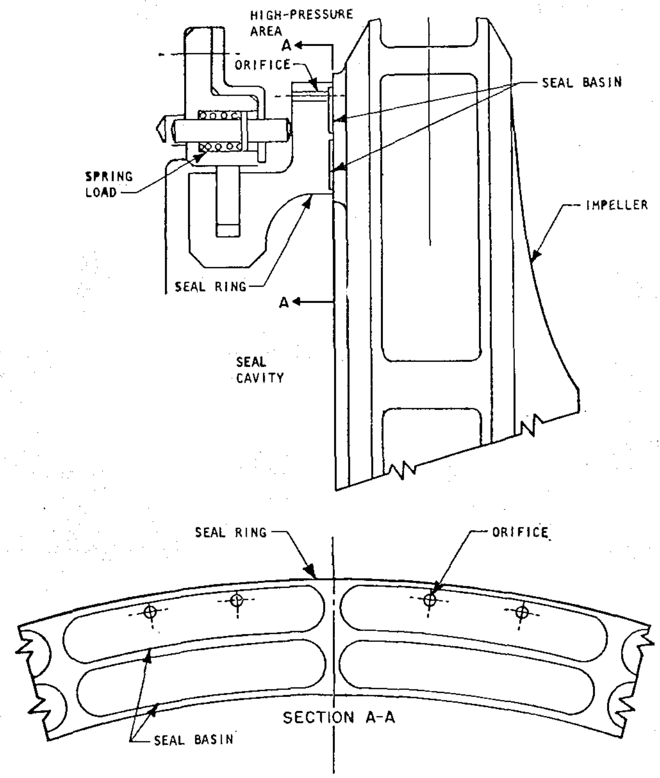

Fig. 20 Schematic of hydrostatic impeller seal

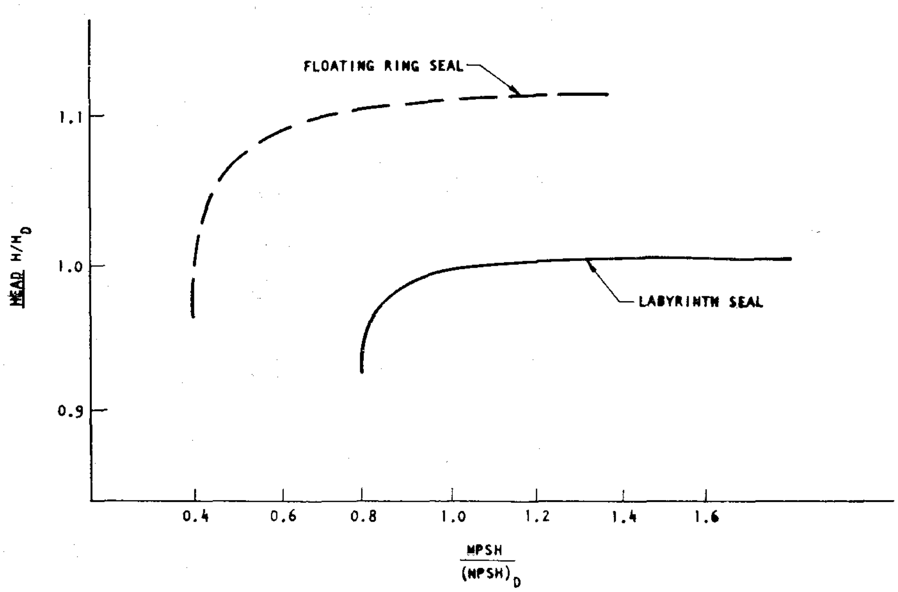

Fig. 19 NPSH variation with seal type

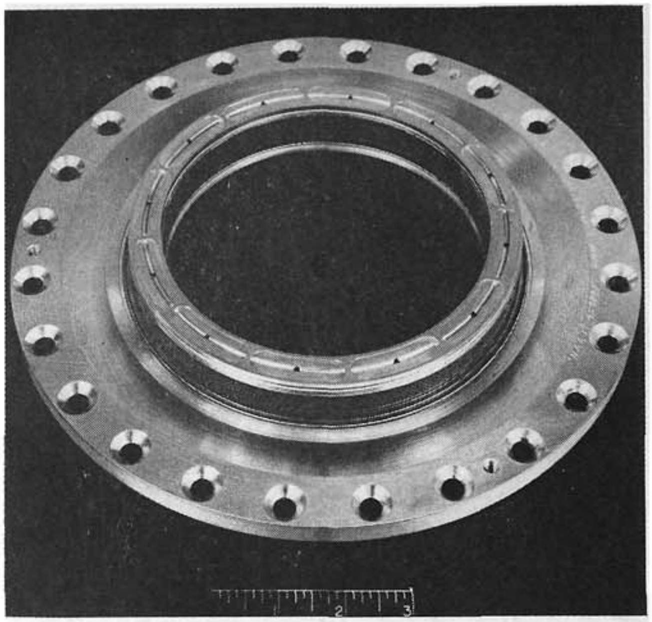

Fig. 21 Hydrostatic face seals

of 1000 psia and running at equivalent rubbing velocities up to 600 fps. Fig. 20 is a schematic of the seal showing the location of the basins, the orifices, and the secondary axial seal. Seal leakage at 1000 psia upstream pressure was 14 gpm at a diameter of $9^1/_2$ in. Concurrently an externally pressurized hydrostatic seal was also designed and tested and is shown in Fig. 21. It consists of a stationary circular ring into which a series of basins have been machined. A double bellows toroidal chamber connects the seal ring to the housing and produces a spring force to hold the seal ring against the rotating mating ring (not shown) prior to operation. During operation, a high pressure fluid is supplied to the basins which lifts the seal free of the mating ring and provides a fluid-film seal. The seal was tested to equivalent surface speeds of 600 fpm sealing a somewhat lower pressure of 250 psig. Both of the hydrostatic seals over portions of the operating range suffered from the quite common malady of hydrostatic devices known as instability. It would be interesting to have the comments of the authors on this as it appears they were not troubled by instability of the hydrostatic seals.

Authors' Closure

Mr. Wolf's commentary on his experiences in similar sealing applications is appreciated. The following is offered in answer to his request for our comments on instability in hydrostatic devices.

It is not our intent to include a mathematical treatise on the hydraulic and mechanical performance of hydrostatic devices in the paper, as it would be too lengthy. Therefore, in keeping with the original theme, we hope a general description will be sufficient comment on stability. Careful attention must be given to the steady state nominal operating conditions of the seal (or bearing) and the appropriate integration of the hydraulic and mechanical performance is necessary in the choice of geometry. Physically we design the hydrostatic device with adequate load capacity to overcome any forced excitation by the rotor. By proper pressure ratio choice a high static stiffness is assured, that stiffness being determined from the slope of the load deflection curve which is a function of the flow regulation as a result of slow quasi-static displacement. Referring to Fig 3 (N_2O_4 design conditions), one can see that if a 0.016 in. diam orifice had been selected for the hydrostatic face seal, the axial stiffness value would have been reduced from 4 million lb/in to about 1.5 million lb/in at the design operating clearance of 0.001 inch. By proper choice of pad geometry, which includes the recess depth and area, the dynamic stiffness is maintained high relative to the combined hydraulic and mechanical spring-mass system. To date we have had no problems with instability in our hydrostatic designs. This includes designs for both incompressible and compressible fluids. Recently a thrust balancer was successfully operated on nitrogen gas and a journal bearing was successfully operated on liquid hydrogen, which is a rather compressible fluid with a slight increase in temperature.

The two hydrostatic seals described in the paper were designed with adequate stiffness values and in addition, are damped highly by the piston ring in the case of the face seal and by the radial sealing dam in the case of the journal seal. We also have used a bellows attachment of the seal ring (similar to the Rockedyne seal of Fig. 21) in other designs. However, in these designs a torroidal bellows shape was selected in order to exploit the advantage they provide in minimun effective diameter shift. This assures minimum moment and thus minimum deflection of the seal ring and therefore, minimum mechanical hindrance of the seal ring following rotor excursions. The only problems related to instability we have experienced cannot be attributed to the bearing design. In this case the working fluid was supplied to a double acting hydrostatic thrust bearing from a multiple piston type pump. Improper assembly (too large a clearance) resulted in low load capacity and low stiffness. Then, with these undesirable characteristics existing, the fluid pulsations transmitted from the pump pistons excited the bearing rotor and excessive vibration was experienced. The addition of an accumulator in the fluid supply line and proper assembly of the thrust bearing resulted in predictable performance.

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