Influence of Volumetric Displacement and Aspect Ratio on the Performance Metrics of the Rotating Spool Compressor

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ABSTRACT

A theoretical study of the influence of geometric design and scaling is presented for the rotating spool compressor. This study uses the previously developed comprehensive compressor model for the rotating spool compressor developed by Bradshaw and Groll (2013). The compressor aspect ratio (axial length to bore diameter) is varied between roughly 0.2 and 3.5 at eccentricity ratios (rotor diameter to bore diameter) of 0.825, 0.85, 0.893, and 0.92. It is found that for a given eccentricity ratio, there exists an optimum aspect ratio that maximizes the volumetric efficiency. Additionally, the eccentricity ratio shows a high level of sensitivity to the overall performance of the compressor. As the eccentricity ratio decreases, the overall isentropic efficiency of the compressor increases until an eccentricity ratio of 0.85. Below an eccentricity ratio of 0.85 the overall isentropic efficiency does not increase despite an increase in volumetric efficiency. The scaling study modifies the displaced volume using a set of scaling rules to determine the size of the compressor features as the compressor displacement changes. The study finds that as the volumetric displacement increases, the volumetric efficiency asymptotically increases. It is also found that there is an optimum in overall isentropic efficiency as the volumetric displacement increases which suggests a trade-off between sealing and port restrictions. Using the results of these two studies a 6^{th} and 7^{th} generation prototype spool compressor is proposed which has the potential to increase the overall isentropic efficiency by 5% and 6%, respectively.

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) and include three key differences from a sliding vane compressor, as shown in Figure 1.

- 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 while never contacting the bore.
- The rotor has affixed endplates that rotate with the central hub and vane forming a rotating spool.
- The 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.

The movement of the rotor is purely rotary with only the vane and tip seals performing any oscillating movement. The eccentric cam will force the movement of the vane to oscillate by twice the eccentricity during a single rotation. The tip seals will oscillate relative to the vane two times per rotation by an amount proportional to the ratio of diameters of the rotor to the housing bore. The tip seal movement amount is roughly an order of magnitude smaller than the eccentricity and follows a sinusoidal path. Analytical details regarding the geometry, including the mathematical expressions describing the chamber volumes, is presented in Bradshaw and Groll (2013).

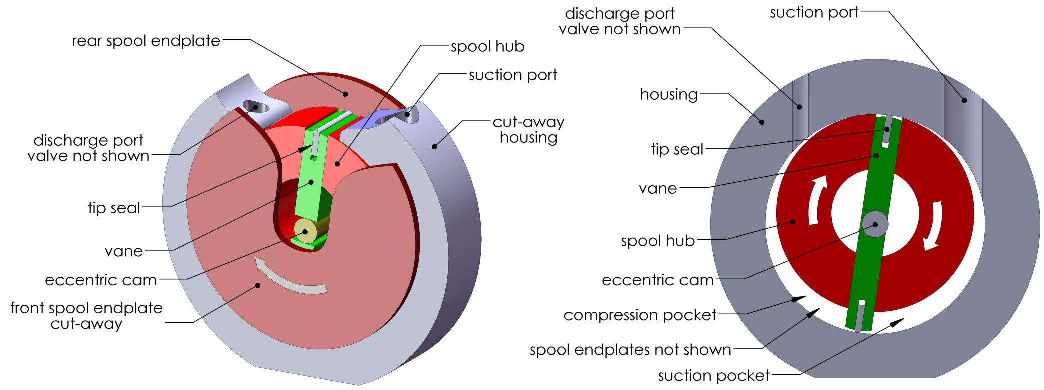


Figure 1: Cutaway view of rotating spool compressor mechanism with key components highlighted.

1.1 What differentiates the spool compressor from existing technologies?

A traditional vane machine (either sliding vane or rotary vane) are relatively difficult platforms to scale into larger capacity ranges (larger displacement). This problem stems from the interface between the vane and the housing bore and/or the interface between the vane and the end walls of machine. The spool compressor mechanism reduces these problems by constraining the vane at the center of rotation and also rotating the end plate with the vane.

By constraining the vane as close to its center of mass the sliding velocity and kinematic forces are kept at a minimum. In addition, the vane itself is restricted from sliding on the housing bore. Instead, a secondary sealing element, the tip seal, is required to take up the gap between the end of the vane and the housing bore. The tip seal size and weight can be configured to reduce the frictional losses due to the contact between the housing bore and tip seal. A study of the tip seal design parameters was presented by Bradshaw (2013).

The rotation of the end plates of the spool compressor mechanism greatly reduces the friction generated by the vane. Vane friction is not completely eliminated because the vane must slide radially relative to the end plate by an amount equal to the eccentricity. Since the end plate is rotating relative to the compressor housing the gap between these two parts requires sealing. This is accomplished with an additional dynamic sealing element called the spool seal. An overview of the spool seal design constraints has been presented by Kemp et al. (2012). The spool seal can be designed to accommodate various machine applications, such as a seal which can handle a wide operating envelope with adequate sealing. Alternatively, the seal can be designed with lower frictional power loss for applications which have high efficiency demands and a relatively small operating envelope, such as a water-cooled chiller. The rotating end plates can also be utilized for accessory functionality, such as the motive force for a vapor injection valve. Mathison et al. (2013) investigated the tradeoff in the placement and operation of a vapor injection port in the spool compressor. The study found that the spool compressor is relatively insensitive to the placement of an injection port which allows for greater design flexibility.

The combination of these dynamic sealing elements give the spool compressor the flexibility to scale to sizes that have historically proven difficult for other vane technologies. Additionally, this provides flexibility with a specific platform family that is also difficult with traditional vane machines due to the ability to modify the seal designs. In a relatively short amount of time the spool compressor has been able to demonstrate reasonably good performance, as shown by Orosz et al. (2012), compared to conventional technologies which suggest the potential is high.

1.2 Parametric Analyses of Compressors

A parametric analysis of design parameters is a useful tool to gain insight into a particular compressor technology. Many are performed on geometric parameters of conventional compressor technologies such as the studies by Yang et al. (2013),Chen et al. (2004), Rigola et al. (2005), Dagilis and Vaitkus (2009), and Bell et al. (2012). Hsieh and Wu (1996) performed an analysis of the heat transfer mechanism of a reciprocating compressor. Some others are analysis performed on novel compressor types such as those by Bradshaw et al. (2013b), Bradshaw et al. (2013a), Jovane (2007), and Kim and Groll (2007). The goal of each design is to improve the design of the compressor and/or increase the odds of a successful machine when developing a new, compressor, application, or size. This study will look at how the spool compressor performance changes with aspect ratio as well as the displaced volume of the machine.

2. ASPECT RATIO STUDIES

The basis of a spool compressor design starts with the selection of two quantities, the eccentricity ratio (ε), and the length to diameter ratio($^L/_D$). The eccentricity ratio is defined as,

$$\varepsilon = \frac{R_r}{R_s} \tag{1}$$

while the length to diameter ratio is defined as,

$$\frac{L}{D} = \frac{h_{stator}}{2R_s}. \tag{2}$$

Figure 2 shows the basic dimensions of the spool compressor and two examples of the basic spool compressor geometry which have the same displaced volume. For a fixed displaced volume, the basic shape of the spool compressor can change depending on what values of eccentricity ratio and length to diameter ratio are selected, as shown in Figure 2. To explore the impact of the eccentricity ratio and length to diameter ratio, the spool compressor model presented by Bradshaw and Groll (2013) is exercised at length to diameter ratios of roughly 0.25 - 3.75 and eccentricity ratios of 0.825, 0.85, 0.893, and 0.92. The displaced volume is held constant at 54 cm^3 (3.3 in^3).

For this test the operating conditions and fluid are fixed at an air-conditioning load point. Using R410A as the refrigerant the suction pressure is set at 905 kPa (132 psia), discharge pressure of 2282 kPa (331 psia), and 11 K (16 R) of superheat. This corresponds to 4.4 °C (40 °F) and 37.8 °C (100 °F) evaporating and condensing temperatures, respectively. The speed is held fixed at 3600 rpm, and suction and discharge porting is assumed the same for each simulated compressor.

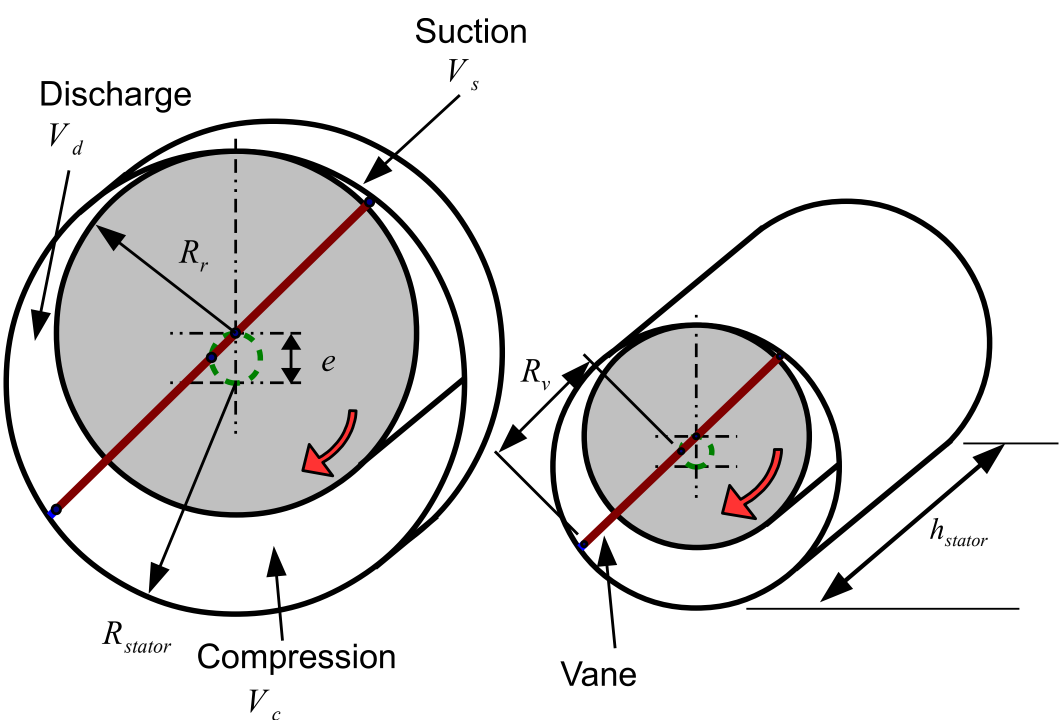


Figure 2: Spool compressor basic dimensions shown on two designs with the same displaced volume and eccentricity ratio with different length to diameter ratio.

2.1 Selected Loss Trends

Figure 3 shows the relative frictional losses of the spool seals and the vane. The relative losses are defined as:

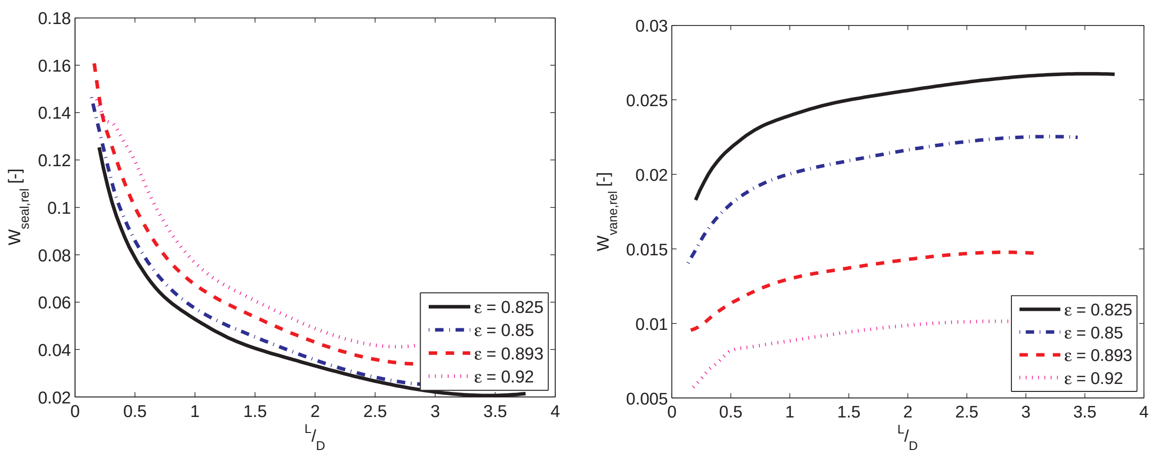
$$W_{rel} = \frac{\dot{W}_{loss}}{\dot{W}_{shaft}} \tag{3}$$

where the spool seal losses are calculated using a mixed-boundary lubrication model which accounts for both mechanical load support and hydrostatic load support, similar to Lebeck (1988). The vane losses are calculated using Newtonian friction model,

$$\dot{W}_{vane} = \mu N \omega = \mu R_v h_{stator} (P_{d,avg} - P_{s,avg})(R_s - R_r)\omega \tag{4}$$

Figure 3a shows that as the length to diameter ratio decreases the relative seal friction increases exponentially. As the length (h_{stator}) decreases the diameter of the cylinder bore must increase exponentially to maintain the same displaced volume. Since the seal friction is proportional to the cylinder bore diameter the seal friction increases exponentially as the length to diameter ratio decreases. This behavior is the same fundamental behavior captured by the non-dimensional quantity, the Zsoro number, presented by Orosz et al. (2012). As the eccentricity ratio gets larger, the cylinder bore diameter increase which also increases the relative friction. Additionally, because the spool seal leakage is also dependent on the spool seal area, the potential leakage across the spool seal follows the same trend as the friction.

Figure 3b shows the trend of the relative losses associated with the vane. When the length to diameter ratio is small, for a given eccentricity ratio, the frictional area of the vane decreases as h_{stator} decreases. Additionally, the vane width, R_v, is also coupled to the eccentricity which will increase as the length to diameter ratio decreases. The result is an exponential trend that is opposite of the spool seal trend but less severe. As the eccentricity ratio decreases these losses increases proportionally because the amount of movement of the vane increases proportionally.



(a) Relative seal loss.

(b) Relative vane loss.

Figure 3: Relative frictional losses as a function of length to diameter ratios for various eccentricity ratios.

2.2 Efficiencies

Figure 4 shows the volumetric efficiency of for various length to diameter ratios and eccentricities. At small length to diameter ratios the seal leakage area is large relative to the displaced volume so the leakage increases. Below a length to diameter ratio of roughly 0.5 the volumetric efficiency drops substantially for each eccentricity ratio. As the length to diameter increases the volumetric efficiency increases to a peak value for each eccentricity between 0.8 and 1.25 and then decreases. The decrease is a result of additional leakage of the tip seals and Top-Dead-Center leak path as the length of the design increases. As the eccentricity ratio gets smaller it is seen that the volumetric efficiency increases proportionally.

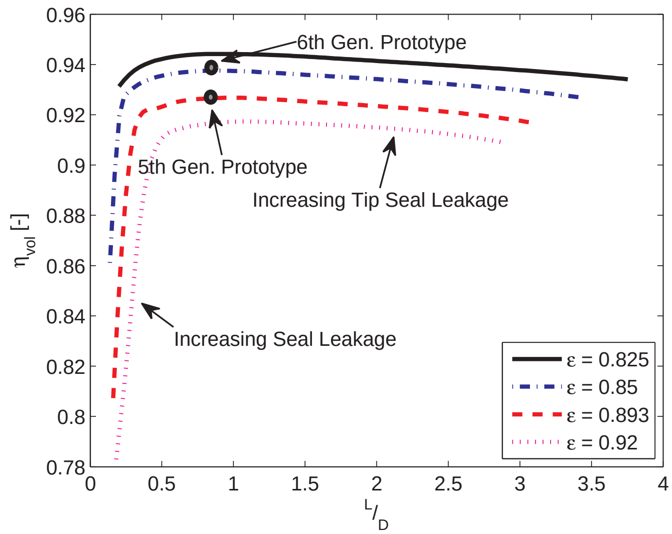


Figure 4: Volumetric efficiency as a function of length to diameter ratio for various eccentricity ratios.

The results suggest that while there is an optimum length to diameter ratio for each eccentricity ratio the current prototype generation (5th generation) is already reasonably close to this value. Therefore, the largest gains to be made would be to reduce the eccentricity ratio. As such, a eccentricity ratio of 0.85 was selected as the eccentricity ratio of the 6^{th} generation prototype spool compressor with the same length to diameter ratio as the 5^{th} generation spool compressor (0.93).

Figure 5 shows the overall isentropic efficiency as a function of length to diameter ratio for various eccentricites. Similar, to the volumetric efficiencies the overall isentropic efficiency is the lowest at the smallest length to diameter ratios. This is a result of both increased seal leakage as well as increased seal friction as a proportion of displaced volume. As the length to diameter ratio increases the efficiency tends to asymptotically increase. However, unlike the volumetric efficiency, the overall isentropic efficiency displays a peak efficiency as a eccentricity ratio decreases. As the eccentricity ratio decreases from 0.92 the overall isentropic efficiency increases for a given length to diameter ratio until an eccentricity ratio of 0.85. As the eccentricity ratio decreases beyond 0.85 no gain or even a decrease is seen in the overall isentropic efficiency. This suggests that there are is an eccentricity ratio for which there is a point of diminishing returns for the overall isentropic efficiency.

The results suggest that avoiding the small length to diameter ratios (less than 0.5) is advised for any eccentricity ratios. Additionally, while the overall isentropic efficiency increases with a reduction in eccentricity ratio, a reduction below a value of roughly 0.85 is not useful. Given the design and manufacturing constraints tend to get more difficult to overcome as the eccentricity ratio decreases, a value of 0.85 is selected for the 6th generation prototype as a good balance and effective efficiency increase.

3. SCALING STUDIES

The study of scaling examines the performance metrics as predicted by the spool compressor model as a function of displaced volume. The operating conditions of these tests are the same as used in the aspect ratio studies, from Section 2. The spool compressor model (Bradshaw and Groll (2013)) was used to predict the compressor performance at displaced volumes from 39 cc (2.4 in^3) to 991 cc (60.4 in^3) which represent roughly 13 - 345 kW (3.5 - 98 tons) of refrigeration (assuming 5.6 K (10 R) liquid subcooling) at the input conditions. The compressor speed was also held constant at 3600 rpm.

3.1 Scaling Rules

To keep the comparison fair, effort was taken to ensure that each displacement simulated did not give an undue advantage or disadvantage compared to another displacement. As shown in the previous section, the compressor performance

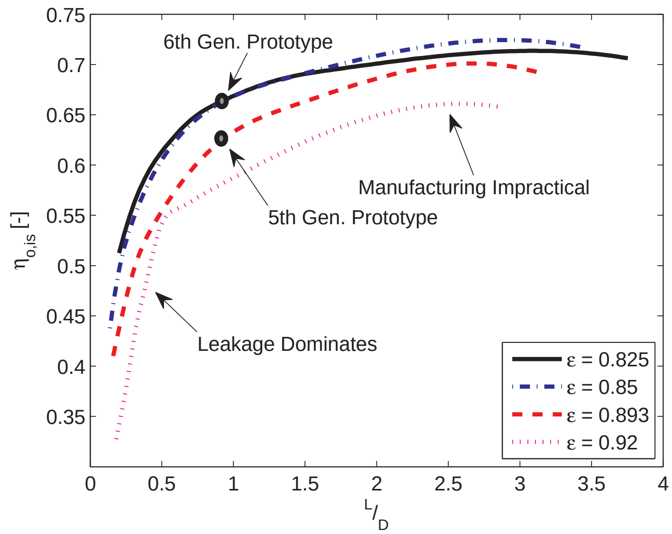


Figure 5: Overall isentropic efficiency as a function of length to diameter ratio for various eccentricity ratios.

is sensitive to changes in the eccentricity ratio, and the length to diameter ratio. Therefore, for these studies, these quantities are held constant with an eccentricity ratio of 0.893 and length to diameter ratio of 0.930. The vane width is constrained to be at least twice the eccentricity, therefore, the vane width is scaled relative to the eccentricity.

The valve dynamics of any compressor are critical for maximum efficiency. However, simulating maximum performance from the valves would require additional rigor which is not relevant to the scope of the current study. Therefore, the valve dynamics were ignored and the discharge valve was modeled as perfect. Thus, the discharge port will open fully when the working chamber achieves greater than discharge pressure and close fully when it is below discharge pressure. The discharge and suction ports were scaled linearly with the radius of the cylinder housing bore. The location of the discharge port was linearly reduced from the smallest displacement to the largest to ensure that the curtain area and backflow leakage remained proportional to the displaced volume.

3.2 Results

Figure 6 shows the results of this study in both volumetric and overall isentropic efficiencies of the compressor. The volumetric efficiency asymptotically increases with capacity. This increase is a result of the displaced volume increases faster than the leakage areas and the volumetric efficiency has nearly hit a maximum by 50 kW (15 tons). The overall isentropic efficiency increases rapidly from 13 to 50 kW (3.5 to 15 tons), picking up over 5% in efficiency points. Then near 250 kW (70 tons) the performance starts to fade as it becomes more difficult to manage heat and include an appropriate amount of discharge port area. Since all of these studies were performed at 3600 rpm, a potential solution to the port area losses would be to slow the rotational speed of the machines at the higher tonnage range.

This study has shown that there appears to be a region where the spool compressor can achieve even higher performance levels than the 6^{th} generation prototype has shown. This has prompted a proposal to develop a 7^{th} generation prototype within this 'sweet spot' for commercial air-conditioning markets.

4. CONCLUSIONS

Two studies were presented using the spool compressor model developed by Bradshaw and Groll (2013). The first study examined the influence of the basic geometry on vane and spool seal friction as well as volumetric and overall isentropic efficiencies. The study concluded that the volumetric efficiency is maximized at a particular length to diameter ratio and as the eccentricity ratio decreases the volumetric efficiency will increase. The overall isentropic efficiency increases with increasing length to diameter ratio and decreasing eccentricity ratio. However, manufacturing limits the ability to produce spool compressors with excessively large length to diameter ratios. This study lead to the development of a 6^{th} generation prototype with a length to diameter ratio of 0.93 and eccentricity ratio of 0.85 which is predicted to

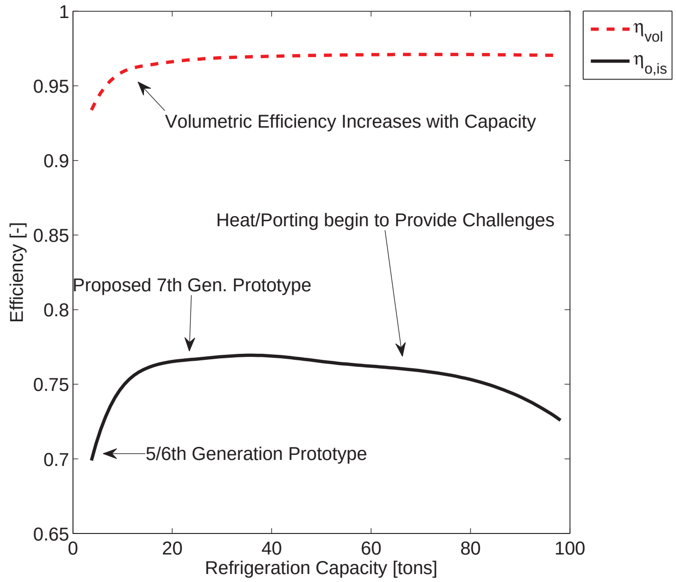


Figure 6: Predicted compressor efficiencies as a function of refrigeration capacity.

have an overall efficiency that is roughly 5% higher than the previous generation.

A second study looked at the changes in the volumetric and overall efficiency as the displaced volume of the compressor increased. As the displaced volume increased the volumetric efficiency asymptotically increases as a result of the net leakage area decreasing relative to the displaced volume. The overall isentropic efficiency follows this trend as the displaced volume increases to a peak at roughly 146 cm^3 (8.9 in^3), then begins to deteriorate as a result of additional heat losses and port flow limitations. These results facilitated the proposal of a 7^{th} generation prototype at roughly 146 cm^3 (8.9 in^3) which has an predicted efficiency improvement of 6% in overall isentropic efficiency.

NOMENCLATURE

V volume (cm^3)
R radius (m)
e eccentricity (m)
h_{stator} stator height (m)
$\frac{L}{D}$ length to diameter ratio (-)
$\dot{W}$ power (W)
P pressure (kPa)
N normal force (N)
Greek
ε eccentricity ratio (-)
ω rotational speed (rad sec^{-1})
μ dynamic friction coefficient (-)
Subscript
stator stator
r rotor
d discharge
s suction
c compression
v vane

REFERENCES

Bell, I. H., Groll, E. A., Braun, J. E., King, G. B., and Horton, W. T. (2012). Optimization of a scroll compressor for liquid flooding. *International Journal of Refrigeration*, 35(7):1901 -- 1913.

Bradshaw, C. (2013). Spool compressor tip seal design considerations. In *8th International Conference on Compressors and their Systems*, page 341.

Bradshaw, C. R. and Groll, E. A. (2013). A comprehensive model of a novel rotating spool compressor. *International Journal of Refrigeration*, 36(7):1974 -- 1981. New Developments in Compressor Technology.

Bradshaw, C. R., Groll, E. A., and Garimella, S. V. (2013a). Linear compressors for electronics cooling: Energy recovery and its benefits. *International Journal of Refrigeration*, 36(7):2007 -- 2013. New Developments in Compressor Technology.

Bradshaw, C. R., Groll, E. A., and Garimella, S. V. (2013b). Sensitivity analysis of a comprehensive model for a miniature-scale linear compressor for electronics cooling. *International Journal of Refrigeration*, 36(7):1998 -- 2006. New Developments in Compressor Technology.

Chen, Y., Groll, E., and Braun, J. (2004). Modeling of hermetic scroll compressors: Model validation and application. *HVAC & R Research*, 10(3):307--329.

Dagilis, V. and Vaitkus, L. (2009). Parametric analysis of hermetic refrigeration compressors. *Mechanika*, 6(80):23--29.

Hsieh, W. H. and Wu, T. T. (1996). Experimental investigation of heat transfer in a high-pressure reciprocating gas compressor. *Experimental Thermal and Fluid Science*, 13(1):44--54.

Jovane, M. (2007). *Modeling and analysis of a novel rotary compressor*. PhD thesis, Purdue University.

Kemp, G., Orosz, J., Bradshaw, C., and Groll, E. A. (2012). Spool seal design and testing for the spool compressor. In *International Compressor Engineering Conference at Purdue University*, number 1259.

Kemp, G. T., Garrett, N., and Groll, E. A. (2008). Novel rotary spool compressor design and preliminary prototype performance. In *International Compressor Engineering Conference at Purdue University*, number 1328.

Kim, J. and Groll, E. (2007). Feasibility study of a bowtie compressor with novel capacity modulation. *International Journal of Refrigeration*, 30(8):1427--1438.

Lebeck, A. (1988). Contacting mechanical seal design using a simplified hydrostatic model. *Tribology International*, 21(1):2 -- 14.

Mathison, M. M., Braun, J. E., and Groll, E. A. (2013). Modeling of a novel spool compressor with multiple vapor refrigerant injection ports. *International Journal of Refrigeration*, 36(7):1982--1997.

Orosz, J., Kemp, G., Bradshaw, C., and Groll, E. A. (2012). Performance and operating characteristics of a novel rotating spool compressor. In *International Compressor Engineering Conference at Purdue University*, number 1257.

Rigola, J., Perez-Segarra, C., and Oliva, A. (2005). Parametric studies on hermetic reciprocating compressors. *International Journal of Refrigeration*, 28(2):253--266.

Yang, J.-S., Mei, L., Noh, K.-Y., Moon, S.-H., Sa, B.-D., Choi, G.-M., and Kim, D.-J. (2013). A sensitivity study of size parameters in a twin-type rolling piston compressor. *International Journal of Refrigeration*, 36(3):786 -- 794.