Spool Seal Design and Testing for the Spool Compressor

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ABSTRACT

Several seals, designed to minimize leakage between the rotating spool assembly's endplates and the stationary compressor housing, are introduced, constructed and tested. Some designs are deemed impractical due to either high leakage or high torque. A novel one piece hybrid design that blends the function of a face seal and piston ring is tested and shown to achieve excellent results. A test apparatus is introduced to isolate and measure the spool seal's performance independent of the compressor. It is concluded that the hybrid design is highly effective and well suited for many applications of the spool compressor.

1. INTRODUCTION

The rotating spool compressor is a novel rotary compressor mechanism most similar to the sliding vane compressor. Primary differences are described by Kemp *et al.* (2008, 2010) and include three key differences from a sliding vane compressor.

- The vane is constrained by means of an eccentric cam allowing its distal end to be held in very close proximity to the housing bore (typically less than 0.30mm) while never contacting the bore.
- The rotor has affixed endplates that rotate with the central hub and vane forming a rotating spool.
- The practical use of dynamic sealing elements to minimize leakage between the suction and compression pockets as well as between the process pockets and the compressor containment

These differences are shown in Figure 1 which presents a cutaway view of a rotating spool compressor with the key geometric features highlighted.

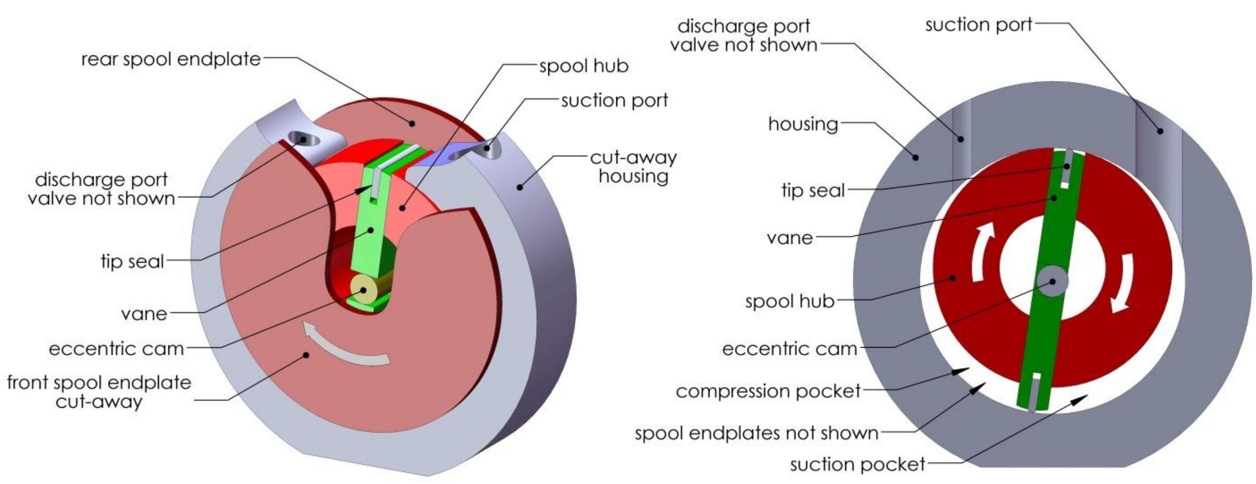

Figure 1: Rotary spool compressor schematic partial and full cutaway views.

1.1 Containment Leakage

Satisfactory performance of the spool compressor requires minimal leakage between the compressor containment and the machine's process pockets. This requires sealing the interface of the rotating spool endplates and the stationary compressor housing shown in Figure 2. Sealing may occur at the planar interface of the endplate and housing or the circumferential interface of the endplate and the housing. Depending on the compressor configuration the containment may be at suction, intermediate or discharge pressure. The containment pressure will determine the leakage flow direction. High pressure containment promotes leakage from the containment to the suction pocket. Low pressure containment promotes leakage from the compression pocket to the containment as shown in the bottom detail in Figure 2.

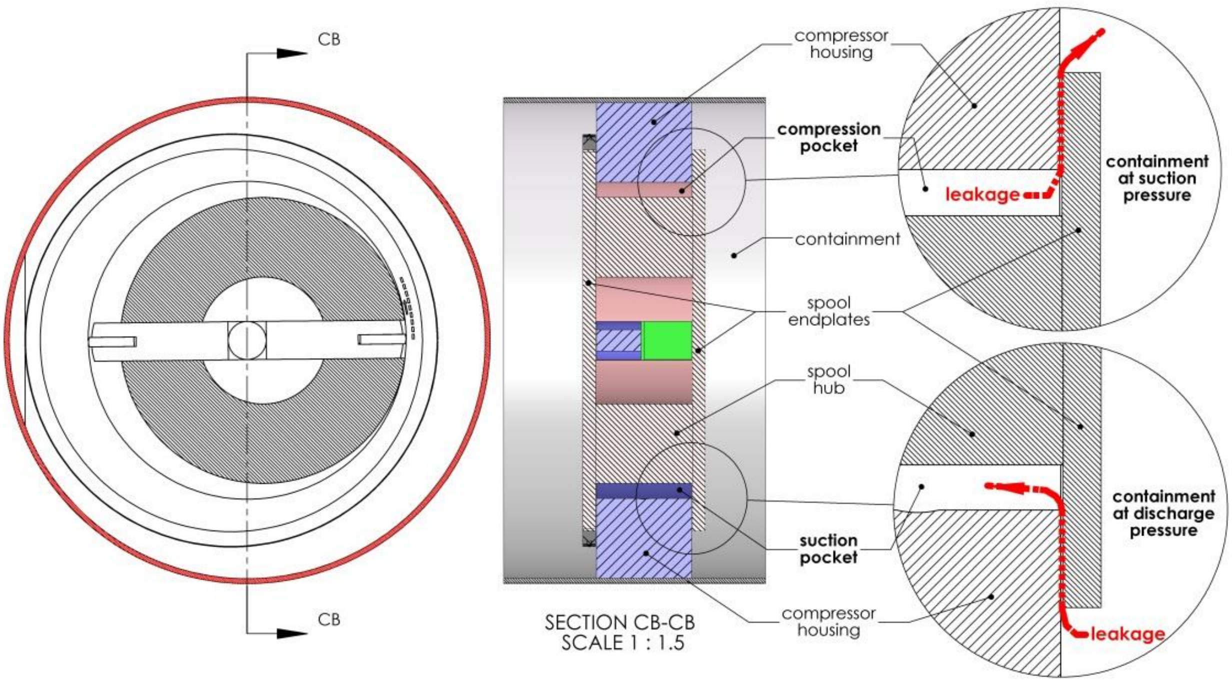

Figure 2: Containment leakage between spool endplates and housing.

1.2 Axial Pocket to Pocket Leakage

An additional performance requirement is to minimize axial pocket leakage. This is the leakage between the compression pocket and the suction pocket at the interface of housing and rotor endplate as illustrated in Figure 3. This leak is bounded radially by the tip seal as the inner radius and the outer radius being the spool seal's axial position. The spool seal is located anywhere between the interface of the spool endplate and compressor housing. This may include the endplate face or outer diameter.

1.3 Seal Land Wrap around Leakage

Depending on the spool seal implementation leakage into and out of the seal land may be a consideration. This is the case if the design allows fluid to be drawn into and out of the seal land between the sealing element and the seal land's inner diameter. Pressure differentials exist about the seal land's full circumference due to the changing displacement and location of the suction and compression pockets. On the compression side of the vane as show in Figure 4, high pressure fluid will be forced along the entire length of the interface between the high pressure pocket and the seal into the land. Any open volume below the seal in the land will act as a "fluid highway" allowing the high pressure fluid to travel to the intermediate/low pressure region. The intermediate pressure region will equalize and leak to the suction pocket. This is particularly troublesome as the entire interface of the compression and suction pocket act as collection and distribution channels, respectfully, with the land being the conduit that connects the two as show in Figure 4.

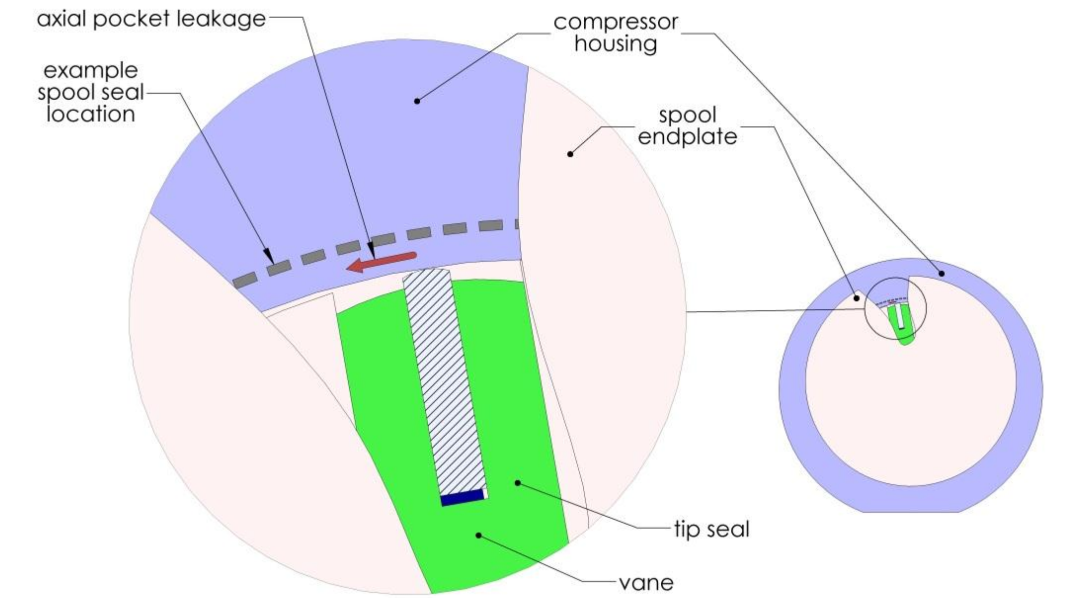

Figure 3: Axial pocket leakage.

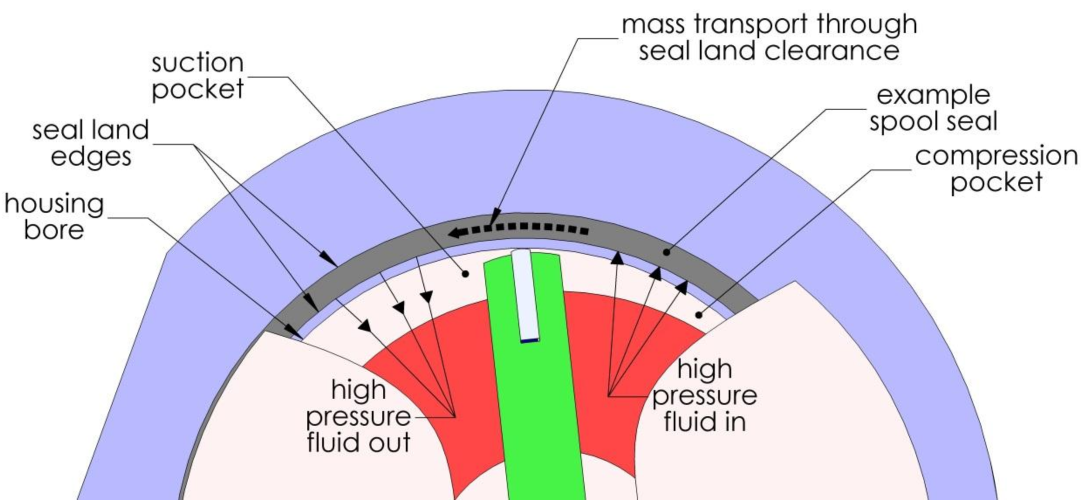

Figure 4: Seal land "wrap around" leakage.

2. DESIGN CONSIDERATIONS

2.1 Performance Factors

A major performance consideration of the spool seal is to balance sealing versus required torque. Like dynamic seals of any type, higher levels of sealing resulting in high volumetric efficiencies can normally be achieved with increasing seal face pressure. However, increasing seal pressure results in increased friction loading, increased parasitic torque and reduced energy efficiency. Ideally the spool seal would eliminate or minimize all of the leakage paths identified in the introduction. Because all of the potential sealing solutions involve relative movement between two adjacent surfaces, sliding friction and/or fluid shear losses are the dominant mechanical losses of the seal.

2.2 Dynamic Loading

The spool compressor's compression and suction pocket size and pressure profiles are dynamic with respect to rotor/vane position throughout rotation. As such, the pressure profile is variable with respect to time and different for all points about the sealing interface between the spool endplate and housing bore. This results in an important design consideration for an effective spool seal:

1. Eliminate or minimize the impact of the process pocket pressure vectors relative to a balanced seal calculation.
2. Equalize the dynamic component of the process pocket pressure vectors bidirectionally to effectively cancel its impact in a balanced seal calculation.
3. Be balanced such that increasing pressure differentials between suction and discharge pressure do not inordinately increase seal friction or vicious shear losses.

3. SEAL REVIEW

Many seals of various design have been tested in multiple prototype spool compressors (RCP2.1, RCP3.1, RCP4.2, RCP5.3) as described by, Orosz *et al.* (2012). All seals are tested with the containment running high-side (discharge pressure). Following is a description and commentary on the performance for the tested seals.

3.1 Circumferential Type Seal

The spool seal may be placed at the circumferential interface of the spool end plate and compressor housing as illustrated in Figure 5. The variety of circumferential seals tested is shown in Figure 6.

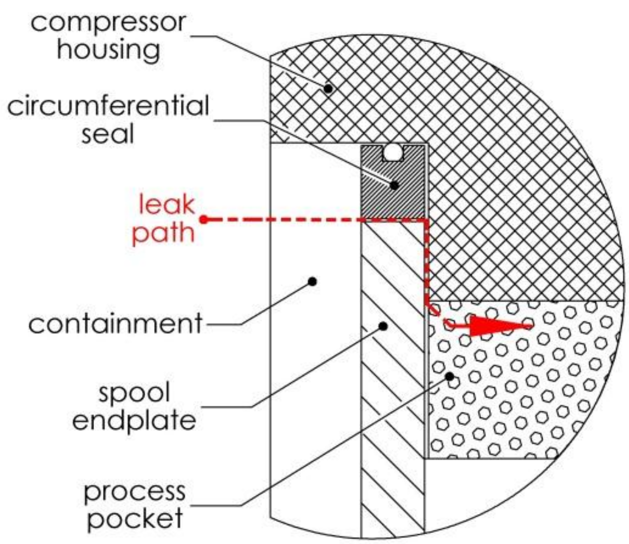

Figure 5: Circumferential spool seal location.

The Labyrinth Seal provides adequate sealing depending on the clearance between the seals ID and the endplates OD; as expected smaller clearances yielded better sealing. However, because of the large diameter and surface speed the torque from viscous shear increases substantially as the clearance is tightened. It was determined that this seal type is not a good solution for refrigerant applications. Test Results are presented in Figure 9.

The Lip Seal was tested only briefly as the torque was unacceptably high. The difficulty is primarily controlling the preload and the dynamic loading at high pressure differentials.

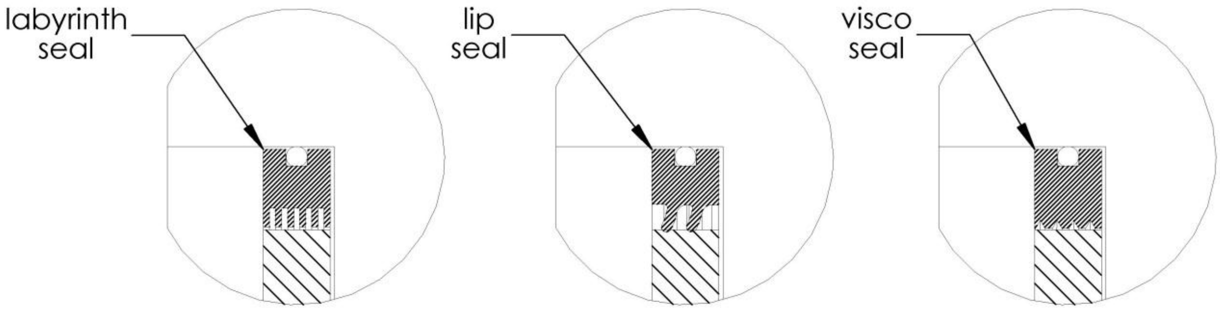

Figure 6, circumferential seal types

The Visco Seal utilizes a spiral groove to pump fluid against the leak path to prevent flow. It was determined that the axial length of the seal would need to be significantly increased to develop the required pumping pressure to prevent leakage. As such this seal was deemed not appropriate.

The circumferential seals had two key pitfalls:
1. high torque due to the large diameter of the endplates
2. these seals did nothing to control the axial pocket-to-pocket leakage or the seal land wrap around leakage

Research of the circumferential seals has been discontinued.

3.2 Face Type Seal

The spool seal may be placed between the inner plane of the spool endplate and the housing as shown in Figure 7, functioning much like a face seal. This configuration offers several advantages over the circumferential seals:

1. the seal can better control leakage as it is a direct contact seal
2. secondary axial pocket-to-pocket and seal land wrap around leakage are minimized
3. the effective diameter of the seal is smaller lowering the surface speed and torque
4. the seal balance can be better controlled to control torque and face pressure

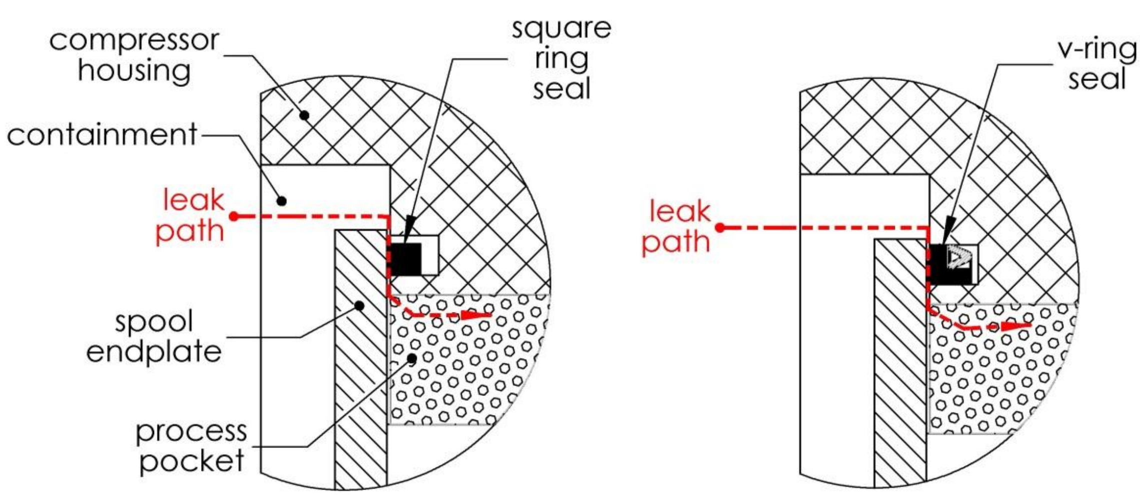

Figure 7: Face seal style.

Because the spool is eccentric to the housing bore the seal can be located concentric to the rotor, concentric to the bore, or at some location between the two as shown in Figure 8. It can be seen in Figure 8 that the seal placed concentric and close to the housing bore provides better control of the secondary leakage by minimizing the leak path volume of the axial pocket to pocket leak. However, the seal located concentric to the housing bore causes a radial wiping vector on the seal where concentric to the rotor does not. Testing has shown that a seal land located concentric to the bore as close as possible provides the best results.

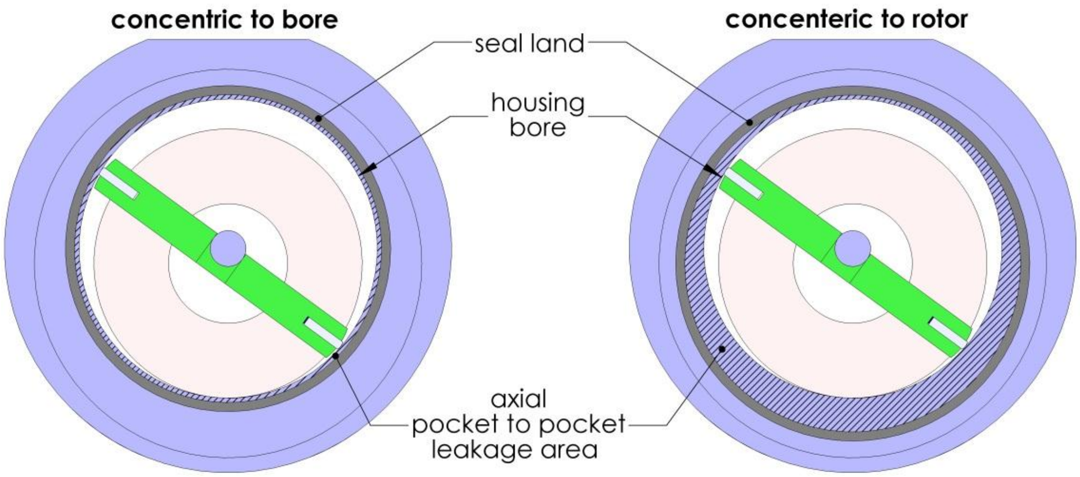

Figure 8: Seal land locations.

The Square Ring Seal must be biased (not shown in Figure 7) to ensure contact with the rotating endplate. Testing indicates that this seal can be pressure balanced to provide good sealing with satisfactory torque. However while sealing remains good across a wide range of pressure ratios the torque is punitive at high pressure ratios. Additionally the ID of the seal must be a near line fit with the housing seal land to prevent wrap around leakage. This can cause issues of hysteresis preventing seal actuation. This seal can be constructed of a PEEK type thermoplastic with a low coefficient of friction that produces significantly lower torque and greatly improved performance. However the high thermal expansion coefficient of the PEEK material relative to a ferrous housing results in an unsatisfactorily large gap between the seal and the seal land ID as temperatures increase. This in turn produces a significant wrap around leak and opens a reverse leakage from under the ring into the suction pocket resulting in unacceptable performance.

The V-ring Seal also must be biased (not shown in Figure 7) to ensure contact with the rotating endplate. This seal is constructed of a PEEK based thermoplastic and the v-ring is neoprene. It performs very similar to the square ring seal with a few notable exceptions. First, the seal expands faster than the housing due to the high thermal expansion coefficient as described with the square ring. As such the wrap around leak is still a significant problem. However, the v-ring prevents high pressure containment fluid from leaking past the seals OD, under the seal in the seal land and into the suction pocket. Second, the v-ring causes hysteresis in the action of the seal. This requires a larger preload biasing to ensure the seal stays in contact with the rotating spool endplate. Overall this seal performs well but the high preload substantially increases the torque resulting in performance that is not acceptable for many applications. Performance of the V-ring seal is presented in Figure 9.

The Hybrid Seal – *As of the writing of this paper the TORAD hybrid seal is not covered under US or international patent protection. As such no detailed information or disclosure of the seal's design or how it functions can be offered at this time. However, the seal will be described in general terms and its early performance reported.* The hybrid seal shares attributes of several seal types including a piston ring and mechanical face seal. The hybrid seal is manufactured from a PEEK based plastic alloy. The thermal expansion issue discussed earlier is not an issue with this design, thus secondary sealing leaks are negligible. The seal allows for easily tuning of the pressure balance. It is highly configurable for absolute pressure differentials and a wide speed range. These seals have been tested with excellent sealing results as indicated by the volumetric efficiency. Performance of the Hybrid seal is presented in figure 9.

3.3 Test Results

An open frame TORAD spool compressor with a displacement of 39cm³ is used for testing. The vane is 51mm wide. The compressor is tested on a hot gas bypass stand as described by Kemp *et al.* (2010) built according to the design of Hubacher and Groll (2003) with R410a as the refrigerant.

The test compressor is assembled with the appropriate spool seals, plumed into the test stand then vacuum evacuated for a period of no less than two hours after a constant minimum vacuum is reached. The compressor is run at a fixed condition controlled by modulation of hot gas and liquid suction valves and water flow rate through the condenser. The stand is allowed to reach steady state as determined by no appreciable change in temperature, pressure or mass flow at the given condition. Twenty data points are collected at .5 second intervals. The system is then adjusted to reach the next desired pressure ratio while holding the discharge at a constant pressure. Testing continues in this manner until all required data is captured.

Test data collected at each condition is averaged then used to calculate the Volumetric Efficiency as a function of pressure ratio. The volumetric efficiency was determined using Equation (1), where the theoretical volume flow was obtained based on speed measurements and the displacement volume:

$$\eta_{vol} = \frac{\dot{m}_{act} \cdot v_1}{\dot{V}_{th}} \tag{1}$$

Tests were conducted utilizing refrigerant R410a with a hot gas bypass test stand at a fixed discharge pressure of 2275 kPa at 3550 RPM. Figure 9 presents volumetric efficiency versus pressure ratio for several of the seals tested.

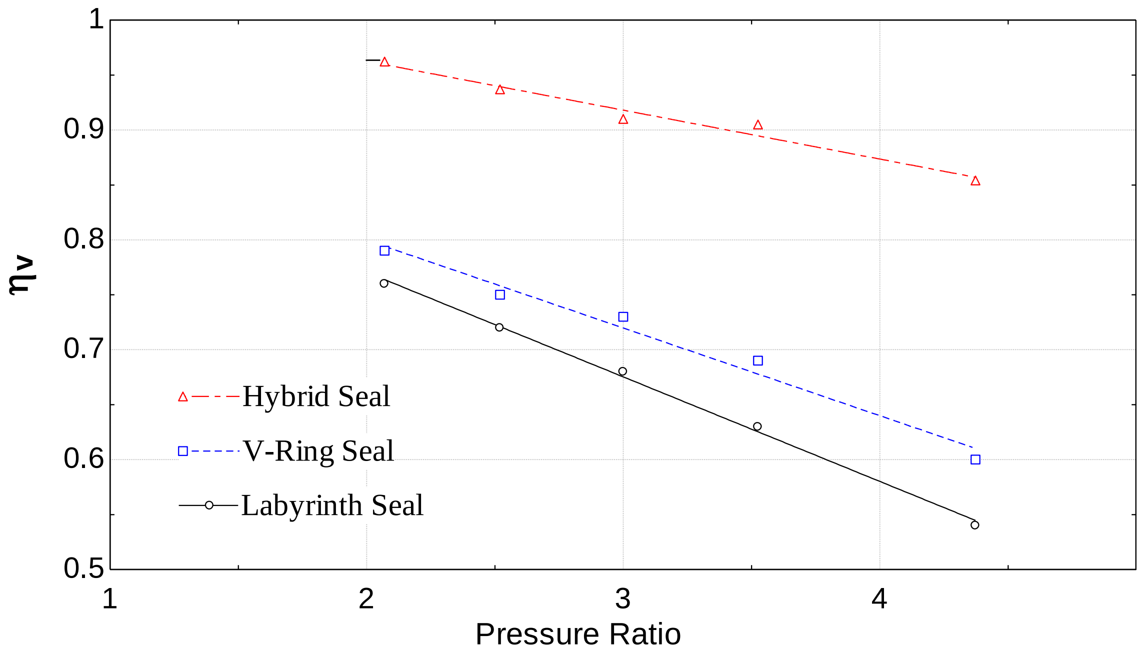

Figure 9, Volumetric Efficiency

Figure 9 illustrates the clear sealing advantage of the TORAD hybrid seal versus the v-ring and labyrinth seals. It is seen that the overall sealing is better with the hybrid seal as well as the loss per unit pressure ratio, slop of the line. It is clear that controlling the leakage as close to the housing bore, (which is the case of the hybrid seal, but not the case for the labyrinth or v-ring seal), reduces the axial pocket and wrap around leaks as described earlier. Planned improvements in the hybrid seal should result in improvements in both the peek volumetric efficiency and loss per unit pressure ratio of the hybrid seal. However, current performance is certainly adequate for many commercial refrigeration and air-conditioning applications.

4. FUTURE WORK

4.1 Seal Test Machine

Optimizing the performance of the spool seal in the compressor is difficult as its performance cannot be isolated from that of other dynamic components. To facilitate the isolated testing and development of the spool seal a test machine has been designed and constructed. Figure 10 illustrates the seal test machine configuration and internal cross section.

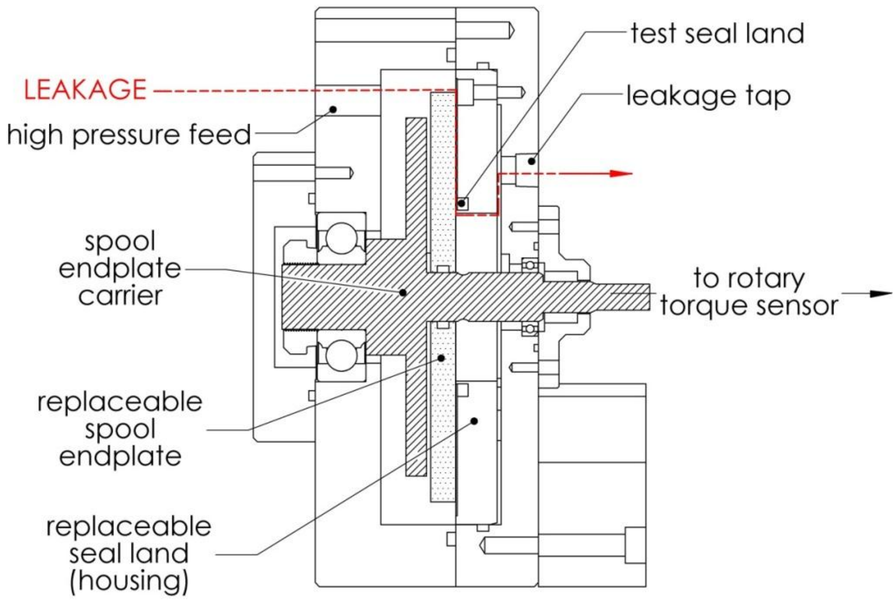

Figure 10: Spool seal test machine.

The seal test machine high pressure feed is plumbed to the discharge side of a slave compressor operating in a hot gas bypass test stand making it easy to control the discharge condition. This results in a high pressure charge in the spool seal test machine nearly identical to the containment of the actual spool compressor. However, there is no suction or compressive work being exerted on the containment fluid. The leakage tap is plumbed to the suction side of the slave compressor. However, there is an oil separator and mass flow sensors for both oil and gas between the test machine and the slave compressor.

The spool test machine simulates the interface of the spool endplate and housing at the housing bore diameter in the actual spool compressor. The primary leakage path in the seal test machine is identified in Figure 8. In order to leak through the machine, high pressure fluid in the containment must pass between the rotating spool endplate and housing, then through the simulated housing bore to reach the low pressure leakage tap as shown. This simulates the exact leakage path between the high pressure containment and the suction pocket of the actual spool compressor.

A seal land cut into the replaceable housing allows fitment of a variety of sealing elements. These sealing elements are then the only means to restrict leakage between the containment and suction pocket. Any mass flow past these seals can be measured to determine the effectiveness of the seals.

The spool seal test machine's input shaft is coupled to a rotating torque sensor which in turn is connected to a variable frequency controlled AC induction motor. The torque sensor provides both instantaneous torque and speed outputs for collection into a high speed data acquisition system where power is easily calculated. This allows the analysis of power and torque of individual seals under near actual load conditions.

In operation the slave machine is brought to the steady state test point. The spool test machine can be run at various speeds to capture both power consumption and leakage (mass flow) past the seal. Thus, various seal designs can be evaluated independently of the compressor's operation accelerating the seal development process. An additional benefit of knowing the parasitic losses of the spool seal is that those losses can be subtracted from the actual spool compressors parasitic losses in operation. This provides valuable insight into the few remaining mechanical losses internal to the mechanism.

A theoretical model of the hybrid spool seal is under development. The theoretical model will be tuned using the data from the isolated spool seal test machine. Given a tuned model understanding the seal's fundamental limitations will be clear. This will allow the performance optimization of the hybrid spool seal to be greatly accelerated.

5. CONCLUSIONS

Sealing the spool endplate housing interface was identified early in the spool compressor development cycle as one of the primary technical challenges. It has been determined that the spool compressor's containment and process pockets can be readily sealed with a variety of spool seal designs. The current best performing seal is TORAD's Hybrid spool seal which demonstrates excellent performance. The spool compressor equipped with these seals has achieved the performance levels necessary for several refrigerant compression applications. However, for the compressor to be viable over a broad range of applications where overall energy efficiency is critical the spool seals will require further development to reduce friction losses without sacrificing sufficient sealing. Detailed modeling of the boundary conditions and dynamic fluid properties about the seal's surfaces are needed to better understand and optimize TORAD's Hybrid seal's design for specific applications.

NOMENCLATURE

$\dot{m}$	mass flow	(kg/sec)	Subscripts	
v	specific volume	(m^3/kg)	*1*	state 1 (suction)
$\dot{V}$	volumetric flow	(m^3/sec)	*act*	actual
η	efficiency	(-)	*th*	theoretical
			vol	volumetric

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