Spool Compressor Tip Seal Design Considerations

Craig R. Bradshaw
Torad Engineering LLC, Alpharetta, GA 30004 USA

ABSTRACT
The rotary spool compressor is a novel compressor type which combines various aspects of rotary and reciprocating devices. The tip seal of this compressor is a dynamic sealing element which should be well understood for maximum performance. An analysis which combines the hydrodynamic film theory balanced with the tip seal dynamics is presented. This model is explored over a variety of tip seal radii, widths, and mechanical spring rates. The experimental volumetric efficiency of a prototype compressor with various seal radii and mechanical spring rates is compared against the model predicted results and shows similar trends.

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. (1, 2) 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.

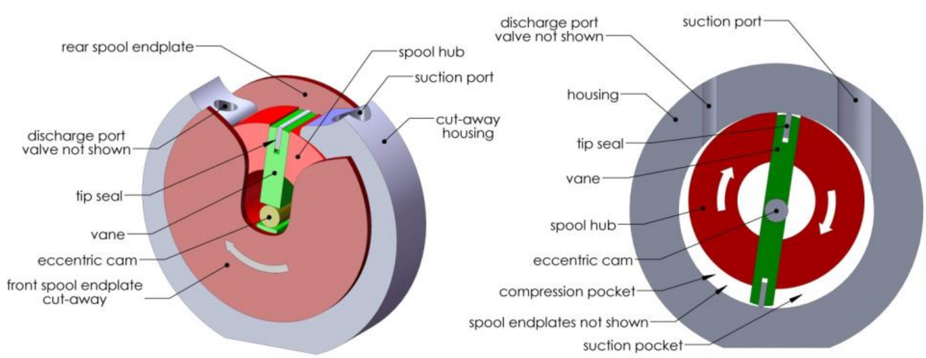

Figure 1: Cutaway view of rotating spool compressor mechanism with key components highlighted.

More demand for higher efficiency components has resulted in a renewed interest in detailed compressor modeling to predict the performance of novel compressors.

A recent approach to compressor modeling, called the comprehensive approach, has provided a complete analysis of positive displacement compressors. Recently, a comprehensive model for a spool compressor was developed and presented by Bradshaw and Groll (2,3).

However, this approach relies on sub-models for things such as leakage, heat transfer, and friction. The unique behaviour of the tip seal warrants the need for additional investigation to develop a deeper understanding of the device and improve the sub-model which represents it.

The spool compressor tip seal shares similarities to an Apex seal in a Wankel compressor (5). The kinematics of the Wankel mechanism have been studied in the past (6) as well as the kinematics and forces associated with the Apex seal (7,8). In addition, it was hypothesized that the Wankel apex seal derived part of its ability to seal by the oil in the compressor chamber that it slid on. For this reason, a typical approach to model the Wankel Apex seal included a combination of hydrodynamic lubrication theory and the dynamics of the mechanism itself (9).

This work uses the approach of a combination of hydrodynamic film theory and the dynamics and kinematics of the spool mechanism to solve for the potential leakage gap in a spool compressor tip seal. This is compared with the experimentally measured volumetric efficiency of a prototype spool compressor to obtain a reasonable understanding if the model trends predicted are realistic.

2 TIP SEAL BEHAVIOR

The tip seal in a spool compressor is a compliant sealing element at the distal end of the rotating vane. A schematic view of the tip seal mechanism is shown in Figure 2.

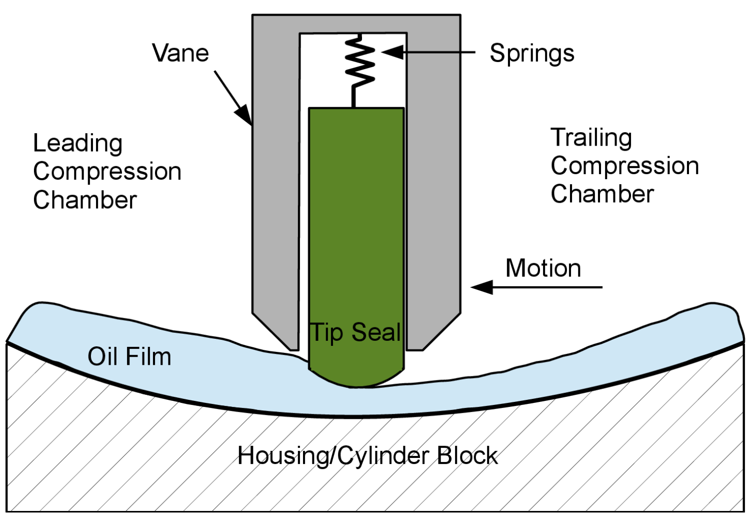

Figure 2: Schematic of Tip Seal Assembly and Proposed Behaviour in Spool Compressor.

The spool compressor vane is constrained near the axis of rotation by means of a round cam set eccentrically on the shaft. By controlling the vane using this cam it is no longer necessary for the vane and housing to come into contact as this interface is not needed to position the vane. In addition, the geometry of the eccentric cam generates a non-uniform gap between the vane and housing. This geometry is further explored in Bradshaw and Groll (2013). An example gap width between the vane distal end and the housing as a function of the rotation angle of the compressor rotor is given in Figure 3.

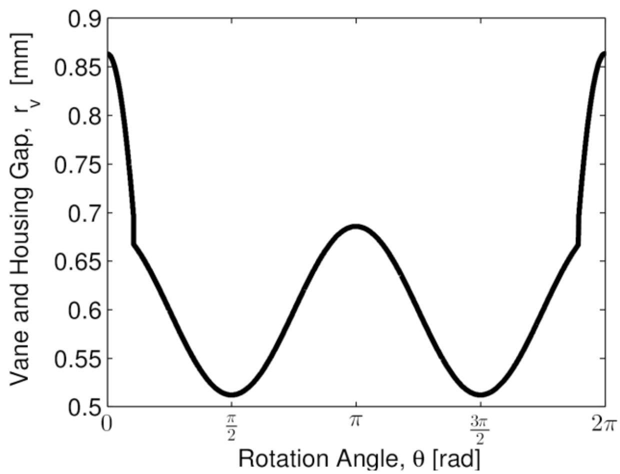

Figure 3: Profile of gap between vane and housing caused by the circular cam, generating the need for an additional compliant seal.

The example gap shown in Figure 3 is from a prototype spool compressor with a displacement of 39.3 cm^3 (2.4 in^3) and housing diameter of 5.46 cm (2.15 in) and a nominal air conditioning capacity of 17.6 kW (5 tonsR). Figure 3 shows that the sealing element must account for a fixed gap of roughly 0.53 mm as well as move dynamically to account for an additional 0.35 mm.

The compressor mechanism sits in a case filled with lubricant oil at discharge pressure. This oil will tend to leak across the spool seals, shown in Figure 1, and into the compression process chambers. Therefore, it is assumed that this constitutes enough oil to allow the tip seal to ride on a hydrodynamic film. This will generate a pressure change which will tend to push the tip seal away from the housing, much like a hydrodynamic bearing. An analysis modeling this behaviour will be described in the next section.

To counteract the hydrodynamic bearing behaviour the tip seal must be biased against the housing. Figure 2 shows that the designed mechanism to achieve this actuation is generated by a combination of force generated by mechanical springs, leading edge pressure which travels behind the seal to push the seal toward the housing and the weight of the seal which generates centripetal acceleration and accelerate the seal toward the housing. The modelling approach used to account for this is detailed in Section 4.

3 HYDRODYNAMIC ANALYSIS

Starting with the general Reynold's equation and assuming that the side-leakage (axial flow) is negligible and that the fluid properties will not change substantially when in contact with the tip the Reynold's equation reduces to:

$$\frac{dp}{dx} = 6u\mu \frac{h-h_m}{h^3} \tag{1}$$

where u is the linear speed that the tip seal travels past the housing, μ is the viscosity and h is the gap between the tip seal and housing and h_m is the gap height where the pressure gradient is zero. This expression can be integrated to obtain an expression for the pressure distribution under the tip seal:

$$P(x) = 6u\mu \int \frac{h-h_m}{h^3}\, dx + C \tag{2}$$

where C is the integration constant. The gap between the housing and tip seal, h, is a function of the linear position, x, as shown in Figure 4.

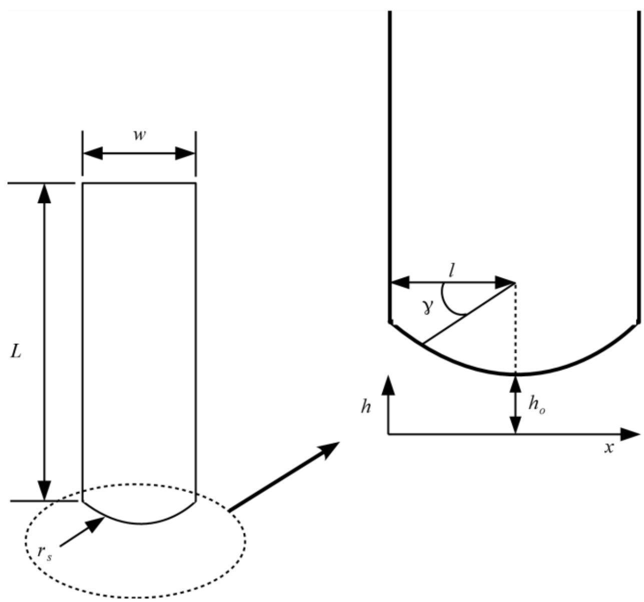

Figure 4: Dimensions of tip seal and coordinates used in hydrodynamic analysis.

The tip seal shape is circular and placed on the center of the tip seal. Therefore, the expression to describe the tip seal profile can be expressed as a function x, or parametrically:

$$x = -r_s \cos\gamma + l$$
$$h = r_s \sin\gamma + r_s + h_o \qquad \gamma_{\min} \le \gamma \le \pi/2 \tag{3}$$

where,

$$\gamma_{\min} = \cos^{-1}\left(\frac{l}{r_s}\right) \tag{4}$$

This now leaves three unknowns in Equation (1), h_m, h_o, and the integration constant C. The values of h_m and C can be solved for by applying the appropriate boundary conditions:

$$P(x=0) = P_l$$
$$P(x=L) = P_t \tag{5}$$

C can be solved directly but h_m requires iteration, which is outlined in Section 5. The minimum gap height, h_o, must also be solved iteratively. This variable is a strong function of the load applied by the combination of tip seal forces, which will be solved in the next section.

4 DYNAMIC ANALYSIS

The tip seal in a spool compressor must react dynamically during the compression process. The tip seal design shown in Figure 2 achieves this by utilizing the higher leading edge pressure, mechanical biasing springs, and the weight of the seal to load the tip seal against the housing. As detailed in the previous section this is reacted by the hydrodynamic load carrying capability of the tip seal as it rides across a film of oil. The summation of these forces is shown in the free-body diagram in Figure 5.

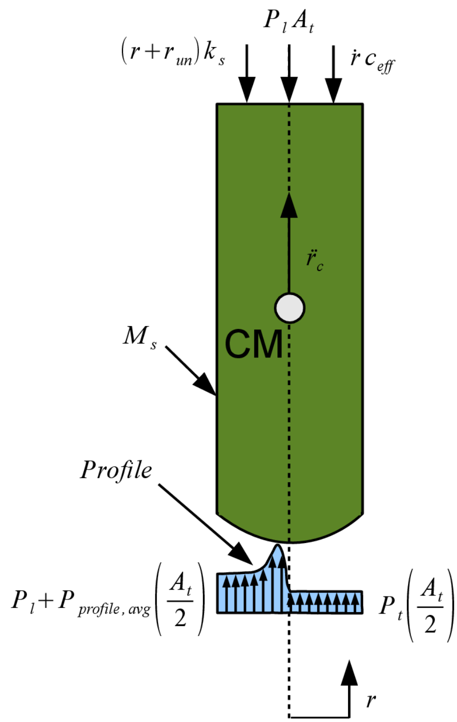

Figure 5: Free-body diagram of tip seal.

These forces are used to generate an equation of motion for the tip seal:

$$M_s(\ddot{r}+\ddot{r}_c)+c_{eff}\dot{r}+k_s r=\frac{A_t}{2}(P_{profile,avg}(h_o)+P(\theta)_t-P(\theta)_l) \qquad (6)$$

where the centripetal acceleration, $\ddot{r}_c$, is defined as $\dot{\theta}^2 R_g$. In addition, the effective damping terms are generated from the viscous shearing of the oil between the tip seal and vane as the tip seal moves inside the vane. The effective damping from the leading and trailing interface is calculated independently as the gap width between each is likely to vary greatly due to the higher pressure on the leading edge of the tip seal.

$$c_{eff}=c_l+c_t\frac{\mu L z_{stator}}{h_l}+\frac{\mu L z_{stator}}{h_t} \qquad (7)$$

where h_l and h_t are assumed constant and 12.7 and 2.5 μm, respectively, and z_{stator} is the axial depth of the compression chamber. Equation (6) is a nonlinear ODE which requires a simultaneous solution with Equation (2). In addition, Equation (6) is constrained because the tip seal cannot move beyond the housing boundary. This constraint is shown in Figure 3 which represents the maximum extension of the tip seal (i.e. when the tip seal is in contact with the bore).

5 MODEL SOLUTION

The models presented in Sections 3 and 4 are coupled by the pressure distribution in the oil film under the tip seal. This provides a challenging mathematical coupling to solve. The ultimate goal is to understand the impact of the tip seal on global compressor performance metrics (e.g. volumetric efficiency). The mathematical coupling will yield the most accurate dynamics of tip seal motion at a high computational cost. An alternative to this is to recast Equation (6) such that an accurate estimate of the net force applied to the bore can be estimated which can be used as an input to the hydrodynamic model from Section 3. This results in the following expression:

$$M_s(\ddot{r}+\ddot{r}_c)+c_{eff}\dot{r}+k_s r=\frac{A_t}{2}(+P(\theta)_t-P(\theta)_l)+F_{input} \tag{8}$$

where F_{input} is the force required to constrain the motion of the tip seal to operate within the bounds of the compressor housing. This is now an independent nonlinear ODE which can be solved numerically to obtain the net force that needs to be reacted by the oil film over the course of one rotation. An example of this force profile is shown in Figure 6.

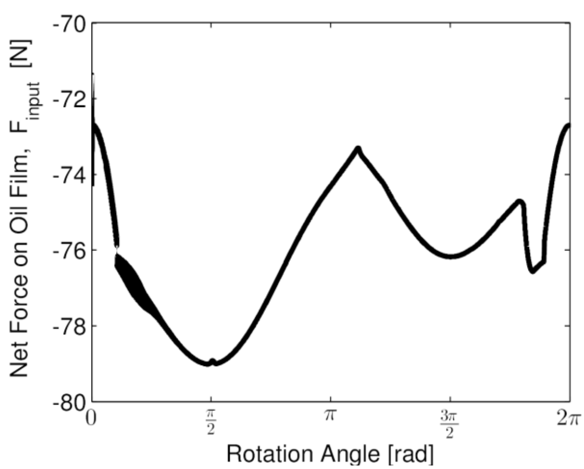

Figure 6: (Typical) Net force onto compressor housing due to tip seal from dynamic analysis.

The variation of this force is due to the combination of the changes in geometry and pressure during the compressor rotation. It is acknowledged that this will impact the instantaneous minimum oil film thickness (h_o) the goal is to obtain the average oil film thickness. Therefore, the force profiles obtained similar to Figure 6 are averaged to obtain a net input force for the hydrodynamic model.

This workflow is outlined in Figure 7 which describes the complete solution process including the iteration of the three unknown variables.

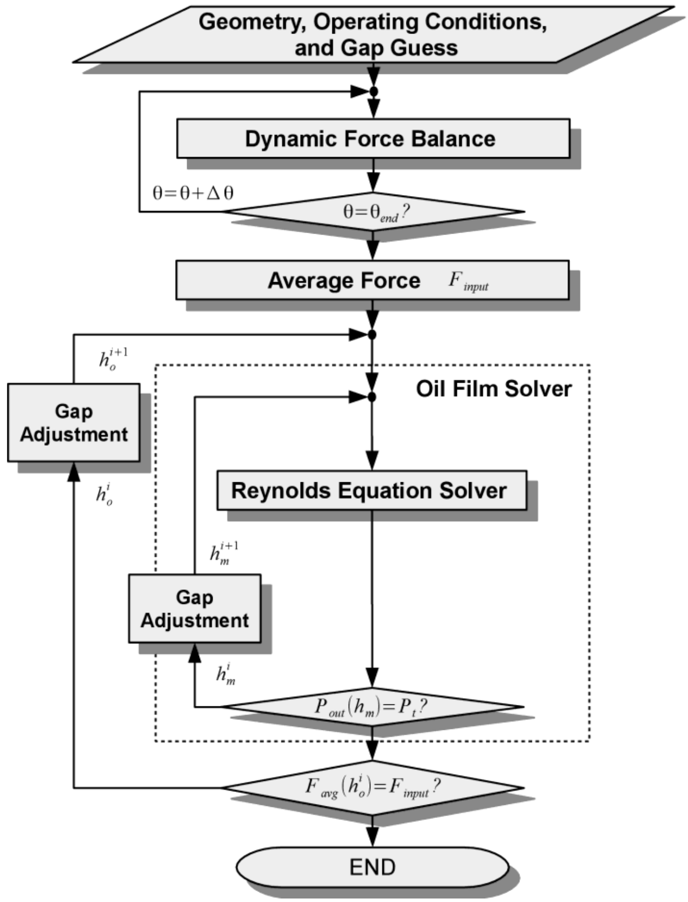

Figure 7: Solution flowchart of tip seal analysis.

Using the input force on the oil film a guess for the minimum oil film thickness is made. Then the film height that the pressure gradient is zero (h_m) is iterated on until the outlet boundary condition is satisfied. Once this loop is completed the pressure profile is integrated to obtain the average reaction force from the oil film. This is compared against F_{input}. If the values do not match h_o is adjusted and the process repeats.

6 MODEL RESULTS

This model was used to explore the impact of design parameters on the predicted minimum oil thickness. The minimum oil thickness represents a gap which leakage between chambers can occur. A larger minimum oil thickness would suggest a higher potential for leakage to occur.

Two geometric parameters were chosen to study the impact on leakage the tip seal radius and width of the tip seal. In addition, the spring rate of the mechanical spring was varied to explore the influence of the mechanical springs. The tip seal radius was varied between 1.78mm (0.070 in) and 17.8mm (0.700 in) at three widths 0.762, 1.27, and 1.78 mm (0.030, 0.050, and 0.07 in). The spring rates are studied between 20 and 80 kNm^{-1}.

Figure 8 shows the predicted minimum oil film thickness (leakage gap) as a function of tip seal radius for three tip seal widths.

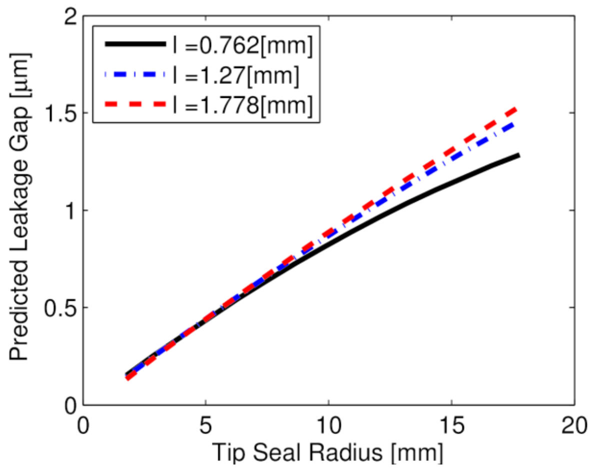

Figure 8: Predicted leakage gap (h_o) for various tip seal radii at three different seal widths.

The leakage gap increases with increasing radius regardless of the width of the tip seal. At radii less than roughly 7mm the predicted leakage gap becomes similar for all widths of seal. However, at larger radii the width of the seal begins to play a larger role. At the largest radii the smallest width tends to produce a smaller leakage gap.

To explain this behaviour it is necessary to look at the net increase in pressure under the tip seal due to the hydrodynamic effect. Figure 9 shows the pressure profile of the tip seal from the leading edge of the seal to the centreline for three seal radii.

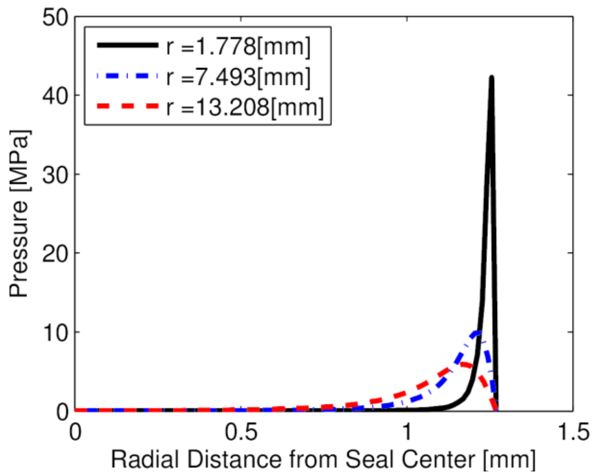

Figure 9: Pressure change from tip seal leading edge to under the middle of the seal for three different seal radii.

The profile of each seal looks similar in shape by entering at near leading edge pressure (pressure difference of zero) and building pressure as the oil film is pushed under the seal and minimizing at trailing edge pressure at the seal centerline. For larger radii the peak pressure increase is the lowest but stretches over a wider area. This effect results in a higher average force compared with the smallest radius which have a high peak pressure over a small linear distance. The higher net force of the largest radius tips result in the oil film pushing the tip seal further off the compressor housing and generating a larger potential leakage path compared with the smallest radii. This result of this study suggests that the thinnest seals with a small tip radius would likely seal the compression pockets the best.

In addition to geometric parameters the stiffness of the mechanical spring is also studied. Figure 10 shows the model predicted leakage gap (h_o) as a function of various spring stiffness's.

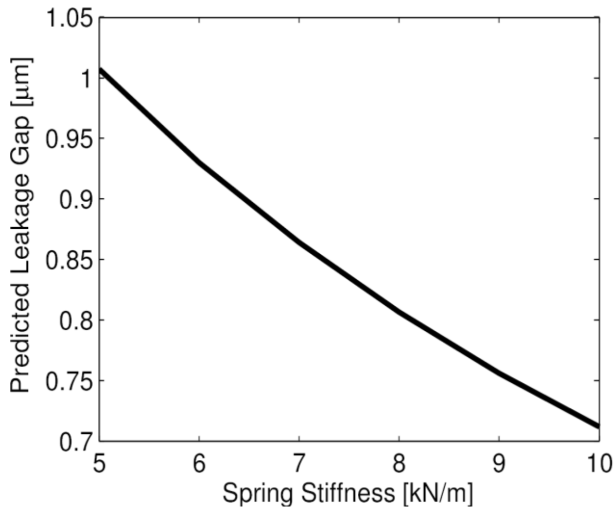

Figure 10: Model predicted leakage gap as a function of various actuating spring stiffness's.

This figure shows that the model predicted leakage gap decreases with increasing spring stiffness. This is a result of the increase in spring stiffness increasing the average force applied by the spring. This suggests that adding stiffer springs would prove useful in providing additional sealing the compressor.

7 EXPERIMENTAL RESULTS

Using a prototype compressor and experimental procedure outlined by Orosz et al. (10) an array of tip seals and spring combinations were tested. Tip seals with radii of 2.06, 7.62, and 15.2 mm (0.081, 0.3, and 0.6 in) were tested with a spring stiffness of 60 kNm^{-1}. In addition a tip seal with a tip radius of 2.06 mm (0.081 in) was tested with spring stiffness ranging from roughly 18 to 70 kNm^{-1}. Figure 11 shows the experimentally obtained volumetric efficiency at the various tip seal radii and spring rates.

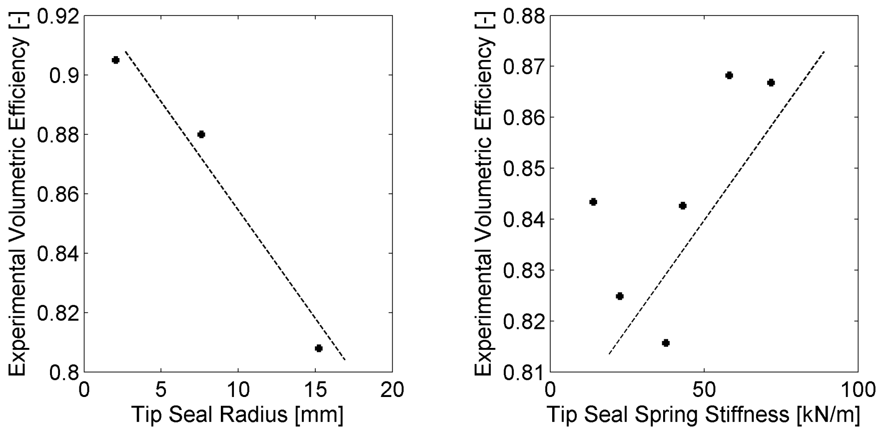

Figure 11: Experimental volumetric efficiency of prototype spool compressor with variable tip seal radius (left) and variable mechanical spring rates (right).

In the left plot in Figure 11 the trend as the tip seal radius increases the volumetric efficiency decreases. This corresponds to the prediction of the tip seal model as the predicted leakage gap (h_o) and the volumetric efficiency will generally be inversely proportional. Meaning as the leakage gap increases the volumetric efficiency will tend to go down.

In the right plot of Figure 11 the trend shows that as the spring stiffness increases the volumetric efficiency also increases. This data displays more disparity which is likely a result of increased uncertainty from inconsistent mechanical springs. However, this too correlates with the trend found in the previous study that as the spring stiffness increases the leakage gap will tend to decrease and the volumetric efficiency will tend to increase.

8 CONCLUSION

A detailed model of the spool compressor tip seal is presented. This model includes the balance of the hydrodynamic force generated by the oil film under the tip seal and the dynamic loading applied by the pressure differential surrounding the seal, mechanical springs, and the weight of the seal itself.

This model was used to explore the influence of the tip radius, tip width, and the spring rate of the mechanical springs. It was found that the potential leakage gap would tend to increase as the tip radius increases and that smaller widths would tend to decrease the sensitivity to this effect. This came as a result of an increased reaction load from the oil film. In addition, an increase in mechanical spring stiffness would tend to reduce the potential leakage gap by adding additional load for the oil film to overcome.

These trends were compared to a series of experimental studies which looked at various tip seal radii and spring rates. A direct comparison to the predicted leakage gap and experimental could not be made but the trends seen in the compressor volumetric efficiency gives an indication of the influence. It was found that the trends shown by the model are seen experimentally as the smallest radii tip seal showed the highest volumetric efficiency. In addition, the highest spring rate also showed the highest volumetric efficiency which also correlates with the findings of the study.

While this works focuses on the sealing component to tip seal design additional considerations must be taken to account for the frictional losses associated with the reduction in leakage gap. Future work will include extending this work to account for frictional losses of the tip seal.

9 ACKNOWLEGEMENTS

The author acknowledges the assistance of Greg Kemp, President of Torad Engineering, and Joe Orosz, COO of Torad Engineering in the design and testing of prototype compressor hardware.

10 REFERENCES

(1) Kemp, G., Garrett, N., Groll, E., 2008. Novel Rotary Spool Compressor Design and Preliminary Prototype Performance. In: Proceedings of the International Compressor Engineering Conference. Purdue University, West Lafayette, IN USA. No. 1328.

(2) Kemp, G., Elwood, L., Groll, E., 2010. Evaluation of a Prototype Rotating Spool Compressor in Liquid Flooded Operation. In: Proceedings of the International Compressor Engineering Conference. Purdue University, West Lafayette, IN USA. No. 1389.

(3) Bradshaw, C., Groll, E., 2013. A comprehensive model of a novel rotating spool compressor . Int. J. Refrigeration. (in review).

(4) Bradshaw, C., Orosz, J., Kemp, G., Groll, E., 2012. A comprehensive model of a novel rotating spool compressor. In: Proceedings of the International Compressor Engineering Conference. Purdue University, West Lafayette, IN USA. No. 1142.

(5) Froede, W. G., 1961. The NSU-Wankel Rotating Combustion Engine. Society of Automotive Engineers International (SAE), SAE 610017.

(6) Leemhuis, R. S., Soedel, W., 1976. Kinematics of Wankel Compressors (or Engines)by Way of Vector Loops. In: Proceedings of the International Compressor Engineering Conference. Purdue University, West Lafayette, IN USA. No. 228.

(7) Beard, J., Pennock, G., 2000. Acceleration of the Apex Seals in a Wankel Rotary Compressor Including Manufacturing Process Variation in Location of the Seals. In: Proceedings of the International Compressor Engineering Conference. Purdue University, West Lafayette, IN USA. No 1457.

(8) Pennock, G. R. & Beard, J. E. 1997. Force analysis of the apex seals in the wankel rotary compressor including the influence of fluctuations in the crankshaft speed. Mechanism and Machine Theory 32(3), 349 - 361.

(9) Iskra, A. & Babiak, M. 2009. Problems With Representation Of The Oil Film Generating Conditions On The Wankel Engine Cylinder Sliding Surface. POLISH CIMAC, 47.

(10) 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.