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AMMONIA TRANSFER ACROSS ROTATING JOINTS IN SPACE

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ABSTRACT

Thermal control of future large space facilities, such as the Space Station, will require the transfer of anhydrous ammonia across rotating joints with near zero leakage. Anhydrous ammonia is the primary heat transfer fluid aboard the station, providing critical thermal management of habitat and payload systems. The solar radiator joints, as well as the various payload pointing systems, are obvious examples of the need for a reliable fluid transfer device. Low weight, tight temperature control, low parasitic drag torque, long life, and high reliability, in addition to near zero leakage, are important characteristics necessary for the successful operation of such a device. In late 1986, Lockheed initiated an IR&D project to develop a Rotary Transfer Coupling (RTC) directed toward Space Station requirements. Fabrication and assembly of this device is now complete and testing is scheduled for January 1988. This paper addresses the design and development of the face seal-type rotary fluid coupling that utilizes a unique 'cover gas' concept (an inert gas such as nitrogen) to provide full containment of the ammonia.

INTRODUCTION

The combined requirements of extremely low leakage, low frictional torque, and long life eliminate many of the rotary transfer methods considered in the past. Flexible hoses, though simple, experience relatively high ammonia diffusion rates in vacuum environments. In addition, pressurized hoses exhibit dramatic increases in drive torque at angular travel greater that 120-150 degrees. Also, hose-type devices either do not allow applications that require continuous rotation, or require the use of complicated 're-wind' mechanisms.

An O-ring type rotary coupling, as was used for Lockheed's Talon Gold Gimbal

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Fluid Coupler, was rejected due to its relatively low life and excessive friction. This unit satisfied most of the performance objectives during testing, but operating torques were considerably higher than desired. Friction non-linearities place a high burden on gimbal controllability and can compromise both pointing accuracy and stability (jitter).

Other seal-type couplings were rejected for similar reasons: Lip seals, because of their high leakage rates; Ferro-fluidics, because of the increased complexity of the design and the multi-staging required to accept high pressure-differentials. Of the remaining available methods of sealing, a carbon graphite mechanical-type face seal best met the requirements of leakage, friction, and life, and was therefore selected as the baseline design (Figures 1-3).

This design approach was based on previous work with rotary coupler and heat pipe design, technology resulting from research at the Langley Research Center (LaRC), and the excellent industrial service record of ammonia pumping refrigeration equipment. The design includes novel solutions in the arrangement of multiple axial face seals, the use of 'scavenger' channels to facilitate internal leakage paths, and the use of a pressurized, nitrogen gas "cover" to eliminate external ammonia leakage.

DESIGN DESCRIPTION

As shown in Figure 4, the IR&D design uses annulus-type channels to provide the transfer capability of the coupling. The coupling was designed to accommodate the Space Station Radiator requirements shown in Figure 2. The device has the capacity to transfer eight separate lines/channels of anhydrous ammonia or, with minor modifications, other refrigerants such as monomethyhalamine or the common halocarbons. The RTC is approximately 14 inches in length, 5 inches in diameter, and has channels of two separate size: 0.688 inches in diameter for the vapor supply, and 0.438 inches in diameter for the liquid return.

The modular rotor and stator segments are machined from CRES 316 for corrosion resistance to the anhydrous ammonia. During the layout stage of the design, particular care was taken to minimize the overall size of the coupling, especially the diameter. This not only reduces the overall weight and the seal rubbing velocities, but causes a direct reduction of parasitic drag torque. In addition, the RTC assembly was sized for typical shuttle launch loads with adequate bearing load margins as shown in Figure 5. The IR&D coupling uses standard 52100 steel (440-C steel for flight) deep groove-type radial ball bearings, which are spring preloaded with a steel wave spring washer to accommodate thermal expansion. PTFE (Teflon) gaskets provide the static sealing between individual parts and were machined to account for any cold-flow during installation.

In the RTC design, each of the 8 stationary carbon-graphite face seals (grade RP-8290) is positioned by a shrink-fit, stainless steel retaining ring which is keyed to prevent rotation. This arrangement forces the rotary sealing interface between the rotating shaft runner and the carbon graphite. The static secondary spring seal (also of PTFE) provides a seating force against the carbon graphite seals for start-up operations, in addition to acting as a radial and axial back-up seal. Because of the seal's compliant nature, tolerancing of parts is less critical. The carbon graphite seal is micro-lapped to a flatness of 1-2 lightbands (1.6 micro-inches RMS) to ensure full seal contact. This flatness is verified for each seal using a monochromatic light and an optical flat. The runner is a CRES 316 substrate with a 0.005 inch thick coating of tungsten carbide applied by a detonation gun. The tungsten carbide is sealed with a UCAR 100 epoxy for protection against bond layer corrosion and is also micro-lapped to 1-2 lightbands flatness.

As shown in Figure 6, the seal orientation allows the relative pressure differential between channels to provide a seal seating force in addition to that of the secondary seals. The actual seating force was a compromise between sealing pressure and the parasitic friction due to drag. All materials were selected for compatibility with anhydrous ammonia and for their very low out-gassing characteristics.

COVER GAS CONCEPT

A key design feature of the RTC is a scavenger/cover gas system. As shown in Figure 7, on both sides of the ammonia channels are scavenger channels. These channels provide internal containment of any leakage from the adjacent liquid or vapor ammonia channels. Outboard from these scavenger channels are nitrogen charged barrier annuli at a slightly higher pressure of 4 psid. This nitrogen is either supplied from a reservoir or carried in a separate line/channel to the coupling. With the nitrogen cover gas system, any ammonia leakage is contained within the scavenger line/channel and is not dissipated externally from the coupling.

PREDICTED PERFORMANCE

The five primary coupling performance characteristics are leakage rates, frictional torque, pressure drops, thermal cross-talk, and life. The performance of the RTC was predicted based primarily on two methods: analysis, and correlation to similar carbon face seal devices such as the LaRC rotary coupler. In addition, data taken from industrial experience with sealing ammonia in refrigeration systems provided a reference during design.

The individual lines and annuli channel sizes are based on both flow rate and pressure drop requirements of the Space Station Radiators. As shown in Figure 8, the channels are sized to meet the RTC pressure drop requirements with margin. The predicted pressure drop was calculated using standard Mollier diagram-type analysis. The maximum pressure drop occurs at the maximum mass flow rate. This maximum mass flow rate occurs at the latent heat of the fluid saturation temperature. Note that the pressure drop across the RTC is a function of rotor-to-stator relative position. This is due to the change in direction and flow length as the rotor turns.

Figure 9 shows the total RTC drag torque (friction). For applications such as the various payload pointing systems on the Space Station, frictional torque must be kept to a minimum. As stated earlier, running friction, in addition to the non-linearities of start-up-friction, can compromise pointing accuracy and stability. This friction is due to a combination of both seal and bearing drag. The seal friction is due to the seating forces exerted by both the secondary seal and the vapor/liquid pressure differential between channels. Table 1 shows the coefficients of friction values for carbon graphite sliding on various materials. A value of 0.08 was selected for the performance analysis.

Depending on the application, very low thermal cross-talk between channels may be required. Radial and axial thermal conductance and convection between the liquid and vapor/liquid were calculated as a function of rotor position. Thermal conductance is at a maximum when the rotor port is 180 degress from, or opposite to, the stator port; the the heat transfer is then completely around the rotor annulus.

Life of the RTC is predicted to be a minimum of 120,000 revs over 10 years. This prediction is based both on analysis (PV factors) and from LaRC accelerated life test data. Volumetric wear of the carbon graphite can be calculated as follows:

$$\text{Wear} = \frac{\text{(Wear Coefficient)(Seating Force)(Sliding Distance)}}{\text{(Material Hardness)}}$$

Table 1 shows typical values for wear coefficients. Hardness of the carbon graphite (grade RP-8290) is approximately 450 Vickers. Seal wear is thus analytically predicted as 9×10^{-5} in^3/seal-year. This is equivalent to a thickness change of 5.6×10^{-5} in/seal-year. LaRC test data show lives in excess of 22 equivalent Space Station years (111,000 revs over 60 days continuous operation). LaRC disassembled and inspected the rotary coupling following this test and found extremely low wear of the carbon graphite face seals and corresponding runner.

Cross leakage on the order of 1×10^{-4} cc/yr is virtually impossible to predict accurately due to the many second order factors influencing it, including surface flatness/finish, and thermo-mechanical distortions. This value, as well as external leakage, must be measured because it is so small.

DISCUSSION

Certain problems can arise when dealing with carbon graphite seals. Caution is necessary during the installation of the steel retaining rings which surround the carbon. Cracking of the brittle carbon graphite is possible due the shrink-fit nature of the retaining ring. A slight interference fit (0.0002-0.0003 in) allows adequate pressure without damaging the seal. When using larger interference fits, care must be taken to avoid over-stressing the carbon graphite.

When using PTFE (Teflon) type gaskets and O-rings for static seals, cold flow of the material must be accounted for. The ratio of final to initial bolt clamping force required for this 'creep' can be determined as follows:

$$\frac{\text{Final Clamp Force}}{\text{Initial Clamp Force}} = \frac{(53.3-190(\text{thickness})-0.2(\text{Temp.C})-33.7(\text{width})+3.6(\text{bolt length}))}{(100)}$$

It should be noted that 90 percent of the cold flow takes place in the first 24 hours.

The modular design of the RTC accommodates up to eight separate fluid/vapor channels by simply 'stacking' the required number of annuli together. One drawback, however, is the additive nature of the tolerance stackup. A minimum start-up seal seating force must be selected, and the corresponding minimum spring seal deflection determined from this. When summed, the individual part tolerances must not exceed the value that allows this minimum seating force.

The small diameter, seal anti-rotation pins pressed into the outside diameter of the retaining rings also presented a problem. The pins, which keep the carbon graphite seals stationary with respect to the coupling's stator, were sized for shear strength and not for stiffness. During initial assembly, the pins were deflecting excessively and were actually bending out of their mating holes on the retaining rings. A change to larger diameter pins was therefore required.

Designers of carbon face seals should be aware that a combined net force due to static seal spring forces and pressure differentials between channels is present and tends to move the rotor axially relative to the stator housing. The bearing preload spring (designed to accommodate thermal elongation of the coupling) was sized larger than this net force. If this force is larger than the restoring spring force, excessive seal wear or loss of seating force may be experienced, depending upon seal orientation.

Certain applications, such as capillary-flow devices (heat pipes), are very sensitive to thermal cross-talk between the liquid and vapor lines of the transfer couplings used. An RTC designed for such applications must minimize the conduction and convection paths between channels. The use of Teflon-type gaskets and channel liners, as well as a switch to the less thermally conductive titanium in place of stainless steel for the RTC's rotor, may be warranted. Another solution would be vacuum venting 'blocking/barrier' channels in the coupling. Large decreases in thermal transfer are possible by opening various passages inside the coupling to the vacuum environment of space.

Designers must also recognize the special problems of ground based verification of two phase flow. Differences in the coupler's performance between a one-g and microgravity environment are expected in fluid pressure drops and thermal crosstalk. Differences in fluid pressure drops between one-g and microgravity environments occur only in the two-phase flow channels. Single-phase flow pressure drops are identical in both environments, but two-phase flow acts differently in micro-gravity. Design of the passages should be based on a 100% vapor or liquid flow (whichever is greater) which is larger than a X% quality, two-phase flow. This will ensure that the design has a more conservative pressure drop than that expected for the two-phase flow in a microgravity environment.

The problem with differences in two-phase flow between one-g and microgravity environments should not largely affect the value of the thermal crosstalk. The thermal crosstalk between the liquid and vapor lines is comprised of convective and conductive heat transfer. The thermal resistance due to conduction is the main resistance between the liquid and vapor. Any change in the heat transfer characteristics of the two-phase vapor flow between one-g and microgravity environments has a minor influence on the thermal cross-talk.

TESTING

Although RTC test data are unavailable at the time of this writing, certain special test considerations are needed to evaluate the performance of an anhydrous ammonia coupling with very small ($< 1 \times 10^{-4}$ cc/yr) leakage rates. The test equipment includes a sealed chamber that surrounds the coupling. Individual rotor exit lines will be routed back into the coupling. The corresponding exit port on the stator-side of the coupling is then capped. Leakage can be tested by pressurizing the inlet port. The chamber can be vented through a water bath and the corresponding pH change measured.

Calibration of the test apparatus can be done by introducing a known quantity of ammonia into the water bath and then recording the pH change. Also, because of similar molecule sizes, helium may be substituted for the anhydrous ammonia to facilitate leakage testing. By using readily available helium 'sniffers', the problems associated with ammonia can be avoided. A drive motor will be configured for both uni-directional and oscillating motion. In this manner, life and leakage rates can be evaluated in both modes.

Due to the extremely small wear of the carbon-graphite face seals predicted, it is difficult to make wear-rate measurements. A possible solution would be accurately weighing the seals before and after the extended life testing. Frictional torque, on the other hand, is relatively simple to determine by using a torque transducer.

CONCLUSIONS

Designers of carbon-face-seal rotary thermal couplings should be aware of the various points/problem areas addressed in this paper. Particular emphasis should be placed on minimizing the overall size of the coupling; a small diameter RTC not only reduces weight and seal rubbing velocities, but also minimizes any parasitic drag torque present. This friction, when experienced on sensitive pointing gimbals and rotary joints, can dramatically affect accuracy and stability. Depending upon the application of the coupling, thermal cross-talk between channels may also be excessive and should be evaluated early in the design stage. The same evaluation should also be done for flow pressure drops across the coupling.

The orientation and sizing of the static secondary seals, which provide the required initial seating force, must also be addressed. Proper compliance between the seal and runner is needed to ensure adequate, but not excessive, axial force. This trade-off between minimum required seating force and seal drag torque must be evaluated.

A carbon face seal, rotary thermal coupling, which uses a pressurized nitrogen gas 'cover' and separate scavenger annuli as leak paths, appears to be an effective design for transporting ammonia across rotating joints in space. In addition to negligible external leakage, this RTC satisfies the Space Station requirements of low drag torque and weight, long life, high reliability, and low thermal cross-talk.

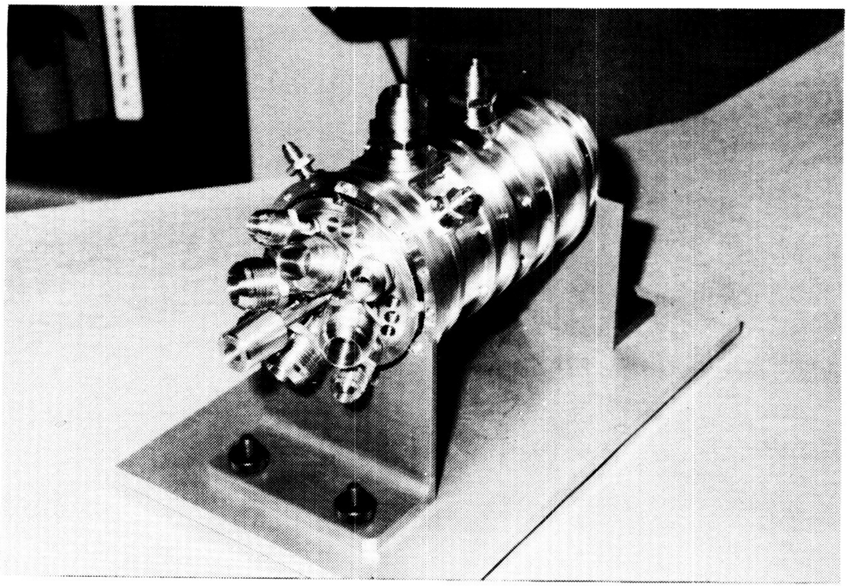
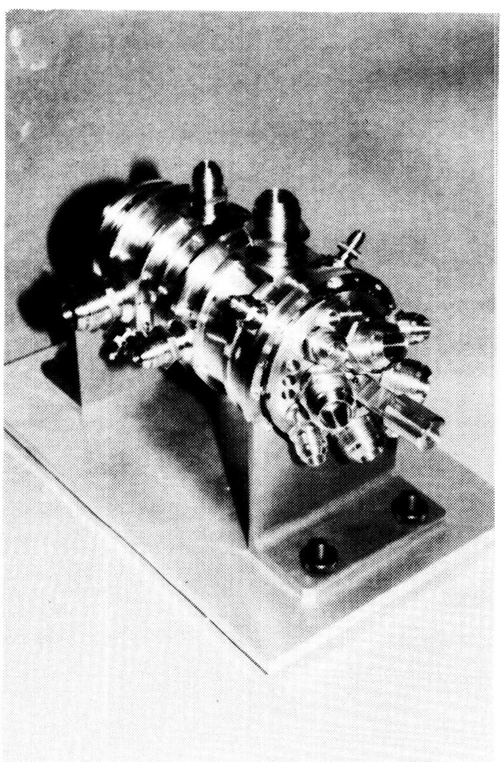


FIGURE 1. IR&D Rotary Thermal Coupler (RTC)



Space Station Radiator Joint Requirements

- -Vapor Supply, Liquid Return Lines
- -Ammonia Compatible
- -Max Flow Rate = 0.832 kg/s
- -Max Line Press. Drop = 685 Pa
- -Min Burst Pressure = 2.48×10^6 Pa
- -Min Non-Maintenance Life = 5 years

FIGURE 2. RTC Designed For Space Station Radiator Requirements

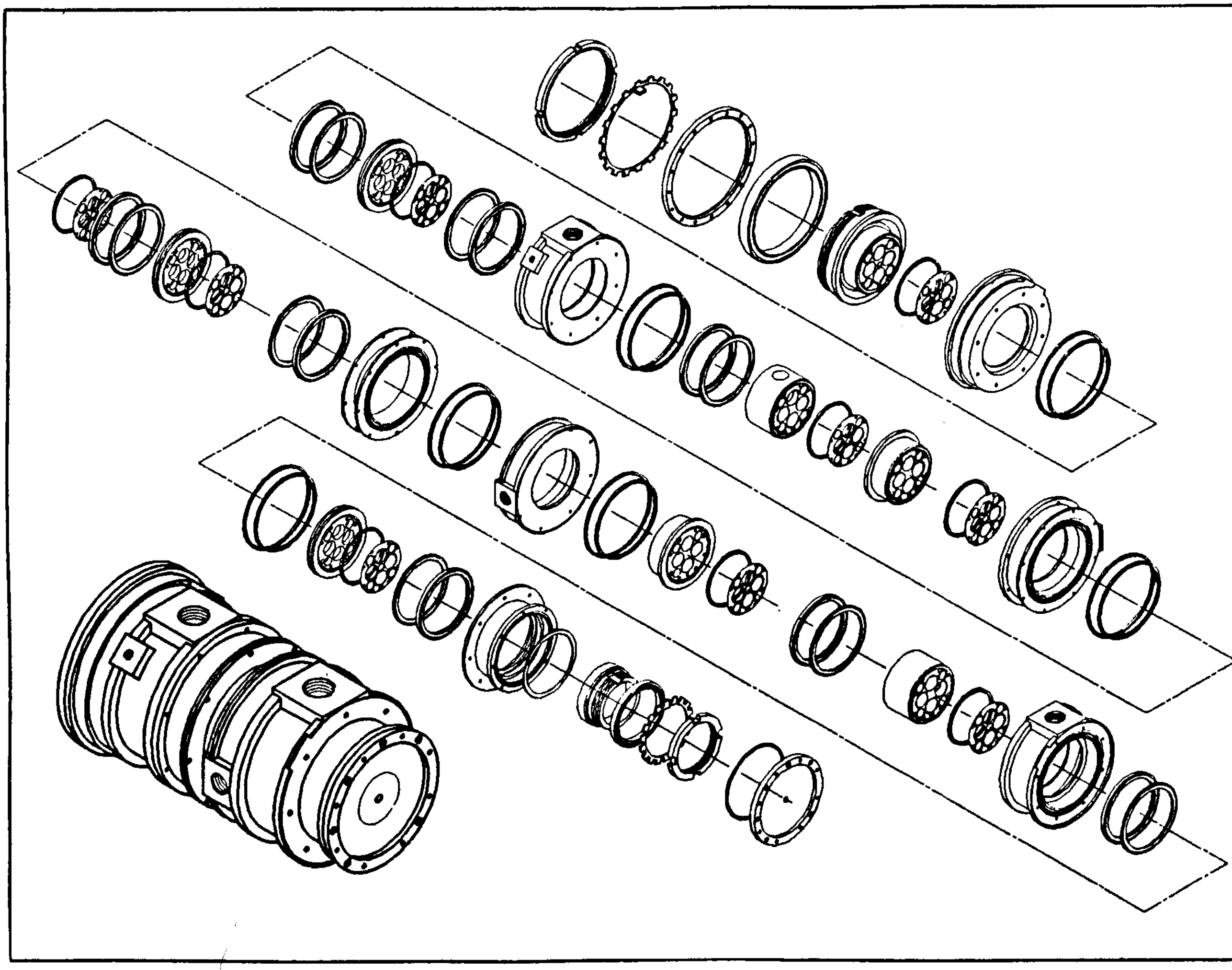


FIGURE 3. 'Disassembled' Rotary Thermal Coupling (RTC)

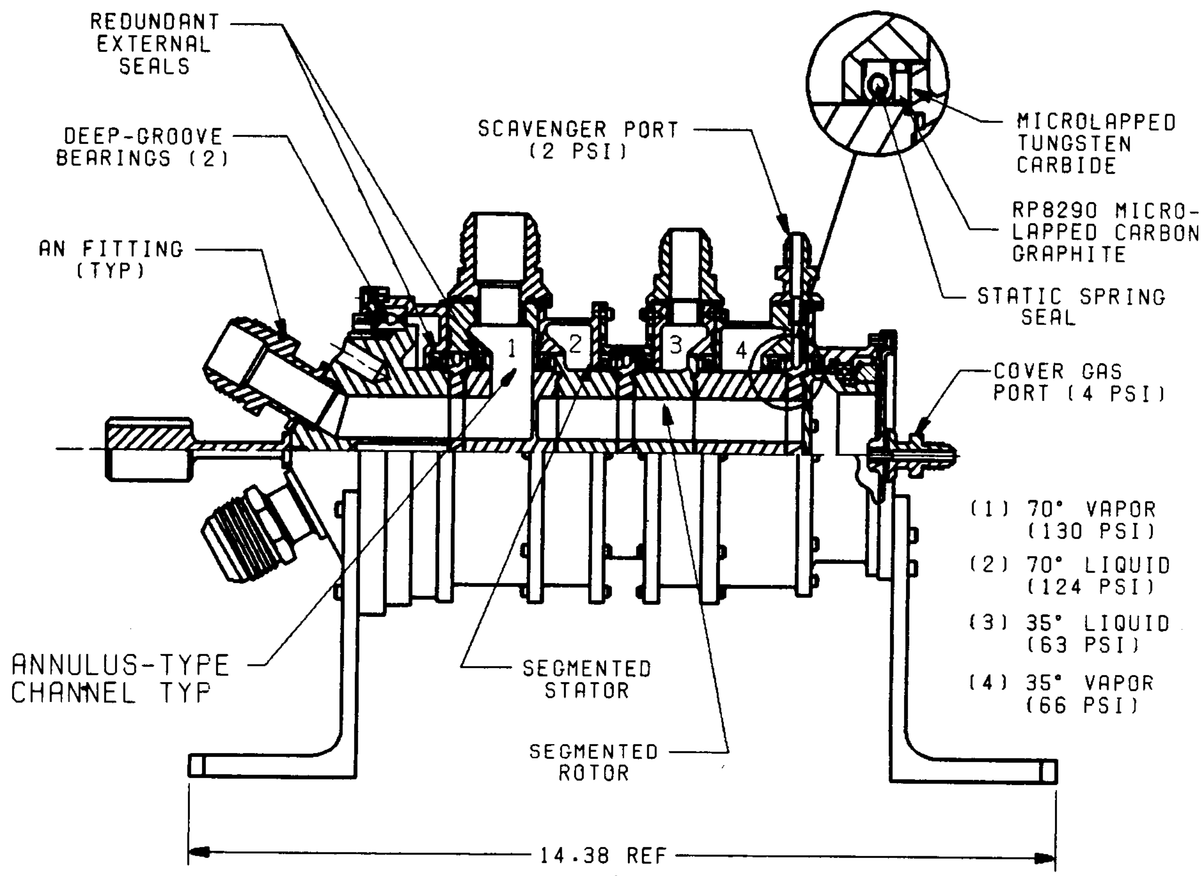
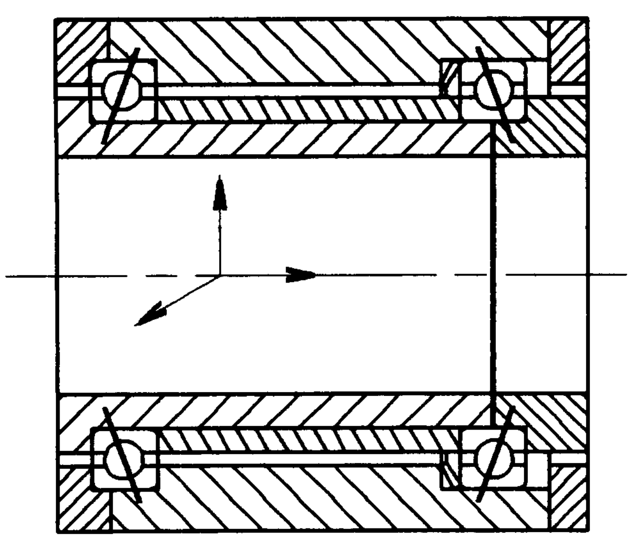


FIGURE 4. IR&D Rotary Thermal Coupler (RTC)



<u>Shuttle Ascent = 6 G's</u>

Assume Equal Load Sharing (Rotor = 9.1 lbm)
Radial Force = 54 Cos(45)/2 = 38 lbf
Axial Force = (54 +50)/2 = 52 lbf

Equiv. Radial Load = PR
PR = Radial Force +1.5(Axial Force) = 116 lbf

4" Bearing Ascent Load Margin = 640%
2.5" Bearing Ascent Load Margin = 210%

FIGURE 5. RTC Bearing Capacity Sized for Typical Shuttle Launch Loads

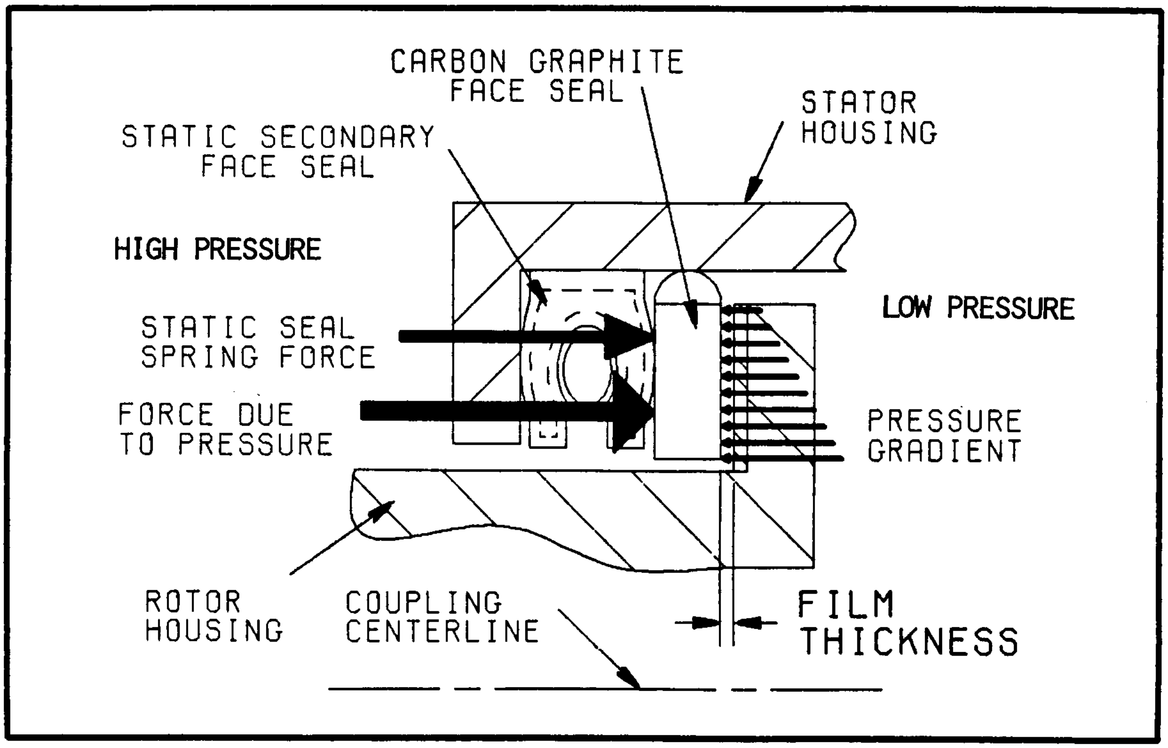


FIGURE 6. Carbon Graphite Seal Axial Loading

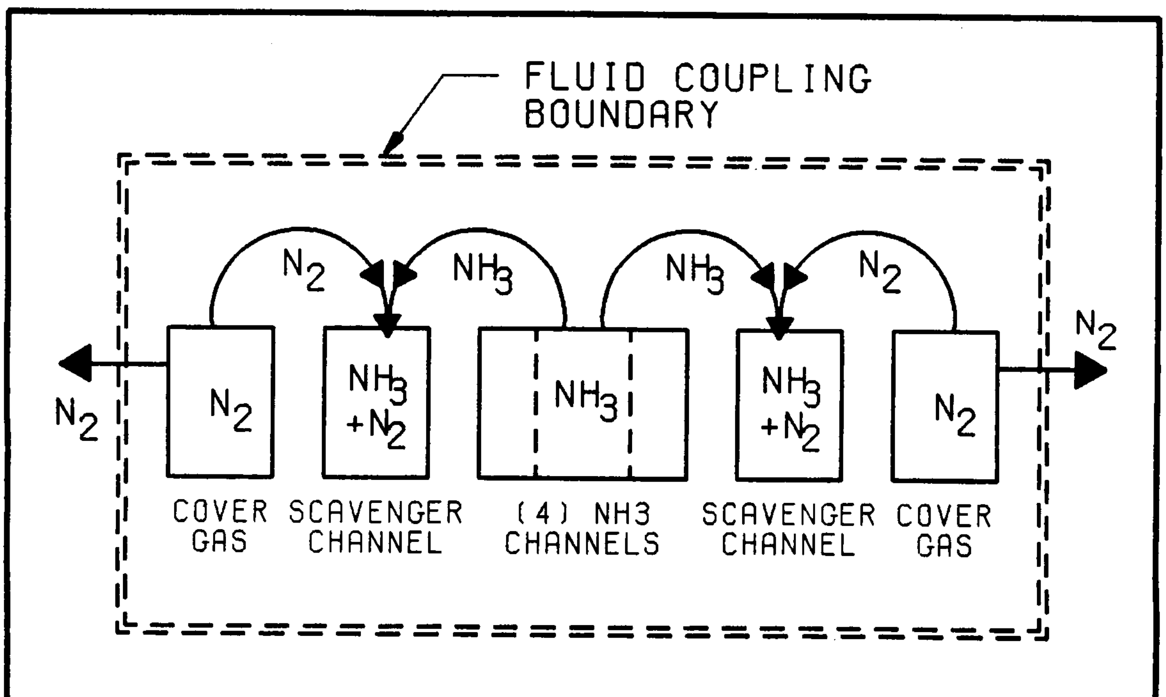


FIGURE 7. RTC Leak Path Schematic

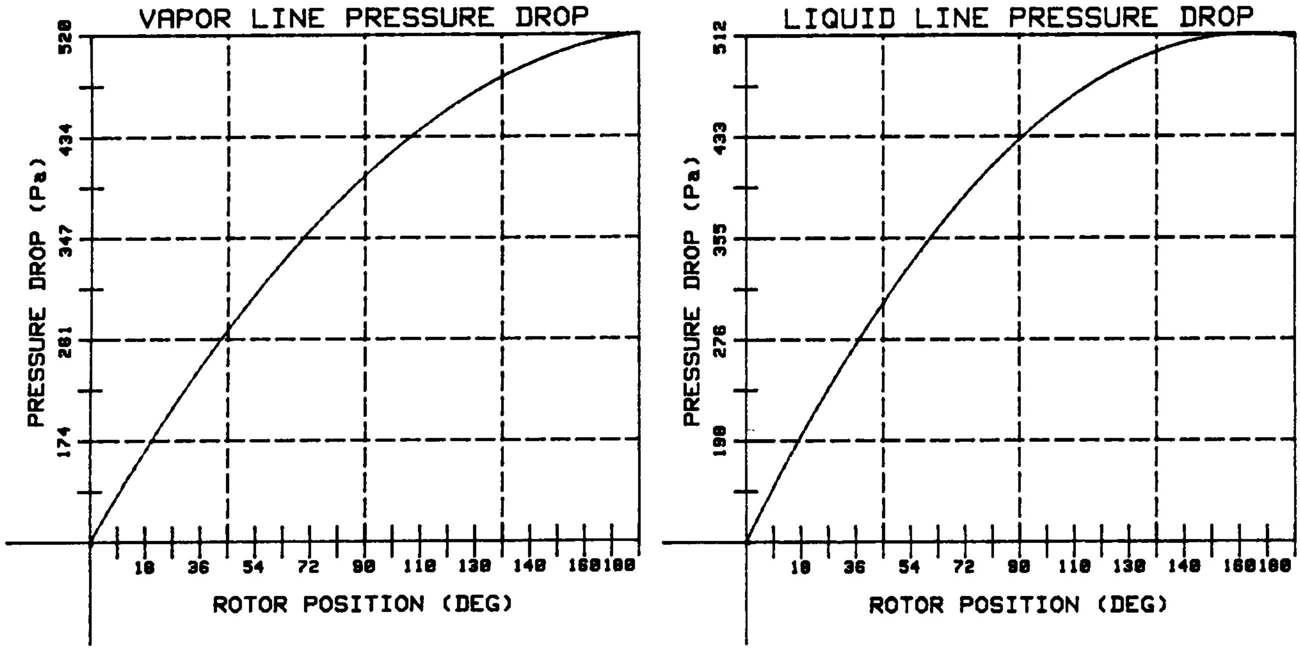


FIGURE 8. Pressure Drop as a Function of Rotor Position

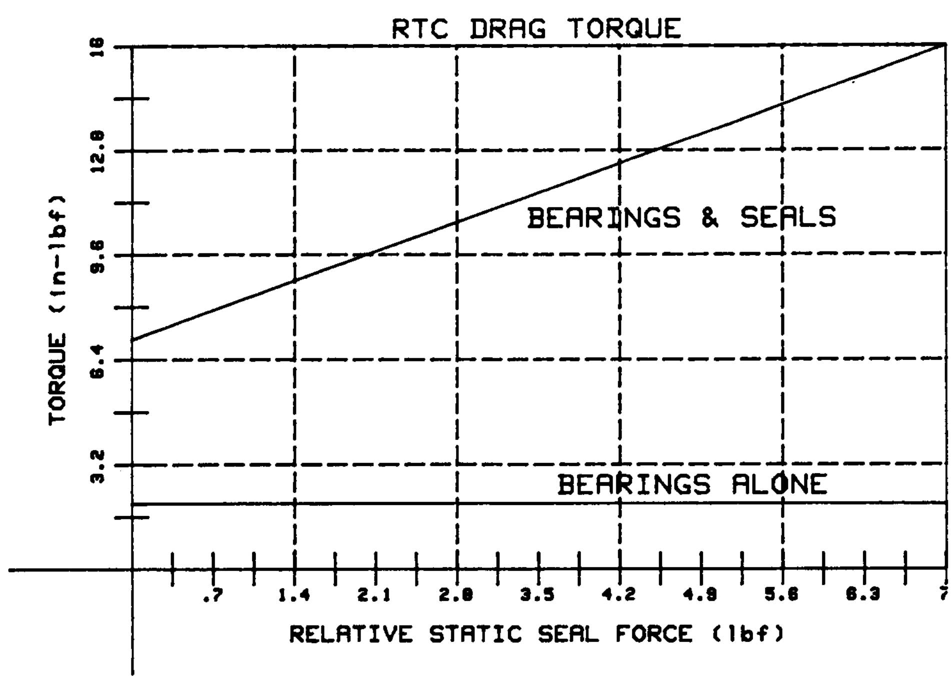


FIGURE 9. Rotary Thermal Coupling Drag Torque

Sliding Materials		Wear coefficient K	PV limit, bar · m/s	Friction coefficient f
Rotating	Stationary			
Carbon-graphite (resin-filled)	NI-resist cast iron	10^{-6}	35	0.07
Carbon-graphite (resin-filled)	Ceramic (85% Al_2O_3)	10^{-7}		
Carbon-graphite (babbitt-filled)		10^{-7}		
Carbon-graphite (bronze-filled)	Tungsten carbide (6% cobalt)	10^{-8}	175	
Tungsten carbide (6% cobalt)		10^{-8}	42	0.08
Silicon carbide converted carbon	Silicon carbide converted carbon	10^{-9}	175	0.05

TABLE 1. Wear and Coeffients for Carbon Graphite
(*Mechanical Design & Systems,* Rothbart)

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