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Optimal Spool compressor Aspect ratio for R134a and R1234z(E)

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ABSTRACT

Simulation models are often employed for parametric analysis and guide compressor prototype development, particularly for novel devices like the spool compressor. In this study, a detailed performance evaluation study is conducted to find the best aspect ratio for 105 in^3 spool compressor using refrigerants R134a and R1234ze(E). The spool compressor is a novel compressor technology that can provide competitive performance for low-pressure refrigerants (like R1234ze(E)) with a compact design. A parametric study is presented for eccentricity ratio (rotor diameter to bore diameter) of 0.75 to 0.9. For each eccentricity ratio, the compressor L/D ratio (axial length to bore diameter) varies from 0.4 to 3. Two different operating conditions are simulated to represent the standard conditions for unitary equipment application and chillers, 42 °F, 45 °F and 125 °F, 130 °F, evaporating and condensing temperatures, respectively. A spool compressor model developed by Bradshaw and Groll (2013) is extended and used to simulate the compressor performance by first extending the model validation to include 30 Rton and 40 Rton cooling capacities using R134a and R1234ze(E). An aspect ratio analysis is developed for various displacement volumes to evaluate the effect of the compressor size on the optimum design. The preliminary results suggest that the volumetric and isentropic efficiencies can be maximized by using eccentricity ratio of 0.75 and L/D values around 1.5.

1. INTRODUCTION

A spool compressor is a novel rotary compressor that has a simple design, compact form and low part count. The spool compressor is similar to a sliding vane compressor with two key differences. The vane in the spool compressor is controlled by an eccentric cam mechanism which maintains a near-zero distance between the distal end of the vane and stator, resulting in reduced friction and wear by preventing sliding contact. A second key difference is the inclusion of a rotating endplate that moves with the vane to additionally prevent sliding friction and wear. Both of these features necessitate dynamic sealing elements to minimize leakage.

Design optimization of a novel positive displacement compressor requires exhaustive performance evaluation which has traditionally been done heuristically but can also be accomplished numerically. Prototype development for a novel compressor can be very expensive and time consuming so The results from numerical models are often used to guide the prototype design and to test its performance in real conditions. The evolution of computer technology makes it easier for researchers to incorporate more physics into the models without significantly increasing the computational cost. So, many compressor models have been developed in the past few decades to substitute or complement the experimental methods. The mechanistic chamber model is a 0-D, physics-based, model and has a good combination of model fidelity and the computational speed (Tanveer & Bradshaw, 2020). Numerous researchers have used the mechanistic chamber model for the performance prediction of positive displacement compressors.

Bradshaw & Groll (2013c) presented a comprehensive mechanistic chamber model for the spool compressor. A detailed analytical geometry model was developed which includes the geometry of the vane. In the overall model, friction, leakage and heat transfer are also considered. The results were compared with the experimental data which suggests that the model can predict the compressor efficiency within 3.13 % MAE.

Bradshaw et al. (2014) evaluated the effect of volumetric displacement and aspect ratio (Axial length relative to cylinder(stator) diameter) on the performance of the spool compressor. The results suggest that the volumetric efficiency increases with the increase in the volumetric displacement. Additionally, Isentropic efficiency has an optimum value for various volumetric displacements. Based on the results presented by Orosz et al. (2012) and Bradshaw et al. (2014), a new compressor design was constructed and tested which includes optimized sub-components including, tip seals, side seals and discharge ports. Testing results suggest that the isentropic efficiency of the current prototype can exceed 80 % based on the shaft power. Following this work, Bradshaw et al. (2016) used a mechanistic chamber model to explore various displacement volumes and geometric parameters for two refrigerants (R410A and R134a). The analysis suggests that the overall isentropic efficiency increases with the displaced volume and ideal length to diameter ratio shifts. Based on the results from the parametric study, a 40 Rtons prototype spool compressor was developed for commercial air-conditioning applications.

This work is an extension of the comprehensive spool compressor model which was earlier developed and presented by Bradshaw & Groll (2013c), Bradshaw et al. (2014) and Bradshaw et al. (2016). In this article, the spool compressor model, is calibrated and validated for 30 and 40 Rton compressor sizes. The tuned model is then used to explore the optimum aspect ratio for R134a and R1234ze(E) in an 105 in^3 spool compressor. The model updates made in the current model formulation are discussed in Section 2.

2. MODEL DEVELOPMENT

The mechanistic chamber model approach is a comprehensive compressor modeling technique that can be used to predict the performance of a positive displacement compressor and has been presented in detail by Bell et al. (2020) and Ziviani et al. (2020). The same modeling approach was tailored by Bradshaw & Groll (2013c) for the spool compressor which will be used for this analysis. The approach resolves the compression process into small quasi-steady steps and tries to find the state of refrigerant inside one or more working control volumes of the compressor at each step. The state of the refrigerant at each step is fixed by defining the density and temperatures. The governing equations are derived from the mass and energy conservation equation and can be represented as,

$$\frac{d\rho}{dt} = \frac{1}{V}\left[-\rho\frac{dV}{dt} + \left(\sum \dot{m}_{in} - \sum \dot{m}_{out}\right)\right], \tag{1}$$

and

$$\frac{dT}{dt} = \frac{-1}{\rho V \frac{\partial u}{\partial T}}\left[\left(\rho V\frac{\partial u}{\partial \rho} + Vu\right)\frac{d\rho}{dt} - (p+\rho u)\frac{dV}{dt} + \sum \dot{m}_{in}h_{in} - \sum \dot{m}_{out}h_{out} + \dot{Q}_{in}\right]. \tag{2}$$

.

These equations require inputs from sub-models, including geometry, mass flow rate and heat transfer models. The details about these sub-models and modeling schemes can be found in (Bradshaw & Groll, 2013a).

Apart from the general solution structure, some model characteristics are specific to the spool compressor, for example, geometry, leakage paths, and mechanical losses. Figure 1 highlights the major leakage paths in the spool compressor. These leakage paths depend on the manufacturing tolerances and must be calibrated to tune the model. Additionally, there are some mechanical losses due to friction, most notably the friction between the vane tip and the stator and in bearings. These mechanical losses also need to be calibrated to get better estimates for compressor power. It is challenging to account for all of the mechanical and fluid losses accurately, so parasitic torque T_{para} is introduced in the model which represents all unaccounted mechanical losses.

2.1 Model calibration

The mechanistic chamber model depends on some parameters which are highly specific to compressor design and are difficult to calculate analytically. These parameters include leakage and friction coefficients. Additionally, due to the compressor design's complex nature, additional losses occur, which requires significant instrumentation and experiments to identify and quantify. These losses include, under/over-compression, flow losses in various flow paths and mechanical losses. To better agree between simulation and experimental results, the discharge coefficients for leakage paths and the mechanical parasitic losses are calibrated using experimental data and then tested for a wide range of cooling capacities and refrigerants. Each of these calibrated parameters is defined below and shown in Figure 1.

- tf_{tdc}: Tuning factor for leakage through top dead center
- tf_t: Tuning factor for leakage through tip seal

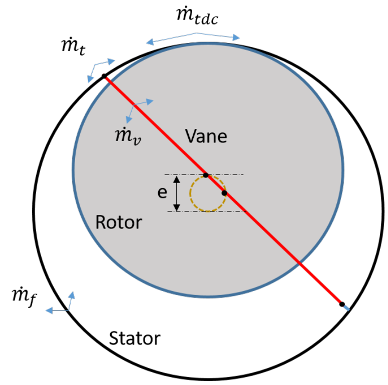


Figure 1: Various leak paths in spool compressor. Modified from (Bradshaw & Groll, 2013b).

- tf_w: Tuning factor for wrap-around gap
- tf_v: Tuning factor for leakage through vane
- tf_f: Tuning factor for leakage through face seal
- T_{para}: Parasitic torque which accounts for all unaccounted mechanical losses in the compressor

An empirical methodology is adopted to account for these losses which are discussed in this section. The existing model was calibrated for 5 Rton capacity and R410A. In the current iteration of the model, the validation is extended for two additional compressor sizes (30 Rton & 40 Rton) and refrigerants (R134a and R1234ze(E)). The calibration process starts by manually adjusting the coefficients mentioned above for a fixed cooling capacity and refrigerant, and variable evaporator temperature, condenser temperature, superheat and compressor speed. The results are compared with the experimental data and parameters are changed based on the trial and error method to achieve good agreement with the experiments. Once the simulation results match the experimental results, within $\pm 3\%$, the calibrated parameters are evaluated to find a trend with various independent variables. It is observed that the leakage coefficient and parasitic torque have no observable correlation with the compressor speed but these coefficients vary with the evaporator temperature, condenser temperature and superheat. One way to calibrate these coefficients is to develop an equation that is a function of all these independent parameters. Alternatively, the combined effect of all parameters can be accounted for as a function of a dimensionless variable, pressure ratio. So, the tuning factors are calibrated as a function of pressure ratio (PR).

$$tf_{tdc} = 0.0588PR + 0.084 \tag{3}$$
$$tf_t = 0.1096PR + 1.7245 \tag{4}$$
$$tf_w = 0.0114PR + 0.0792 \tag{5}$$
$$tf_v = 0.0012PR + 0.0108 \tag{6}$$
$$tf_f = 0.1067PR + 0.2453 \tag{7}$$
$$T_{para} = (21.349PR + 73.914)y \tag{8}$$

Once the tuning factors are calibrated for a fixed displacement volume, they are tested for variable displacements. Therefore, the first five of these tuning factors are the dimensionless numbers independent of the compressor displacement volume and work very well regardless of the compressor size. The sixth tuning factor, parasitic torque, has the dimension of torque and is dependent on the compressor displacement volume. The simulation results suggest that the parasitic torque has the same relationship with the pressure ratio regardless of the displacement volume. Still, the complete equation needs to be proportional to the displacement volume. So, a correction factor y is introduced, which is a function of displacement volume and its curve fit equation can be represented as,

$$y = 0.0186 - 0.0021 * V_{disp} + 0.0003 * V_{disp}^2, \tag{9}$$

where, V_{disp} is the maximum displacement volume of a compression chamber in in^3.

3. ASPECT RATIO ANALYSIS

There are be several design possibilities for a given compressor displacement that will describe a spool compressor and influence the efficiencies of the machine. Different aspect ratios can represent these designs. A geometric scaling model is developed that calculates geometric parameters for different displacements and aspect ratio of spool compressor which is discussed in this section. The aspect ratio of a spool compressor can be fixed by defining two variables 1) eccentricity ratio ($\epsilon = R_r/R_s$) and 2) length to diameter ratio ($L/D = h_{stator}/2R_s$).

A geometry scaling model has been developed which tries to match the desired displacement volume for a given eccentricity ratio and length-to-diameter ratio, which is a function of four variables shown below,

$$V_{disp} = fcn(R_s, e, R_r, h_{stator}), \tag{10}$$

where, R_s is stator radius , R_r is rotor radius, e is eccentricity and h_{stator} is stator length. The dimension of this correlation can be reduced from four to three by introducing the aspect ratio terms, i.e., ϵ and L/D. The final form can be represented as,

$$V_{disp} = fcn(R_s, \epsilon, L/D). \tag{11}$$

The geometry scaling model assumes a guess value for the stator radius and calculates the other geometric variables using the defined aspect ratio. These geometric inputs are then used to calculate the compressor planer area using the method described by Bradshaw & Groll (2013c) which can be converted to displacement volume by multiplying it with the stator height. The geometry scaling model iterates over the stator radius to match the calculated displacement volume with the intended displacement volume. To calculate the compressor geometry, The radius of the top dead center relief ($R_{statorrelief}$) is set equal to the rotor radius. Additionally, the half-width of the vane (R_g) and width of the tip seal slot at the distal end (R_v) is fixed for a fixed displacement case.

The aspect ratio analysis is conducted for 105 in^3 and two different refrigerants and two standard operating conditions, each representing a specific HVAC&R application. The simulated conditions are listed in Table 1. Each refrigerant is simulated for two different operating conditions (i.e. 42 °F/125°F and 45°F/130°F). Additionally, for each refrigerant and operating condition, L/D is varied from 0.4-3 for a fixed eccentricity ratio value. The process is repeated for different eccentricity ratio values ranging from 0.75 to 0.92.

Table 1: List of conditions simulated for aspect ratio analysis.

Property	**Range**
Displacement [in^3]	105
Refrigerants	R134a, R1234ze(E)
Evaporator temperature [F]	42, 45
Condenser temperature [F]	125, 130
ϵ [-]	0.75-0.92
L/D [-]	0.4-3

4. MANUFACTURING CONSIDERATIONS

The geometric designs obtained for various aspect ratios are subject to manufacturing constraints and may not be feasible in reality. These designs are analyzed for any potential manufacturing limitations that can affect the compressor performance and are accounted for in the analysis. These limitations and geometric changes due to these limitations are discussed below.

An existing spool compressor is used as a benchmark for designing and scaling the number of ports for the compressor being analyzed. The 99 in^3 spool compressor prototype, which has been tested and analyzed by Yarborough et al.

(2021), is very close to the 105 in ^{3} compressor. The available prototype has eight discharge ports in two rows. A similar port configuration is adopted for the current analysis as well. Additionally, the number of discharge ports needs to be scaled for this aspect ratio study because it is not always possible to fit eight valves for all cases. So, an algorithm was developed that calculates the maximum number of ports that can be included in a specific design along the radial and axial direction.

The spool stator is manufactured with a discontinuous profile to reduce leakage and manufacturing variability. The region of the stator around the TDC is manufactured with a radius equal to the radius of the rotor, while the rest of the stator has a different radius. At TDC, the rotor and stator come in very close contact, analogous to the contact between parallel plates instead of tangential contact. This feature improves the leakage characteristics of the spool compressor and can vary based on the manufacturing capabilities. In general, the TDC leakage gaps are sensitive to the compressor aspect ratio and it is difficult to maintain the squareness of the gap with an increasing L/D. Assuming the cylinder is bored on a high accuracy horizontal mill, the TDC leakage gap is estimated to vary based on the following expression,

$$g_{TDC} = a + bh_{stator}, \tag{12}$$

where, a is 0.0005in and b is 0.0001in and h_{stator} is in inches (Bradshaw et al., 2017). The other important leakage path is the wrap around gap in axial direction and is described in (Bradshaw & Groll, 2013c). It is also scaled using the expression described in 12 but the values of constants a and b are 0.0015 in and 0.0001 in respectively. The endplate radius is scaled as,

$$R_{endplate} = e + R_s + \phi, \tag{13}$$

where, ϕ represents additional padding to account for manufacturing constraints and a value of 0.05 in is used for scaling the endplate radius $R_{endplate}$.

The inner diameter of the seal is scaled for different aspect ratios using the expression,

$$D_f = 2(R_s + w_w), \tag{14}$$

where, D_f is the inner diameter of the seal and w_w is the radial width of the wrap-around gap.

Finally, the outer diameter of the seal is calculated as,

$$D_{out,f} = D_f + 2W_{seal,face}, \tag{15}$$

where, $D_{out,f}$ is the outer diameter of the seal and $W_{seal,face}$ is the width of the seal.

5. RESULTS

5.1 Model validation

The calibrated model is used to simulate the performance of the spool compressor for a range of displacements and cooling capacities as listed in Table 2. The simulated refrigerants include R134a, R410A and R1234ze(E). The simulation results are then compared with the experimental results which were either provided by TORAD Engineering or were collected by Yarborough et al. (2021). Figure 2 shows the comparison of the simulation and experimental results in terms of parity plots for four different macro-performance parameters which include mass flow rate, compressor power, volumetric efficiency and overall isentropic efficiency. The results suggest that the model can predict the mass flow rate, power and volumetric efficiency within 2.5% MAE and isentropic efficiency within 4% MAE. The individual mean absolute errors for each of the performance parameter is listed in Table 3. Figure 3 compares the predicted discharge temperature with the experimental results and the mean absolute difference for the discharge temperature is 3.46 F. These errors are within the instrumentation uncertainty of the experiments suggesting that the model can be used to predict the spool compressor performance.

Table 2: Range of various compressor attributes for which model is validated.

Property	Range
Displacement [in^3]	3.4-132
Cooling capcity [Rton]	3.5-40
Refrigerants	R410A, R134a, R1234ze(E)

Table 3: Mean Absolute Error (MAE) and Mean Absolute Difference (MAD) for spool compressor performance parameters

	Mass flow rate	Power	Volumetric efficiency	Isentropic efficiency	Discharge temperature
MAPE/MAD	2.01%	2.40%	2.63%	3.68%	3.46F

5.2 Aspect ratio analysis

The calibrated model is used to conduct a parametric performance analysis of 105 in^3 spool compressor for two different refrigerants, R134a and R1234ze(E), and two different operating conditions. The results are presented in Figure 4 using contour plots which highlight the best aspect ratio for different refrigerants and operating conditions. The normalized isentropic efficiency is plotted as a function of eccentricity ratio and L/D. Normalized isentropic efficiency is the ratio efficiency for any specific point divided by the peak efficiency in any particular case. A value of 1 represents the peak efficiency for any case. The results indicate that the aspect ratio can affect the isentropic efficiency by 18%. These trends can be primarily attributed to the combined effect of leakage and mechanical losses, which are shown in Figure 5 and 6. The volumetric efficiency is maximum at lower L/D and drops significantly with increased stator length due to increased leakage area. The seal loss is the major contributor to the overall mechanical losses, which peaks at lower L/D and higher eccentricity values resulting in significantly lower isentropic efficiency in this region. The mechanical loss for the vane is maximum at L/D of 3 and eccentricity of 0.75.

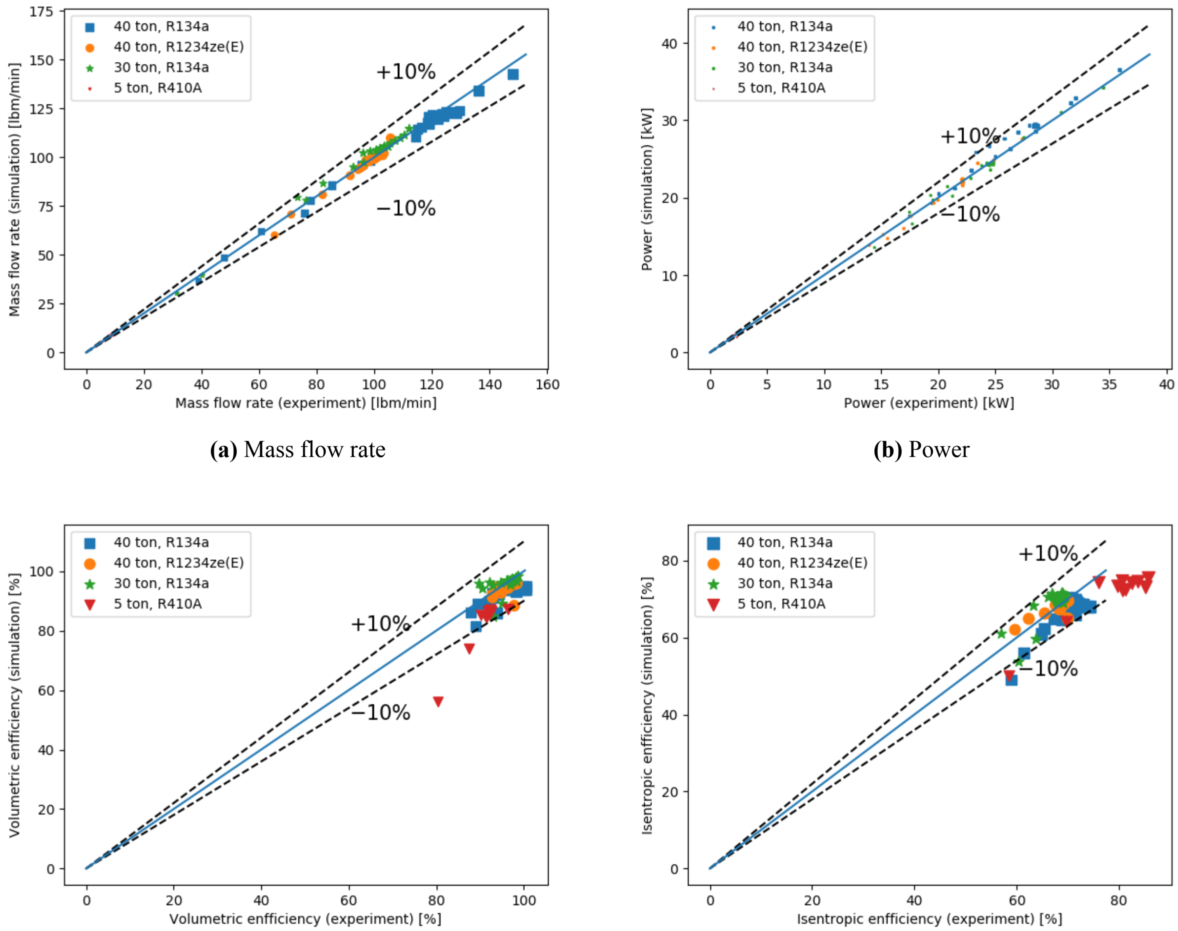
Additionally, The best isentropic efficiency is achieved at lower eccentricity values and L/D values around 1.5. The isentropic efficiency is expected to be better if the eccentricity value is reduced further, but manufacturing a spool compressor with e<0.75 is not feasible, so a lower limit of 0.75 was selected for this analysis.

The results also suggest that the optimum aspect ratio recommendation does not change significantly for different operating conditions because the pressure ratio for these conditions is not too different from each other.

Finally, an efficiency drop of 2% is observed for R1234ze(E) compared to R134a. It is due to the difference in the pressure ratio between the refrigerants and the fact that parasitic loss is a function of the pressure ratio. The pressure ratio for R134a and R1234ze(E) at 42° F/125 ° F is 3.85 and 3.93 respectively. Due to this slight difference, the parasitic loss for R1234ze(E) is higher than R134a, resulting in lower isentropic efficiency. Additionally, the peak efficiency region for R1234ze(E) is stretched slightly towards the higher L/D as compared to R134a.

6. CONCLUSIONS AND FUTURE WORK

The mechanistic chamber model's leakage coefficients and mechanical losses are calibrated for 30 and 40 Rton spool compressors. The model is validated for a range of compressor sizes and refrigerants. The results indicate that the model can predict the compressor performance within 4% of mean absolute percentage error, which is acceptable for aspect ratio analysis. . It can be concluded that the calibrated model can be used for the performance evaluation of different compressor designs. The tuned model is used to perform a parametric analysis for 105 ^{3} spool compressor to find out the best aspect ratio for R134a and R1234ze(E). The results suggest that the best efficiency can be achieved at L/D values around 1.5 and an eccentricity ratio below 0.8. Two different operating conditions were simulated as well and it was found that the compressor performance does not change significantly for the selected conditions.



(a) Mass flow rate

(b) Power

(c) Volumetric efficiency

(d) Isentropic efficiency

Figure 2: Model validation using various refrigerants and displacement volumes. Marker size is proportional to the experimental uncertainty.

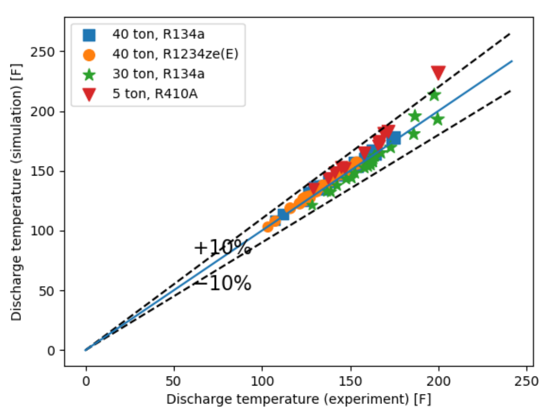
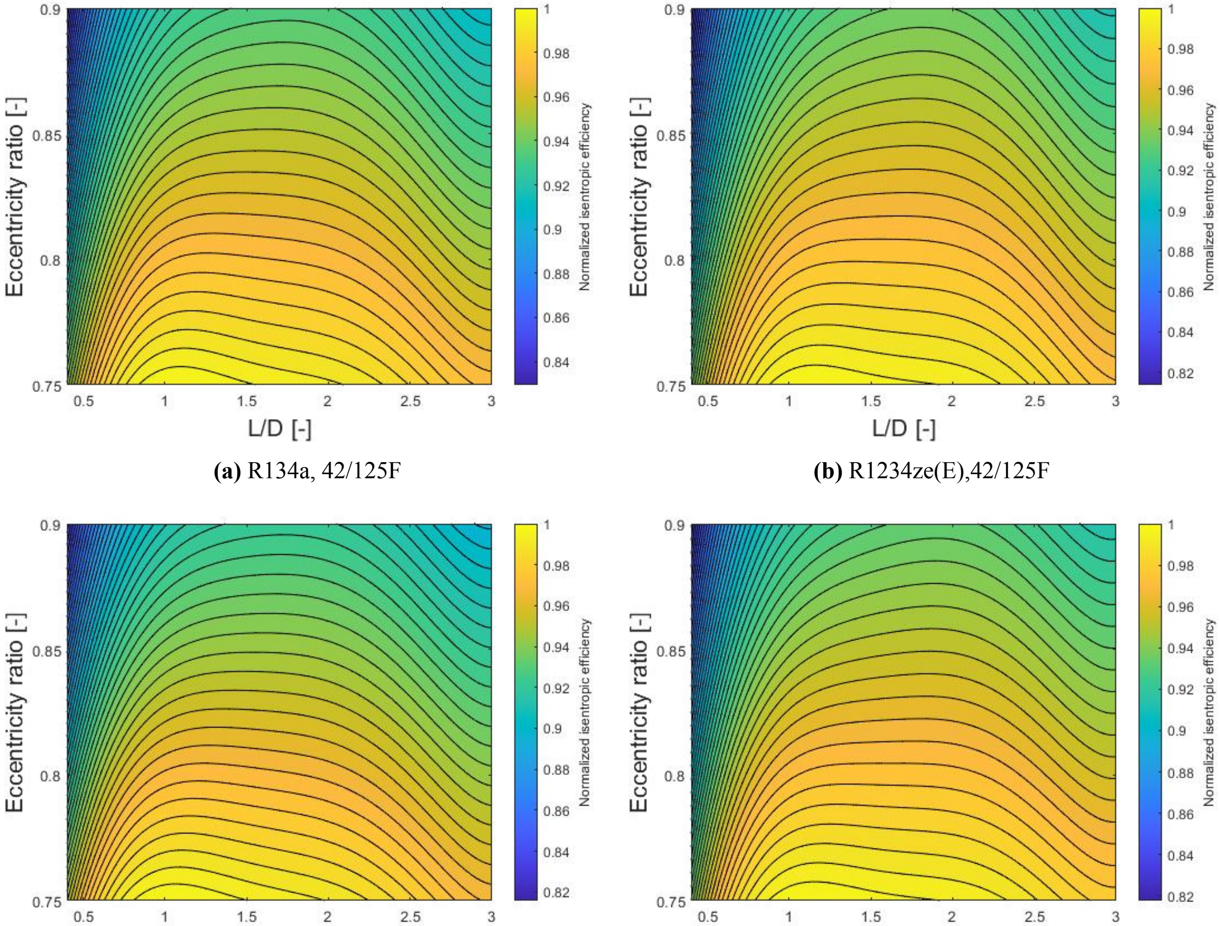


Figure 3: Discharge temperature for various refrigerants and displacement volumes. Marker size is proportional to the experimental uncertainty.



(a) R134a, 42/125F

(b) R1234ze(E),42/125F

(c) R134a, 45/130F

(d) R1234ze(E), 45/130F

Figure 4: Isentropic efficiency for different refrigerants and operating conditions.

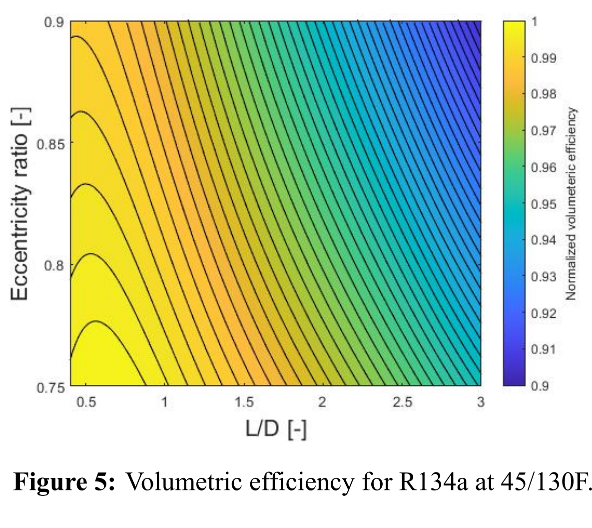
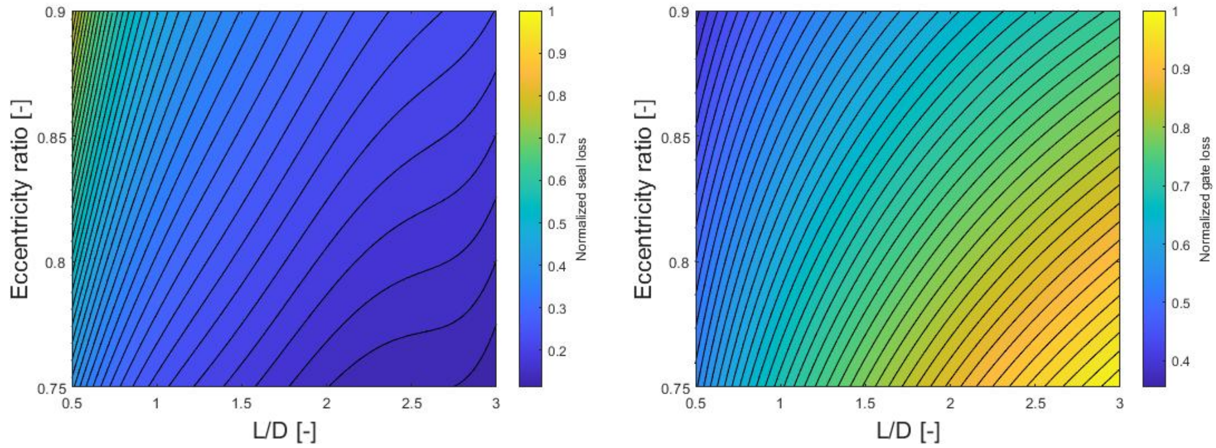


Figure 5: Volumetric efficiency for R134a at 45/130F.



(a) Seal losses

(b) Vane losses

Figure 6: Mechanical losses in spool compressor for R134a various aspect ratio at 45/130F condition.

The study's overarching goal is to find out the optimum displacement range for spool compressors using the low-GWP refrigerants. The current work will be extended to find out the best aspect ratio for different size spool compressors. A coherent methodology will be adopted to scale the various compressor components. The final aspect ratio for compressor size will be simulated for different refrigerants to find out the best suitable compressor size for each refrigerant. The goal is to find optimum geometry for each size compressor on R1234ze(E)

NOMENCLATURE

$\dot{m}_{in}$	Total mass entering the control volume	[kg/s]
$\dot{m}_{out}$	Total mass leaving the control volume	[kg/s]
$\dot{Q}_{in}$	Heat transfer between refrigerant and comp. wall	[kj/s]
ϵ	Eccentricity ratio	[-]
ϕ	Leakage gap at TDC	[in]
ρ	Density	[kg/m^3]
D_f	Inner diameter of the seal	[in]
$D_{out,f}$	Outer diameter of the seal	[in]
e	Eccentricity	[in]
g_{TDC}	Leakage gap at TDC	[in]
h_{in}	Specific enthalpy of the refrigerant at suction	[J/kg]
h_{out}	Specific enthalpy of refrigerant at discharge	[J/kg]
h_{stator}	Stator length	[in]
P	Pressure	[kPa]
PR	Pressure ratio	[-]
$R_{endplate}$	End-plate radius	[in]
R_r	Radius of rotor	[in]
R_s	Stator radius	[in]
T	Instantaneous temperature in control volume	[K]
t	Time	[s]
T_{para}	Parasitic torque	[lbf-in]
u	Specific internal energy	[J/kg]
V	Instantaneous volume of the control volume	[m^3]
V_{disp}	Displacement volume	[in^3]
$W_{seal,face}$	Width of the seal	[in]
w_w	Radial width of the wrap around gap	[in]
y	offset parameter to adjust the parasitic losses for various displacement	[-]

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