2012

Spool Compressor Tip Seal Design Considerations and Testing

Greg Kemp
greg.kemp@toradengineering.com

Joe Orosz

Craig Bradshaw

Eckhard A. Groll

Spool Compressor Tip Seal Design Considerations and Testing

Greg Kemp[1*], Joe Orosz[1], Craig R. Bradshaw[2], Eckhard A. Groll[2]

[1] Torad Engineering LLC, Alpharetta, GA 30004

[2] Ray W. Herrick Laboratories, Purdue University, West Lafayette, IN 47907

* Corresponding Author

ABSTRACT

The simple analysis, design, construction and testing of the spool compressor's tip seal is presented. Analysis yields key design considerations for seal geometry, preloading and pressure balance. Several seal designs are constructed and tested for leakage. Volumetric efficiency is calculated for three tip seal configurations and presented as a function of pressure ratio. It is determined that a two piece seal that expands axially results in less leakage. Further, it is determined that fluid pressure activation of the tip seal improves performance and eliminates the need for a highly loaded biasing spring. It is concluded that the tip seal performance is acceptable for a production compressor and can be constructed such that it would provide an acceptable service life.

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.

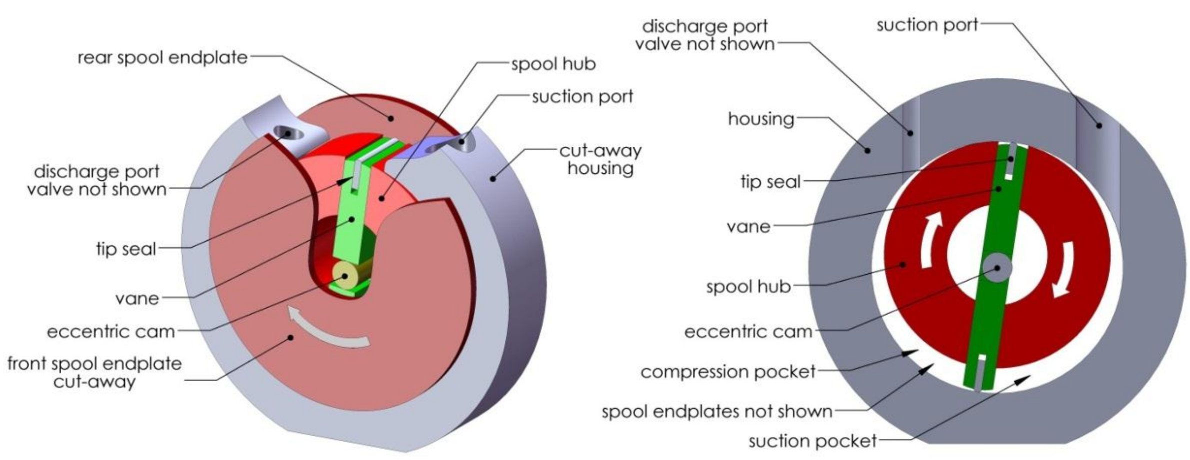

Figure 1: Rotary spool compressor schematic partial and full cutaway views.

1.1 Tip Seal Leakage

A key requirement to adequately sealing the spool compressor is closing the small end gap between the vane's distal end and the housing bore. Due to the short leakage distance between the compression pocket (P_{high}) and suction pockets (P_{low}) at the tip of the vane this is a critical sealing location. A tip seal is fitted in a channel at the distal end of the vane to prevent fluid leakage between the pockets. The tip seal must be designed to minimize leakage at its interfaces with the vane and housing bore. There are three possible leakage paths that must be considered as illustrated in Figure 2they include:

L_b = leakage up the back of the seal if the seal is not compliant with the vane's seal land
L_s = leakage around the sides of the seal if the seal is not compliant with the rotor endplates
L_t = leakage over the top of the seal if it is not compliant with the housing bore

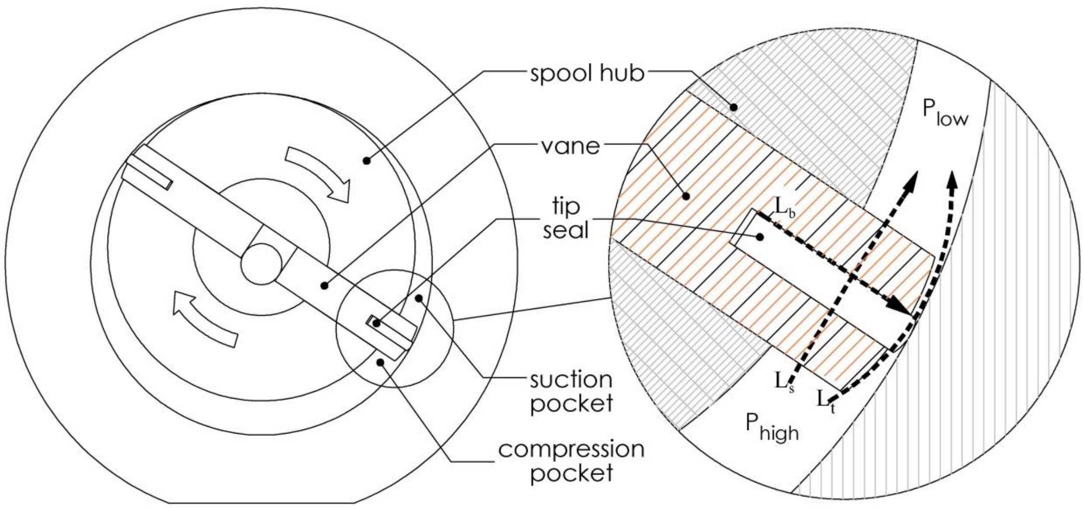

Figure 2: Tip seal leakage paths.

Back leakage (L_b) is assumed to be minimal, therefore, not considered in this work. This is due to high pressure forces on the front face of the seal combined with a large contact area and long leak path between the back of the tip seal and vane seal land.

1.2 Side Leakage

Side leakage (L_s) is a result of clearance between the sides of the tip seal and the rotor endplates. If the seal is one solid piece of material some end clearance is necessary to allow for differential thermal expansion. A line to line fit would cause excessive wear and potential binding of the seal preventing proper actuation.

1.3 Top Leakage

Top leakage (L_t) will occur if the tip seal is not in compliant contact with the housing bore. The spool compressor tip seal is similar to the apex seal used in the Wankel compressor and engine for which significant research has been conducted. Beard and Pennock (2000) describe the various challenges and considerations in the design and development of the Wankel apex seal. Most notable is the rapid acceleration required to keep the seal on the housing bore's surface. In the Wankel this acceleration profile is due to the trochidal shaped housing with which the apex seal must remain compliant. The spool compressor housing bore is round, greatly diminishing the problem of maintaining compliant contact between the tip seal and the housing bore. However, the tip seal still requires biasing to provide the net acceleration force necessary to maintain compliance with the housing bore. This is because the tip seal is carried in the rotating vane which has a slight axial movement relative to the housing bore as a result of the eccentric offset and geometric interface between the vane and eccentric cam. Figure 3 illustrates the required relative distance between the tip seal and the bottom of the vane's tip seal land to maintain the tips seal's compliance with the housing bore for the test machine in this study. Figure 3 also illustrates the calculated acceleration of the tip seal required to maintain a compliant position against the housing bore. Where the acceleration is negative the tip seal is being forced inward radially by interacting with the housing bore. Thus, no biasing force under the seal is required. Where the acceleration is positive the tip seal must be accelerated outward radially to maintain compliance with the housing bore. Positive acceleration of the tip seal is required in the second and fourth quadrants of rotation to maintain compliance with the bore. Failure to maintain compliance in the second quadrant results in leakage between the compression pocket at intermediate pressure and the suction pocket at suction pressure. This leakage results in a reduction in volumetric efficiency. Failure to maintain compliance in the fourth quadrant results in leakage between the compression pocket at discharge pressure and the trailing suction pocket that has just closed and is still very close to suction pressure. This leak results in expansion of the leakage fluid requiring recompression which consumes additional power degrading overall energy efficiency.

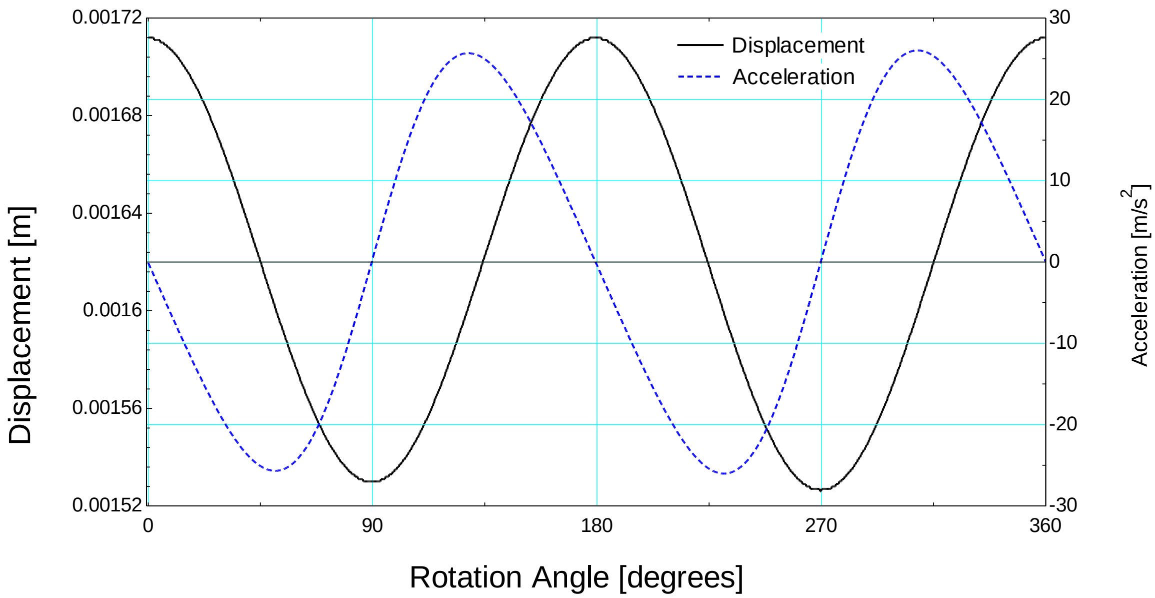

Figure 3: Tip seal and displacement vs. angle of rotation.

2. ANALYSIS

2.1 Side Leakage

It is obvious to the casual observer that side leakage (L_s) will be a minimum when the tip axial gap between the seal and the spool endplates is minimized. In an ideal compressor, the seal would be compliant with the endplates.

2.2 Top Leakage

Top leakage is complex due to dynamic fluid and pressure forces on the tip seal. As a first-cut analysis, a static "worst case" condition is analyzed. Using this simplification, a static free body diagram is used, as presented in Figure 4, to develop a seal balance to determine resultant and required biasing forces in the radial plane.

$$\text{Force}_{\text{ net radial acceleration}}\ (F_{net,accel}) = Forces_{\ bottom}\ (F_{b,p} + F_{b,c} + F_{sb}) - Forces_{\ top}\ (F_{t,p} + F_{t,sf}) \tag{1}$$

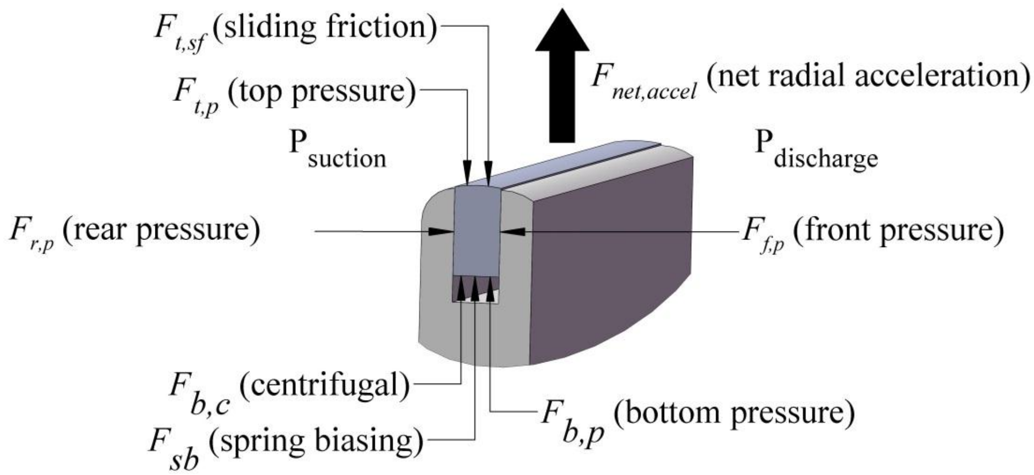

Figure 4: Forces considered in free-body analysis of spool tip seal.

By making the following assumptions, the forces in Equation (1) can be expanded:

- Only the peak acceleration requirement is considered
- The top face of the tip seal has ½ of its surface exposed to each process chamber
- The pressure gradient on the back face of the tip seal is linear with a coefficient of k_b.
- The pressure on the front face of the tip seal is full discharge pressure
- In the case of the unvented seal it is assumed that the pressure on the bottom face of the tip seal is equal to the average of $P_{suction}$ and $P_{discharge}$
- In the case of the vented seal (to be discussed) it is assumed that the pressure on the bottom face of the tip seal is $P_{discharge}$
-

This expansion is shown in Figure 5 which shows the free-body diagram of the spool tip seal as a function of spring force, pressure, area and coefficient of friction.

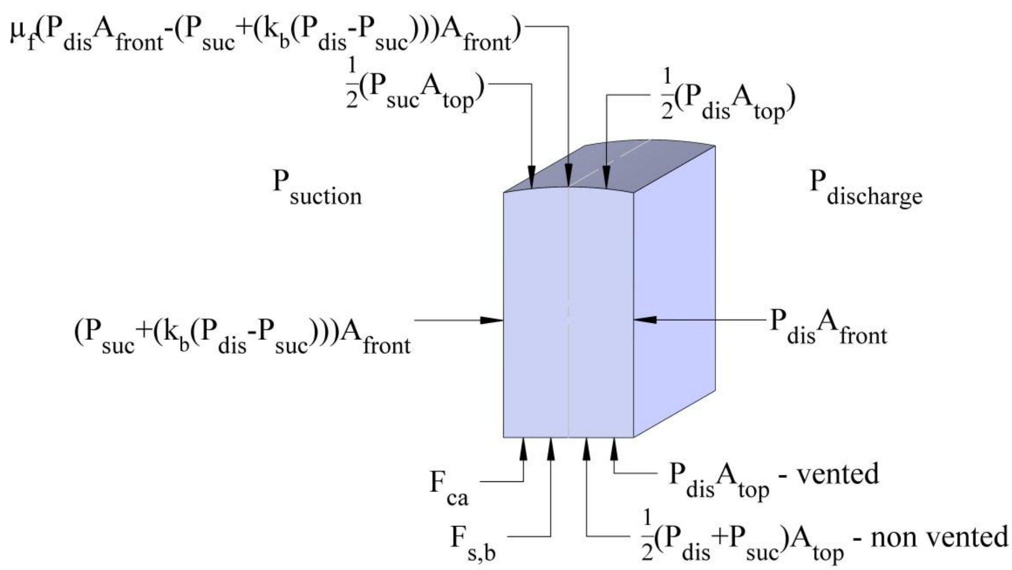

Figure 5: Static free-body diagram of tip seal forces.

Substituting the variables in Figure 5 for the forces in balancing Equation (1) the equation is solved for the spring force, $F_{s,b}$. This is the biasing force necessary to accelerate the tip seal such that it will remain in compliant contact with the housing during the portions of rotation in quadrant 2 and 4 where maximum acceleration is required. The balance equation is solved for both vented (pressure assist) and non-vented (spring only) tip seal designs as will be discussed later. The required spring biasing force for the non-vented (spring only) seal design can then be written as:

$$F_s = F_{acceleration} + \mu_f A_{front}(1 - k_b)(P_{dis} - P_{suc}) - F_{ca} \tag{2}$$

Then, the required spring biasing force for the vented (pressure assist) seal design can subsequently be written as:

$$F_s = F_{acceleration} + \frac{1}{2} A_{top}(P_{suc} - P_{dis}) + \mu_f A_{front}(1 - k_b)(P_{dis} - P_{suc}) - F_{ca} \tag{3}$$

The net radial acceleration shown in is the acceleration required to keep the tip seal in constant contact with the housing bore throughout rotation. The net radial acceleration force ($F_{acceleration}$) calculated using the seal mass and the maximum required acceleration from

$$F_{acceleration} = m * a_{max} \tag{4}$$

Finally, the centrifugal force component (F_{ca}) is calculated as,

$$F_{ca} = mr\omega^2 \tag{5}$$

3. DESIGN OBJECTIVE & CONSIDERATIONS

The design objective is to create a tip seal/spring combination that has sufficient preload and pressure loading that it will stay compliant to the housing bore to minimizing leakage. Further that it has the lowest preload possible to minimize friction which will extend service life and reduce parasitic losses.

3.1 Side Leakage

As previously mentioned a line fit between the spool endplates and the tip seal is not practical due to differential thermal expansion considerations. A simple seal with a very close tolerance fit, made from the same material as the rotor would be ideal to minimize leakage. The thermal expansion of the two materials would provide a nearly constant clearance across a broad operation range. However, this would require a ferrous seal which is high in weight relative to engineering polymers, requires expensive processing and also has a relatively high coefficient of friction and high preloading to accelerate due to the higher mass of the seal.

3.1 Top Leakage

Top leakage occurs if there is a gap between the tip seal and the housing bore. The tip seal design must account for the requirement to positively accelerate or bias the seal toward the housing bore during operation. Biasing can be achieved by multiple means. Coil springs, leaf springs, in process fluid pressure and high side containment fluid pressure are some examples. Biasing forces must be sufficient to maintain the tip seal in a compliant position but not so high as to cause excessive friction with the housing bore resulting in performance losses and accelerated wear. The biasing force must overcome the sliding friction at the back and sides of the tip seal as it translates radially in the vane’s seal land. A second consideration is the shortened life that is inevitable with a heavily loaded spring. Lastly the overall mass of the seal determines the required acceleration and thus force necessary to maintain adequate sealing performance. The ideal design solution would combine a light weight, durable, low friction material with a light biasing preload force. This design would implement proportional balanced fluid force to assist activation.

4. PROTOTYPE DESIGNS

3.1 Seal Material
A PEEK plastic alloy derivative is chosen as the material for all seal testing. This material has excellent thermal properties, is lightweight, and wears well as demonstrated by its use in many compressor applications.

3.2 Seal Geometry
Two seal geometries are considered as illustrated in Figure 6. The first is a one piece face vented design that has minimal clearance between the seal and the vane land boundaries. Side leakage is controlled by means of a close tolerance fit. The second is a two piece mitered non-vented seal that expands axially such that the seal faces are compliant with the spool endplates.

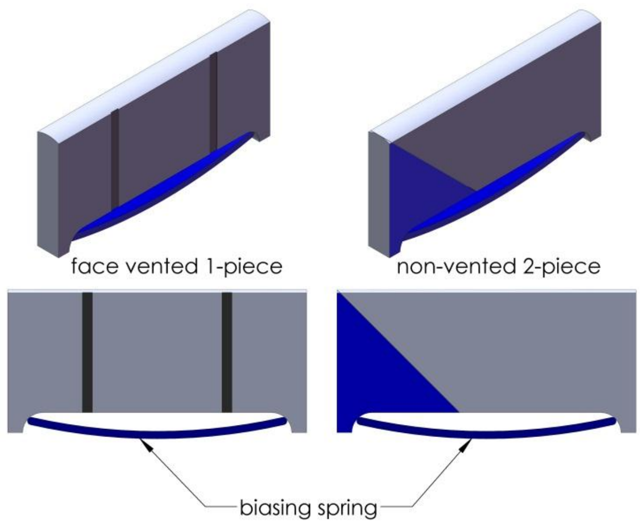

Figure 6: Test seal geometries, one-piece with a high-pressure vent (left) and non-vented two piece design (right).

3.3 Seal Biasing
The tip seal can be fit within the vane such that minimal process fluid or containment fluid reaches the vane seal land. In this instance, the seal is biased only by the biasing leaf spring as illustrated in the non-vented example in Figure 6.

If the tip seal is face vented, as shown in Figure 6, high pressure process fluid from the compression pocket reaches the underside of the seal assisting in seal biasing. However, this results in outgassing of the remainder volume below the seal in the subsequent suction cycle, diminishing the volumetric efficiency and adding heat to the suction fluid.

Alternatively the seal can be biased with high pressure fluid from the compressor containment. Fluid is vented to the vane tip seal land below the tip seal, as shown in the containment vented example in Figure 7. If the seal tip is fitted sufficiently tight leakage past the seal should be minimal with no significant impact on volumetric efficiency. Following are tip seal design point constraints and assumed constants:

Physical Properties	**Process Variables**
width = 51mm / compressor spool width	ω = 372 rad/s (3550 rpm)
height = 12mm / maximum leak path	a_{max} = 27m/s² / speed and eccentricity
r = .0216m / determined by seal height	P_{suc} = 793 kPa
μ_f = .40 / peek alloy coefficient of friction	P_{dis} = 2,275 kPa
	K_b = .65

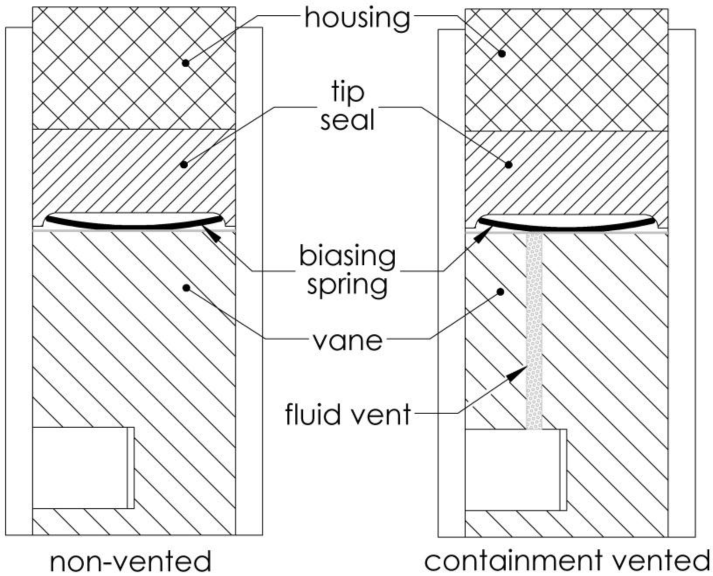

Figure 7: Pressure assisted tip seal.

Required spring force (F_s) versus seal thickness [m] is presented in , using the design parameters and Equations (2) and (3) for the vented and non-vented designs.

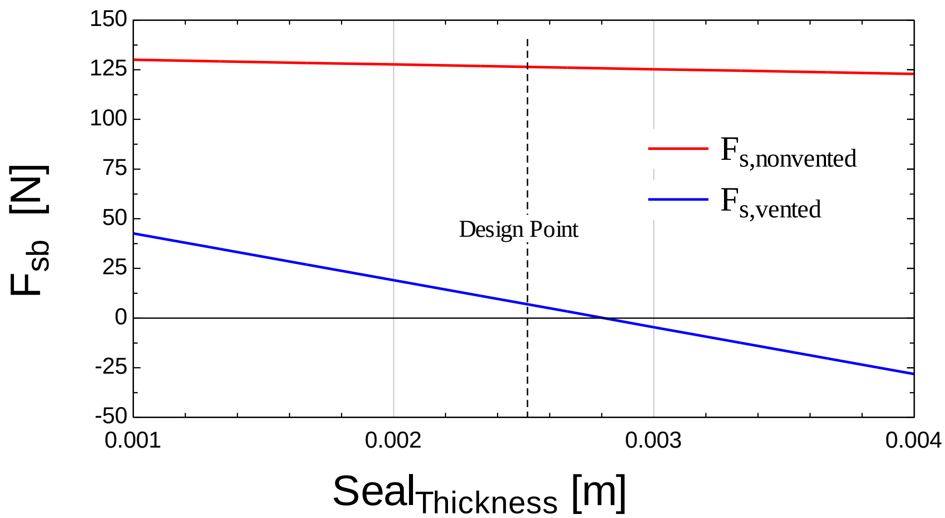

Figure 8: Biasing spring force.

For a non-vented seal, the spring force greater than 125N is not practical given the small space and required spring cycle life. Thus, an unvented seal is not tested.

An face vented seal requires only a lightly preloaded biasing spring which should provide an excellent service life. However venting the compression chamber to the bottom of the tip seal will have a negative effect on volumetric efficiency. This is due to the increased remainder volume under the seal that will re-expand into the suction pocket.

A containment vented seal also allows for a lightly preloaded biasing spring. However if the seal has too much clearance in the vane seal land containment fluid will leak into the processes pockets having a detrimental effect on the compressors performance.

Table 1 provides the base data for seals that are constructed and tested.

Table 1: Geometric and design parameters of seals tested.

Seal (ID)	1 piece	Material	ρ (kg/m³)	μ_f (-)	Thick (mm)	Mass (kg)	$F_{s,b}$ (N)
V2.1.M1.3.0	1 piece	$PEEK_1$	1310	.40	3.0	.0028	5.6
V2.2.M1.3.0	2 piece	$PEEK_1$	1310	.40	3.0	.0028	5.6

4. EXPERIMENTAL TESTING

4.1 Test Environment

An open frame TORAD spool compressor (prototype RCP5.3 as described in Orosz *et al.* (2012)) 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) and Orosz *et al.* (2012) built according to the design of Hubacher and Groll (2003) with r410a as the refrigerant.

4.4 Test Procedure

The test compressor is assembled with the appropriate tip seals, plumbed 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. The test condition was at a constant speed and fixed discharge pressure as presented in table2.

Table 2: Test parameters of the various tip seal configurations.

Test Number	Seal (ID)	Seal Type	Speed (RPM)	$P_{discharge}$ (kPa)	F_{spring} (N)	Fluid Activation Venting
1	V2.1.M1.3.0	1 piece	3550	2275	35	Face
2	V2.2.M1.3.0	2 piece	3550	2275	160	Face
3	V2.2.M1.3.0	2 piece	3550	2275	20	Containment

5. TEST DATA

5.1 Data Reduction

Test data collected at each condition is averaged and then used to calculate the Volumetric Efficiency as a function of pressure ratio. The volumetric efficiency was determined using Equation (5), where the theoretical volume flow was obtained based on speed measurements and the displacement volume:

$$\eta_{vol} = \frac{\dot{m}_{act} \cdot v_1}{\dot{V}_{th}} \tag{5}$$

5.2 Test Results

Figure 9presents the data for the three different seals tested.

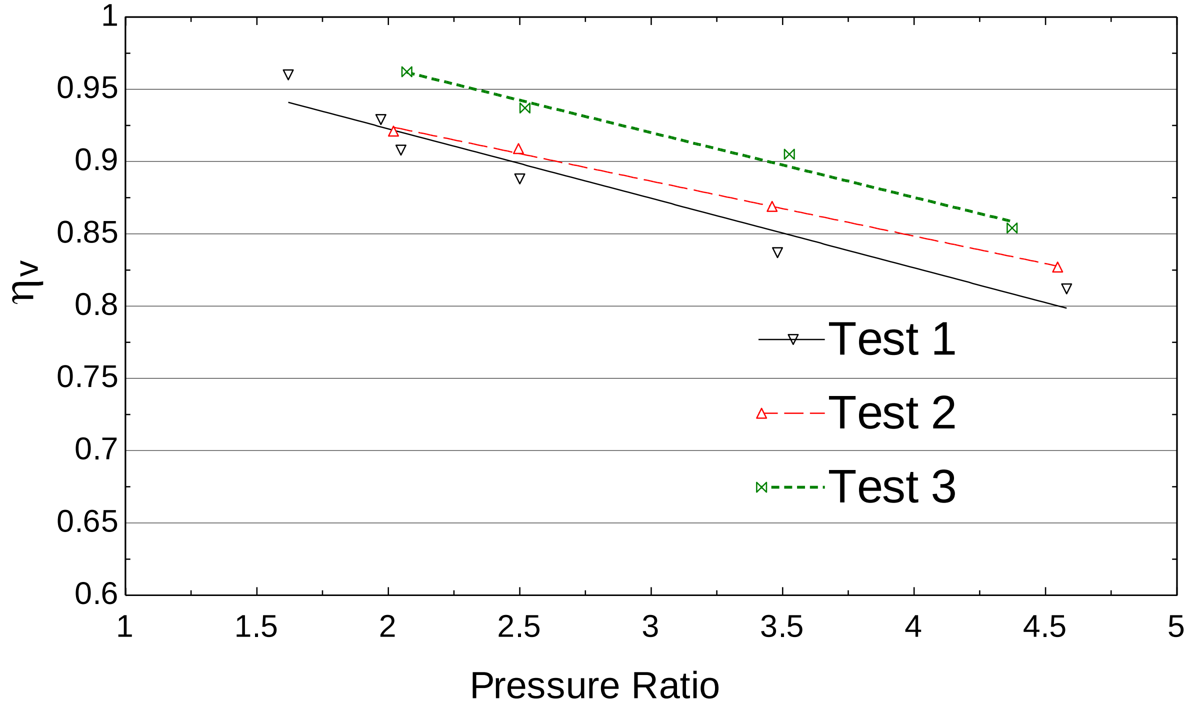

Figure 9: Volumetric efficiency as a function of pressure ratio for the three tip seal designs.

Test 1 - The seal function had a discontinuity in the volumetric performance below pressure ratios of 1.75. This was attributed to the seal being face vented with a dead volume under the seal for spring clearance. The dead volume of gas re-expanded as the vane traveled from the discharge area across the TDC and into the suction port area. Plotting the data shows the dead volume impact to be about 2.5%. The slope of the pressure loss line is 5.2% mostly due to the re-expanded gas volume. This seal configuration has a second problem. During the second quadrant of rotation compression fluid vented under the seal is not elevated much beyond suction pressure. Therefore there is insufficient force to accelerate the seal as required to maintain compliant contact with the housing bore.

Test 2 - This test configuration utilized a substantial increase in spring force from test one. It also employed use of the 2-piece mitered seal. The data shows that the discontinuity from test 1 is gone and that the slope of the pressure loss curve is improved. This is credited to improved activation of the seal and the addition of a mitered corner piece that closes off the leakage around the side of the vane. The dead volume impact is slightly less than 4%.

Test 3 - This test configuration has a lightly preloaded biasing spring combined with containment fluid vented below the seal and the 2-piece mitered tip seal. The tip seal activation was comparable with the heavy spring activation in the 2nd test. Slope of the pressure loss curve was similar or even slightly better than test 1 at 3.8% per Pr. The added benefit of this configuration is the isolation of the dead volume under the seal from the suction pocket. This prevents any re-expansion of compression gas back into the suction pocket. The isolation was improved by placing an O-ring strip between the biasing spring and the bottom of the tip seal minimizing high pressure fluid leakage from below the seal.

6. CONCLUSIONS

A rotating spool compressor is build and tested on a hot gas bypass stand. Various tip seals are designed based on a simple static analysis of the forces acting on the tips seal in the state of maximum pressure forces. The analysis shows that a 2-piece mitered seal provides better sealing by blocking side leakage. Volumetric efficiency improves when containment vented (pressure assisted) biasing of the tip seal is utilized. It is demonstrated that a lightly preloaded biasing spring combined with processes fluid activation of the tip seal provides excellent sealing. This solution should provide a path to a long service life.

It is clear that the simplified model illuminates the key design elements that must be considered for an optimal seal design. However, the static evaluation and assumptions of pressure distribution on the seal faces are not sufficient to tune the design satisfactorily. A dynamic model including fluid film and flow dynamics at the tip seal surfaces is necessary to optimize the spool compressor performance to achieve the highest energy efficiency and machine life.

NOMENCLATURE

A	area	(m²)
F	force	(N)
m	mass	(kg)
$\dot{m}$	mass flow	(kg/sec)
N	number	(-)
P	pressure	(kPa)
r	radius	(m)
v	specific volume	(m³/kg)
$\dot{V}$	volumetric flow	(m³/sec)
$\dot{W}$	power	(watts)
μ_f	friction coefficient	(-)
ω	angular velocity	(rad/sec)
η	efficiency	(-)

Subscripts

1	state 1 (suction)
act	actual
ca	centrifugal acceleration
comp	compressor
dis	discharge
front	seal front profile
is.o	overall isentropic
max	maximum value
spring	biasing spring
suc	suction
th	theoretical
top	seal top profile
vol	volumetric

REFERENCES

Hubacher, B., and Groll, E. A., “Performance Measurements of a Hermetic, Two-Stage Carbon Dioxide Compressor”, Proc*. of the 2003 Int’l Congress of Refrig.*, Washington, DC, August 18-22, 2003.

Beard, J.E., Pennock, G.R., “Acceleration of the Apex Seals in Wankel Rotary Compressor Including Manufacturing Process Variation in Location of the Seals”, 15th International Compressor Engineering Conference at Purdue University, Jul 25-28, 2000.

Kemp, G.T., Garrett, N., Groll, E. A., 2008, "Novel Rotary Spool Compressor Design and Preliminary Prototype Performance”, 19th International Compressor Engineering Conference at Purdue University, Jul 14-17, 2008.

Kemp, G.T., Elwood, L., Groll, E.A., 2010, “Evaluation of a Prototype Rotating Spool Compressor in Liquid Flooded Operation”, 20th International Compressor Engineering Conference at Purdue University, Jul 12-15, 2010.

Orosz, J., Kemp, G., Bradshaw, C., Groll, E., 2012. Performance and Operating Characteristics of a Novel Rotating Spool Compressor. In: Proceedings of the International Compressor Engineering Conference. Purdue University, West Lafayette, IN USA. No. 1257.

ACKNOWLEDGEMENT

Thank you to the partner companies and Torad investors for their commitment to the development and commercialization of the spool machine technology.