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Design Improvements of a Rotating Spool Compressor using a Comprehensive Model

Seminar 65 - Compression Challenges for Low-GWP Refrigerants

Orlando, Florida

Learning Objectives

1. Provide an overview of the novelty compressor designs
2. Evaluate the utility of the comprehensive model as a tool used for compressor design
3. Apply the new modeling tools for compressors
4. Describe the compression mechanism of a spool compressor

Acknowledgments

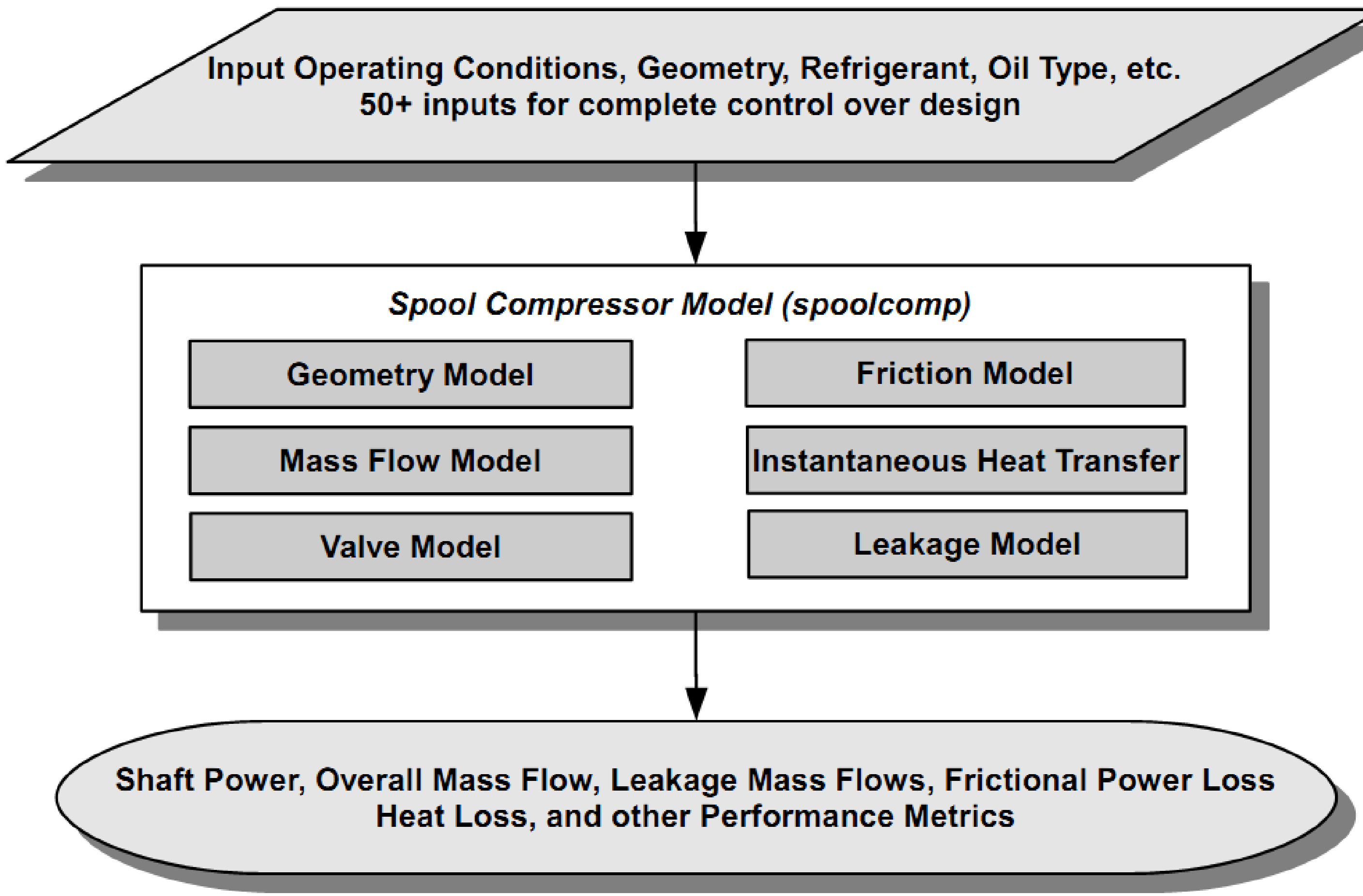
- Greg Kemp, CEO, Torad Engineering
- Joe Orosz, COO, Torad Engineering

Overview

- What is the spool compressor?
- The spool compressor model
  - General principals
  - Included sub-models
  - Performance predicting experimental results
  - Limitations
  - Utility (5th vs. 6th Generation prototype performance)
- Scaling study
  - Study motivation and givens
  - R134a device scaling
  - Seal analysis and design
  - Final basis for prototype design

What is the Spool Compressor?

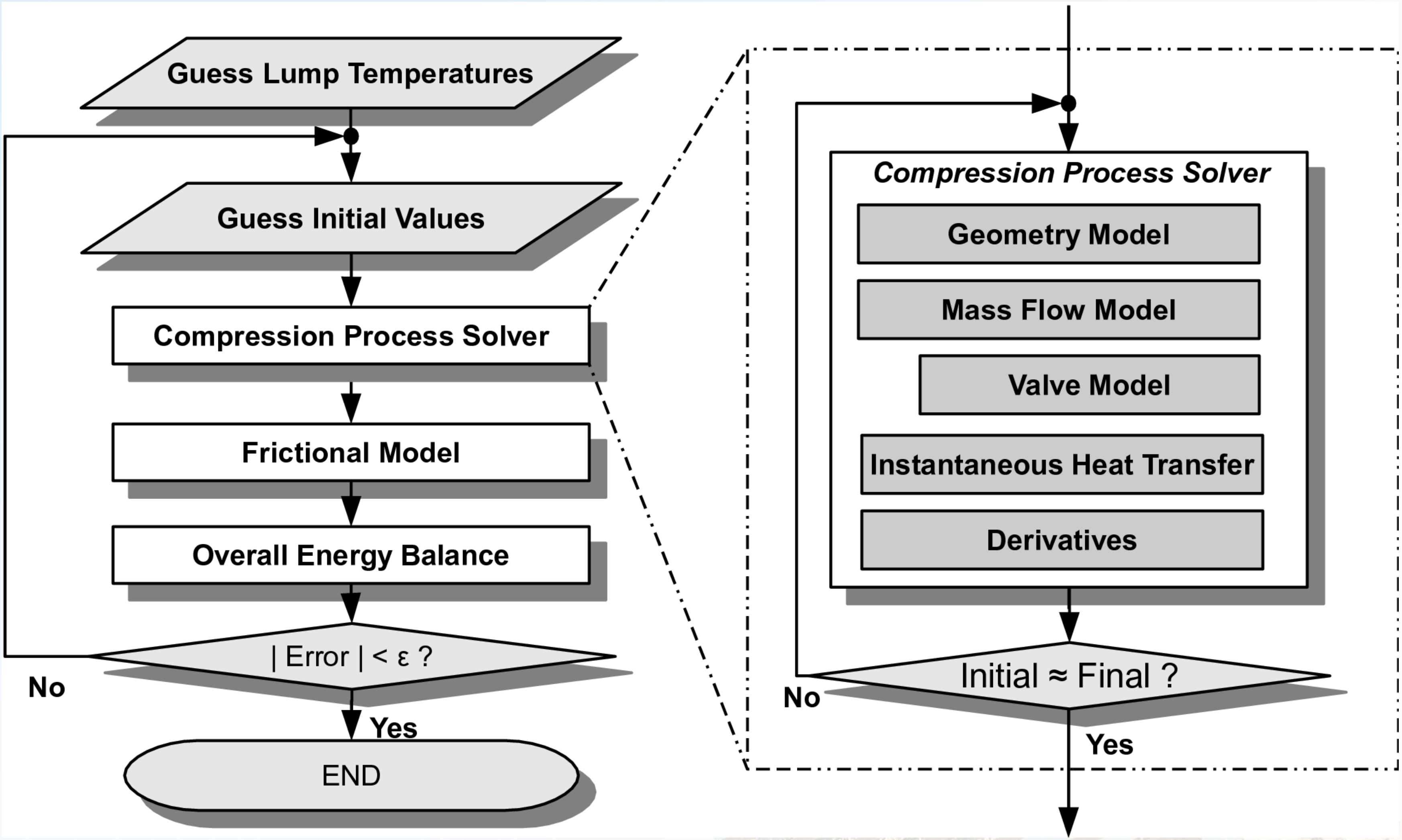
Basic Model Flowchart



Advantages and Limiting Assumptions

- Advantages
  - Highly accurate prediction of performance metrics
  - Captures unforeseen interactions between different components within the compressor
  - Once validated, can be used to predict performance of future design iterations of the compressor (Optimization)
- Limiting Assumptions
  - Assumes each chamber has a uniform pressure and temperature at each instant in rotation
    - Hinders ability to isolate extreme pressure and temperature differences within the compressor

Rotating Spool Compressor Comprehensive Model



Full model details presented in Bradshaw and Groll (2013)

Compression Process Solver

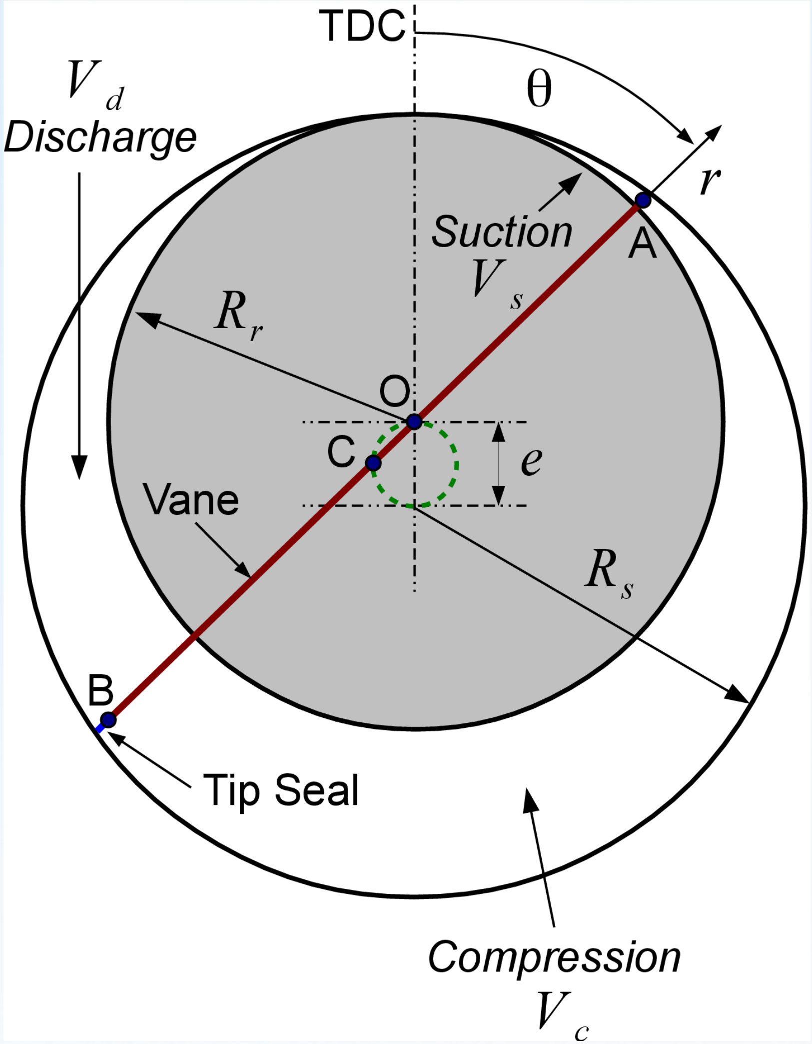
- Combined mass and energy balance can be solved in series for dρ/dθ and dT/dθ:

(properties) (leakage) (geometry) (heat transfer)

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

$$\frac{dT}{d\theta} = \frac{-\rho h\frac{dV}{d\theta} - \left(uV + \rho V\frac{\partial u}{\partial \rho}\right)\frac{\partial \rho}{\partial \theta} + \frac{1}{\omega}\left(\dot{Q} + \sum \dot{m}_{in}h_{in} - \sum \dot{m}_{out}h_{out}\right)}{\rho V\frac{\partial u}{\partial T}}$$

Spool Geometry Definitions



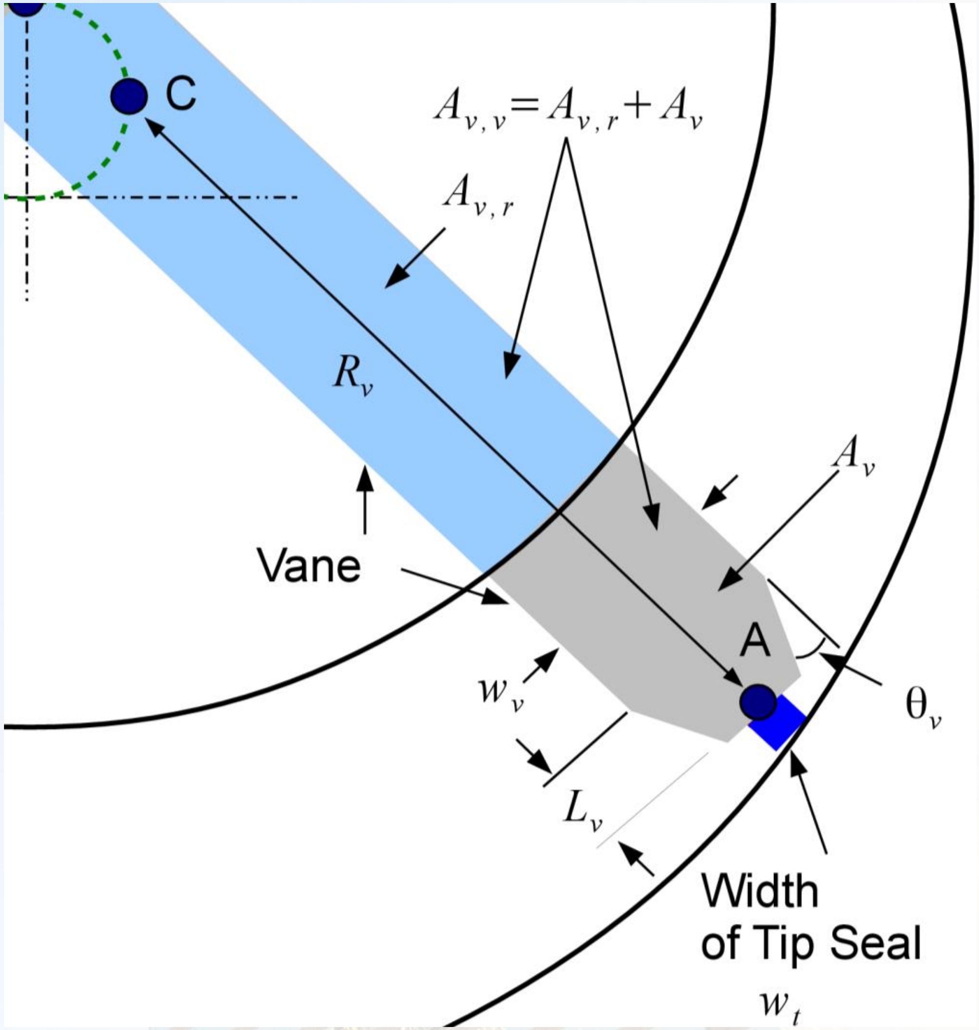
Geometry Definitions from Bradshaw and Groll (2013)

- At any given instant there are three distinct working chambers separated by the vane/TDC
- Eccentricity is defined as the difference between the rotor and stator radii
- Since the mechanism is planar the approach first uses polar coordinates to calculate a surface area which is then multiplied by the depth

$$V(\theta) = A(\theta)h_s$$

Vane Geometry Correction

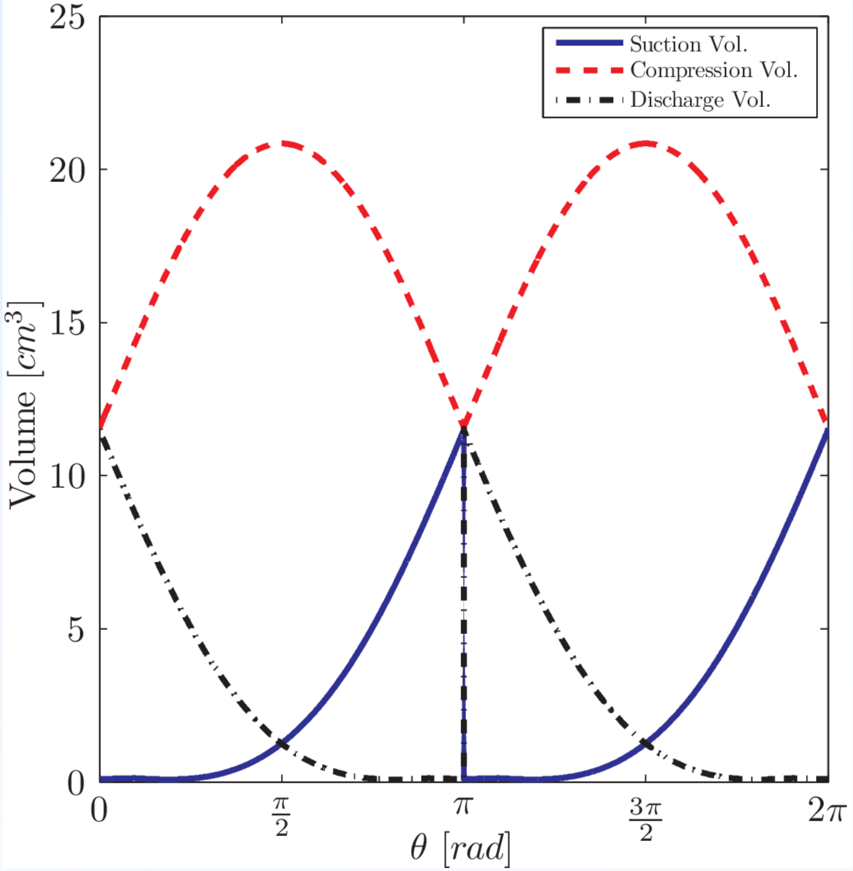
- Volume of chambers modified to account for the volume of the vane geometry
- Depending on location the vane can either add or subtract volume to the working chamber
- Special consideration for the vane geometry is taken as the vane crosses the TDC plane



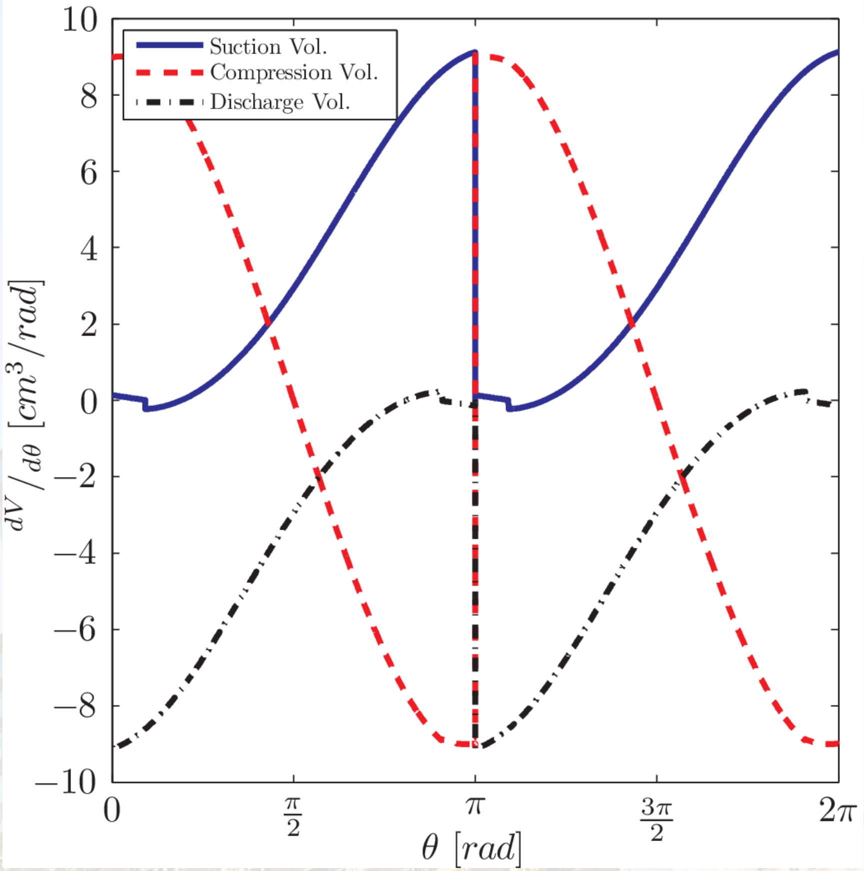
Vane and tip seal geometry definitions from Bradshaw and Groll (2013)

Geometry Model Results, 5th Generation Spool Compressor

Volumes

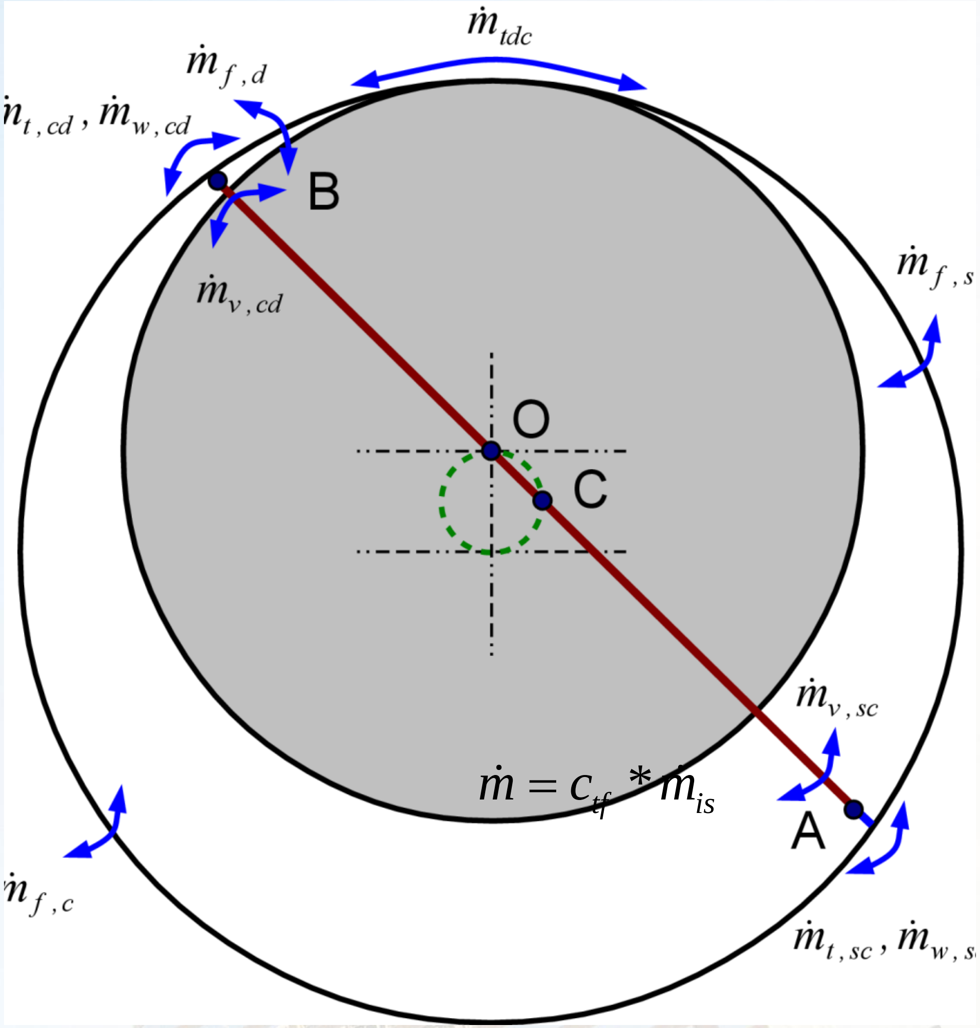


Change in volume



Leakage paths and models used

- 10 leakage paths identified
- All modeled as isentropic compressible flow of an ideal gas
- Modified with a correction factor
- correction used to 'tune' massflow predictions

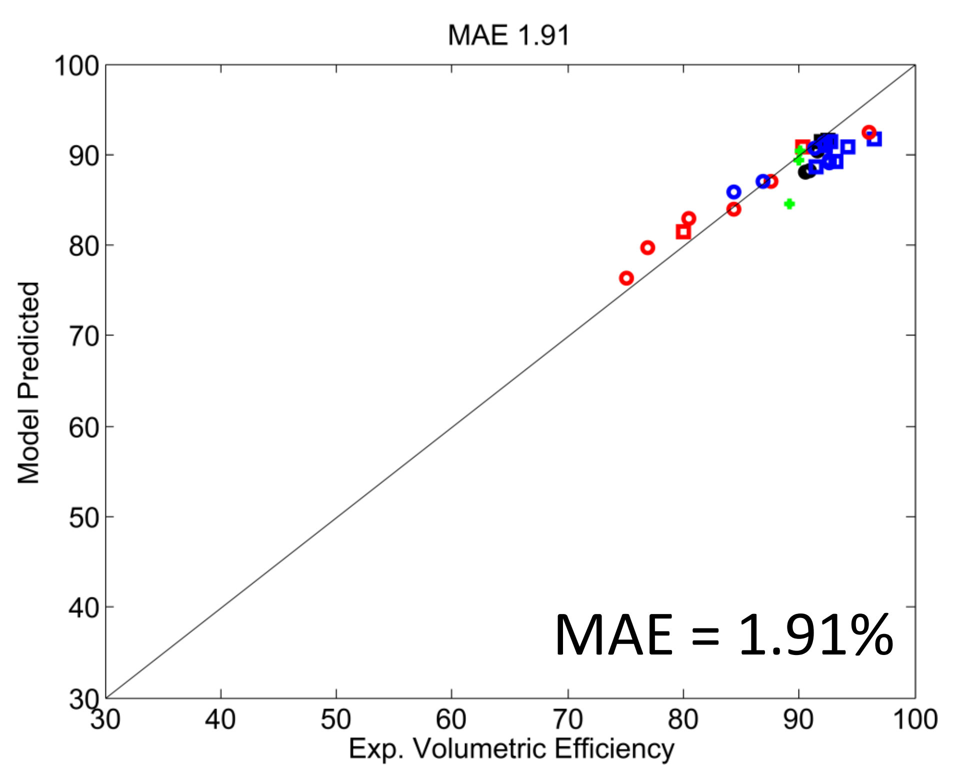


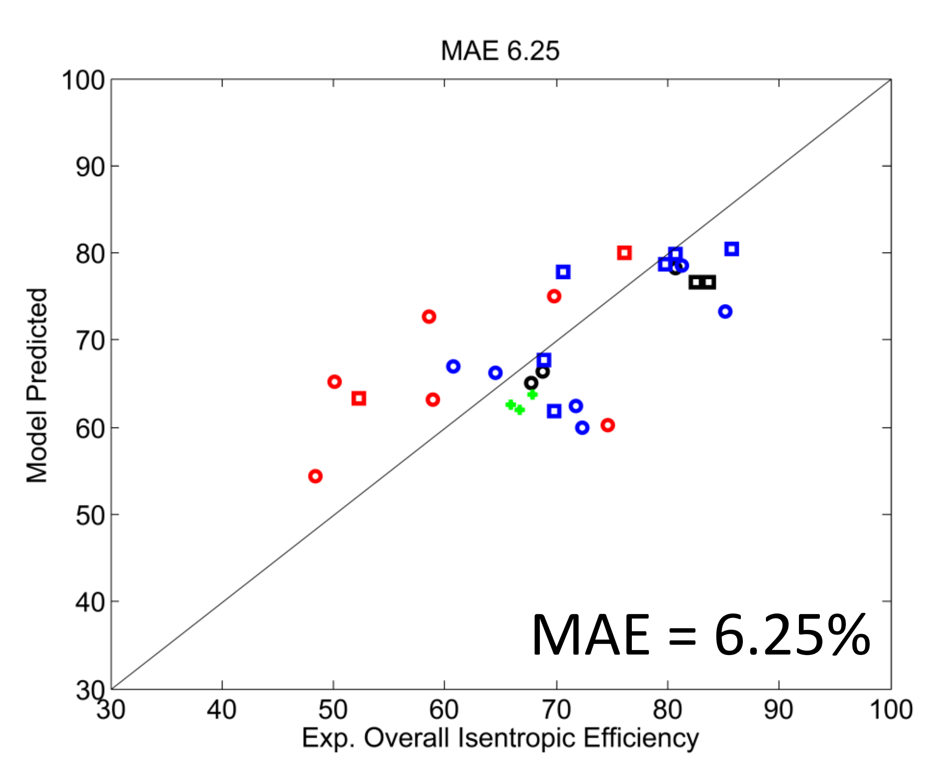
Leakage path definitions from Bradshaw and Groll (2013)

Friction Sub-Models

- Spool Seals (Bradshaw et. al 2015)
  - Oil shear plus dry friction on rotating endplates
- Tip Seals (Bradshaw 2013)
  - Oil shear along cylinder bore
- TDC (Bradshaw and Groll 2013)
  - Oil shear between rotor and cylinder
- Vane (Bradshaw and Groll 2013)
  - Coulomb friction between vane and rotor slot
- Viscous Drag of Rotating Endplates (Bradshaw et. al 2015)
  - Drag generated by rotating endplates rotating in enclosed space with lubricant

Model Predictive Capability, 2 Prototype Platforms



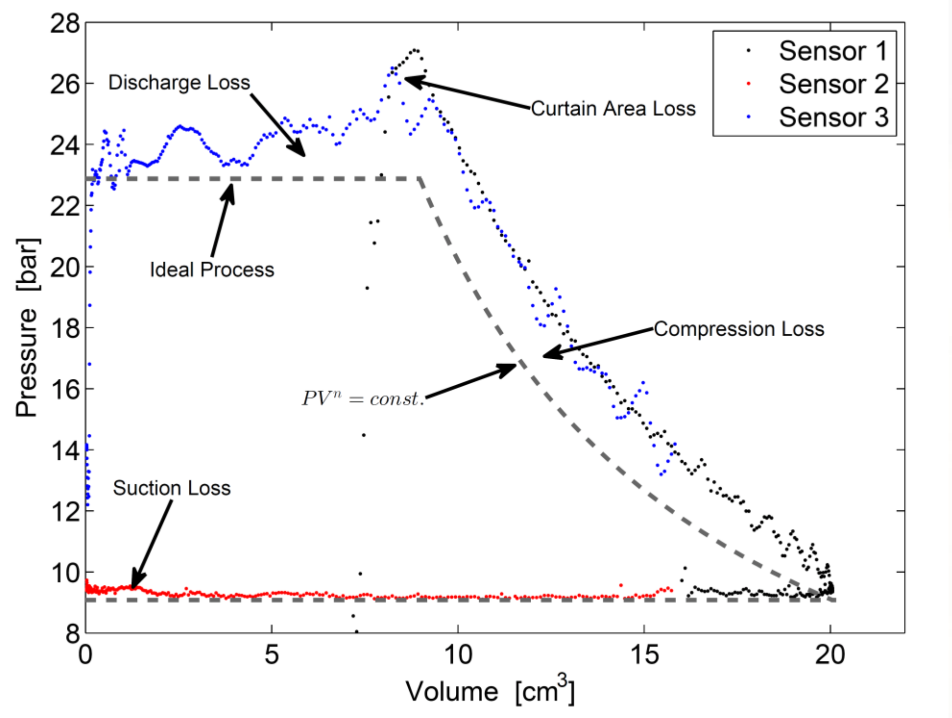


Platform	V_{disp} (in^3)
5th Gen.	2.40
6th Gen.	3.33

□, □	SEER B/A
□	Superheat
○	Baseline (40/100), 2500 rpm
○	Cond. Temp
○	Evap. Temp
+	Speed

Model Limitations

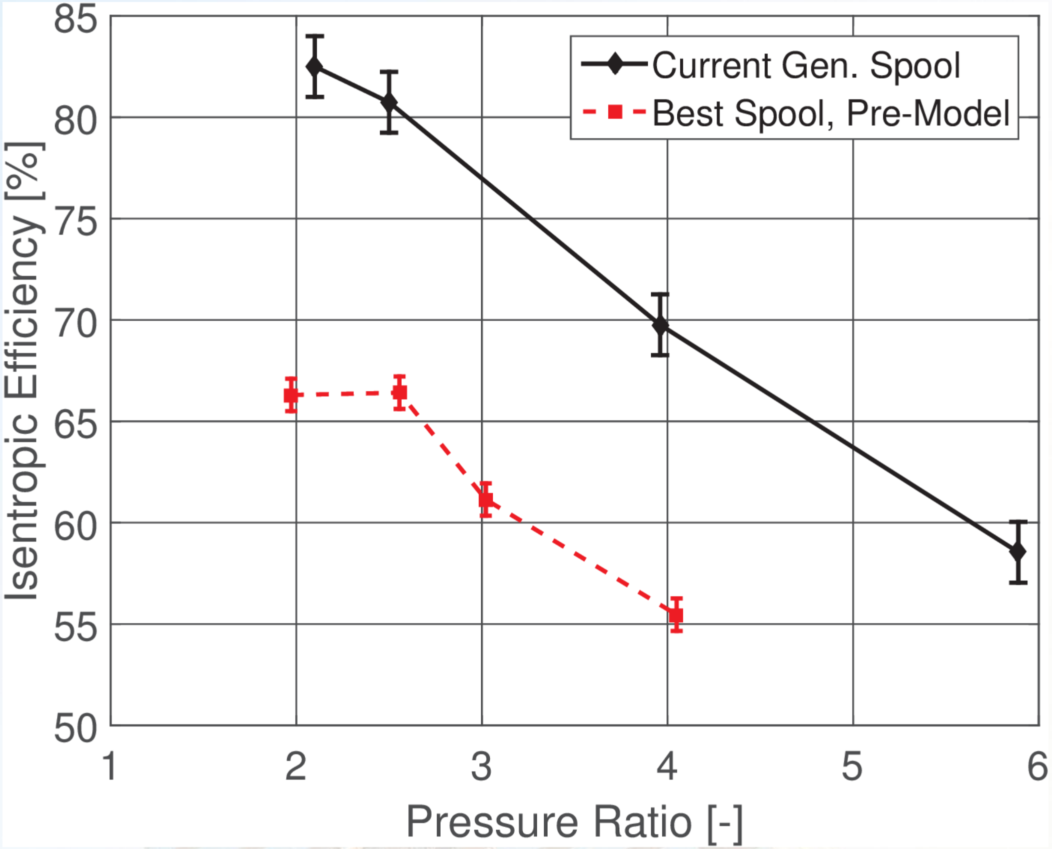
- Valve dynamics and porting
  - Valve dynamics are computationally expensive and not used when exploring large spaces
  - Port losses are not well captured
- Frictional losses and heat balance
  - Vane, tip seal, and TDC friction sub-models difficult to experimentally validate
  - More detailed heat flow paths should be added (most pertinent to hermetic and semi-hermetic designs)



Experimentally obtained indicator diagram of 5th generation spool compressor with R410A as the working fluid, from Bradshaw et al., 2015

Improvements Using Model

- Current generation spool (6^{th} Gen.) compared with previous generation (5^{th} Gen.)
- 5^{th} generation prototype was designed prior to development of modeling tool
- 6^{th} generation utilized modeling tool (described in Bradshaw et al. (2014, 2015)
- Methodology described in 7^{th} generation design study



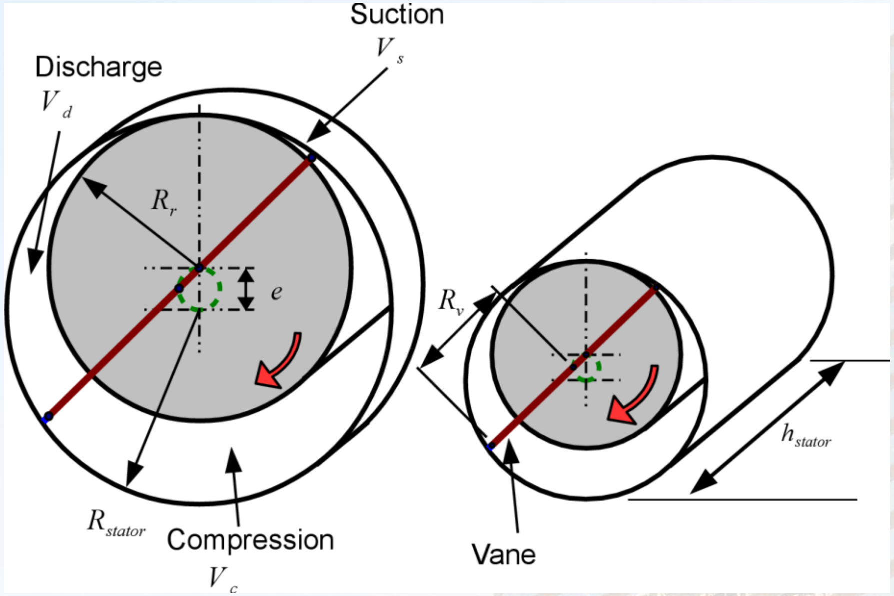
Overall isentropic efficiency as a function of pressure ratio for the 5^{th} and 6^{th} generation spool compressors (data from Orosz et al. 2014)

Design Study

- **Objective:** Design a rotating spool compressor for light-commercial air-conditioning
- **Givens:**
    1. Design for 40F/100F evaporating and condensing temperatures, respectively
    2. Working fluid shall be R134a
    3. Rotational speed shall be 1750 rpm
- **Design Variables:**
    1. Compressor Displacement (Cooling Capacity)
    2. Intrinsic Compressor Geometric Variables

Scaling Study

- For each displacement, the key geometry was varied
  - Eccentricity ratio; Rotor radius/Stator Radius
  - Length to diameter ratio; Stator axial length/Stator diameter



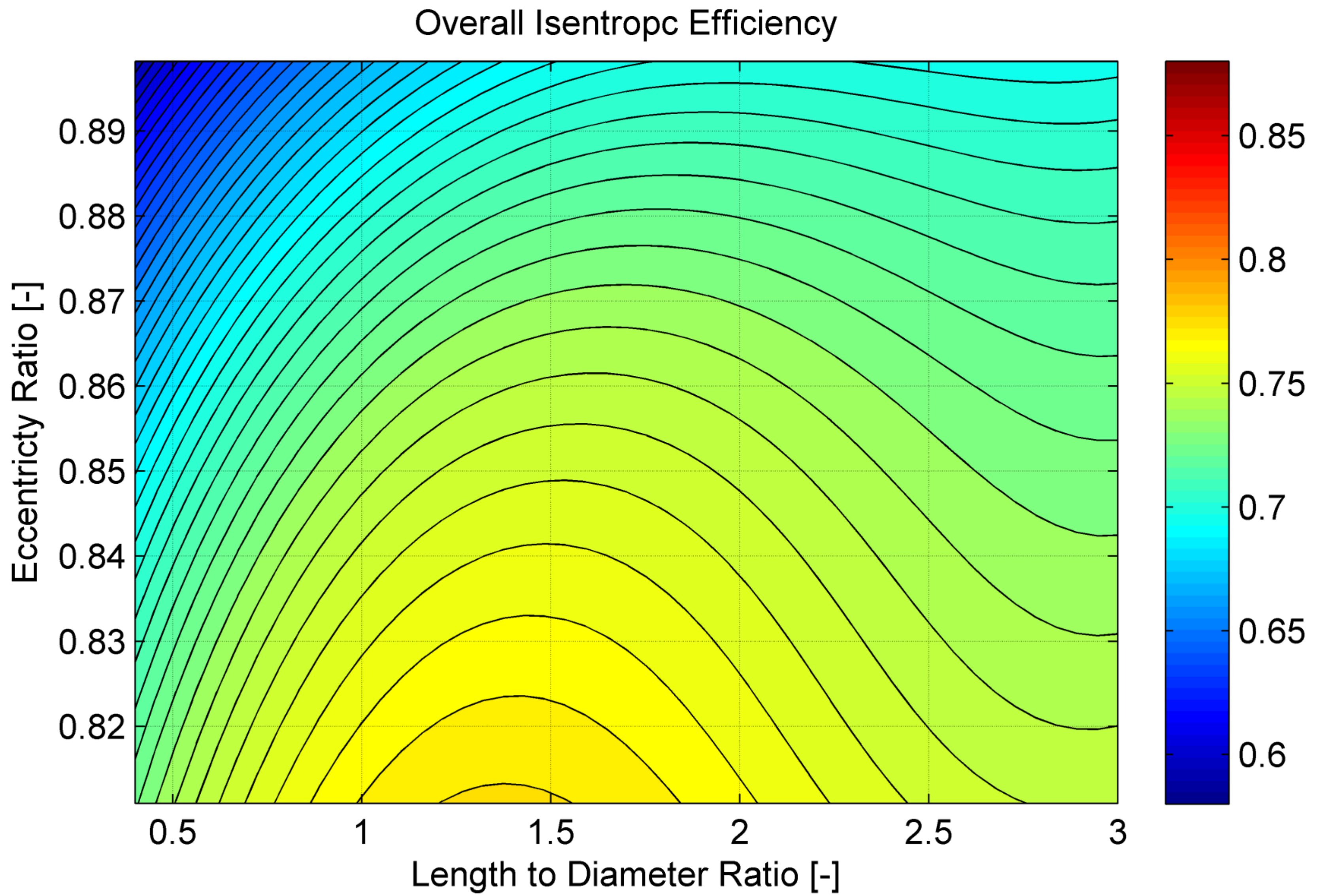
- 4 displacements on R134a
  - 5, 20, 40, 80 Tons

Scaling Study, cont'd

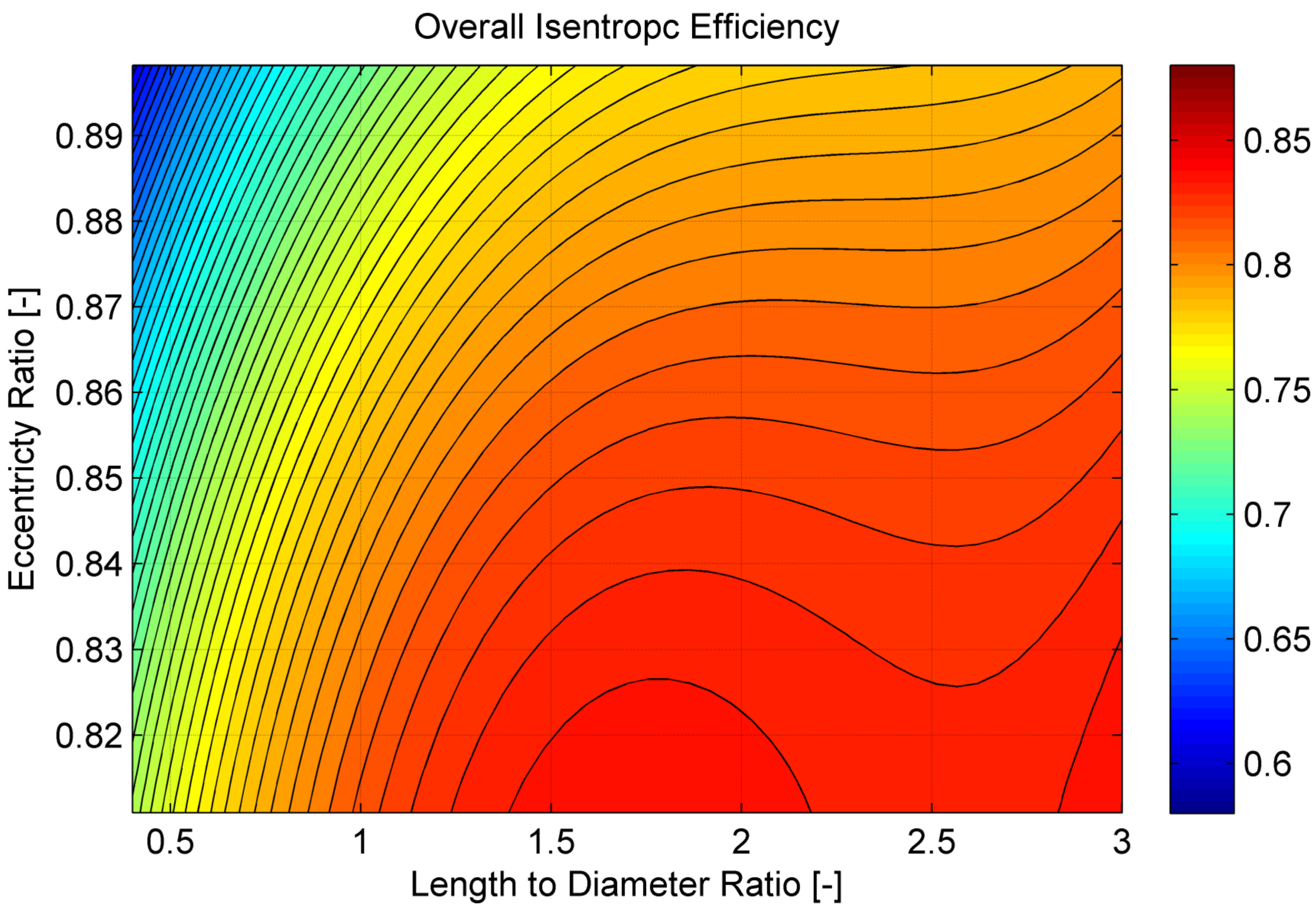
- Water-cooled conditions (40/100)
- 1750 rpm compressor speed
- Manufacturing tolerances applied to length of compressor at 0.0001 in/in
- Design rules are incorporated that force geometry constraints for manufacturing and reliability considerations.
  - Total eccentricity
  - Rotor diameter
  - Eccentric bearing size.
- Calculated displacements, assuming 90% volumetric efficiency

Tons	5	20	40	80
V_{disp} (in^3/rev)	15.48	61.90	123.8	247.6
V_{disp} (cm^3/rev)	254.7	1014	2029	4057

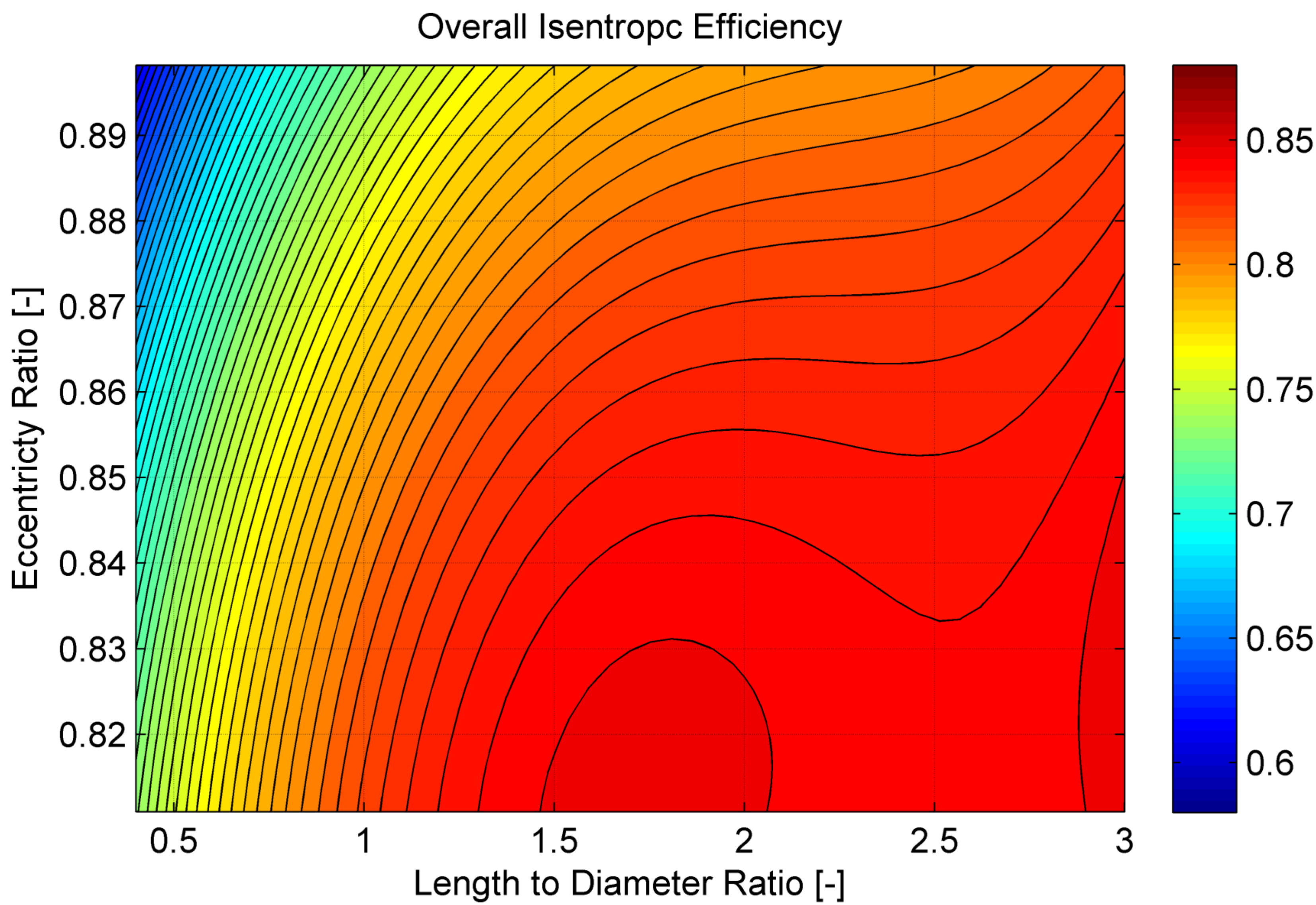
R134a Results – 5Ton



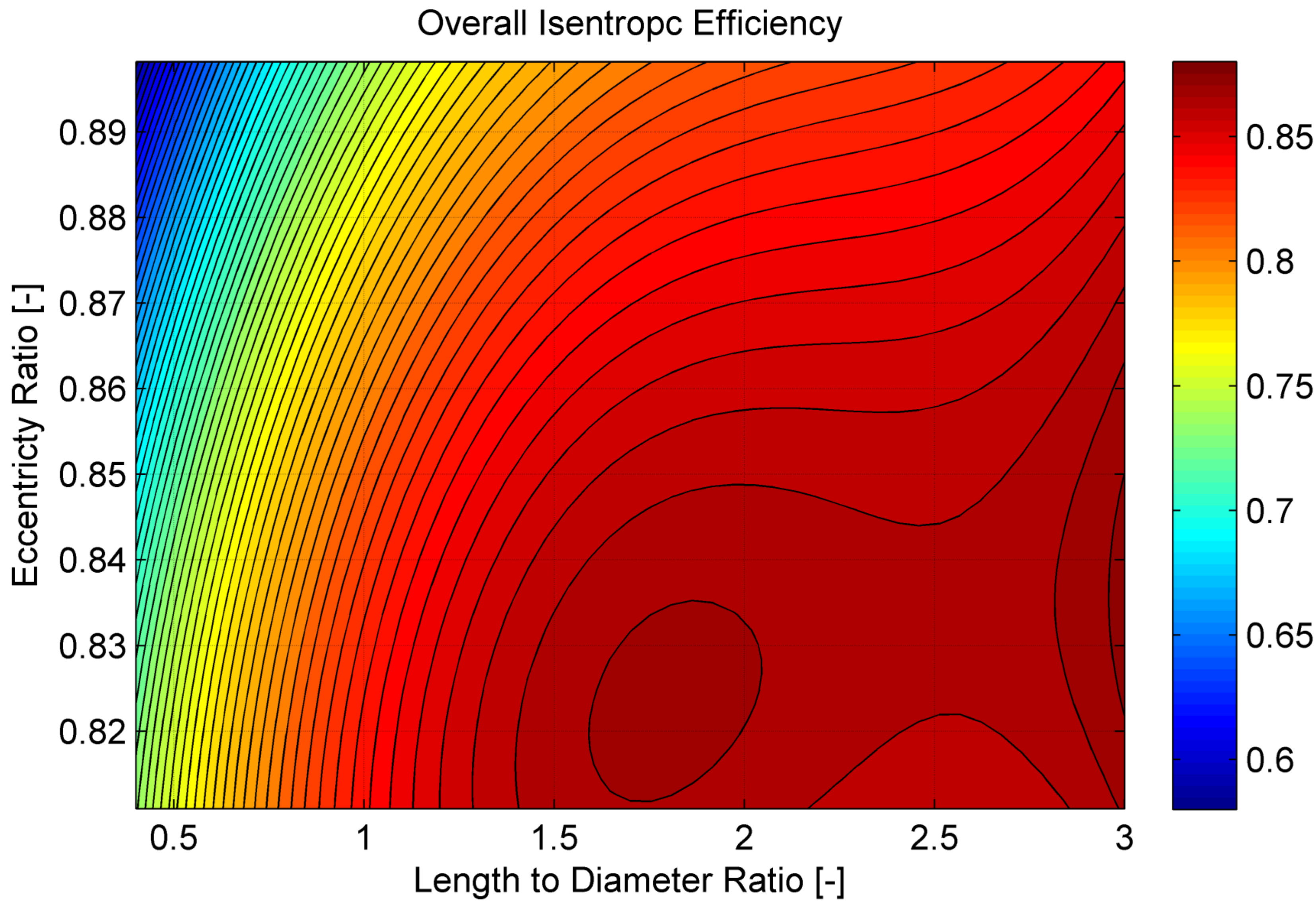
R134a Results – 20Ton



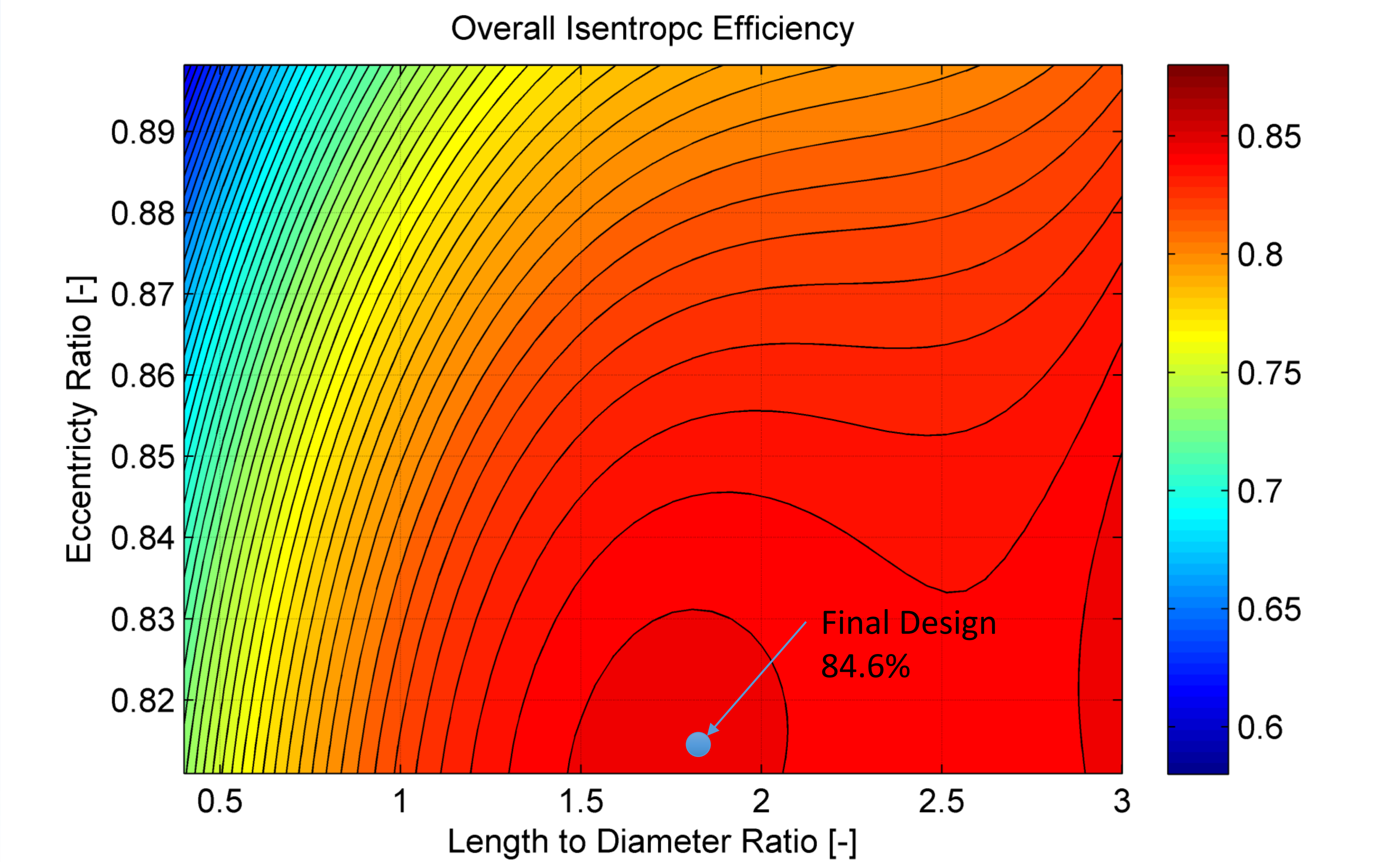
R134a Results – 40Ton



R134a Results – 80Ton



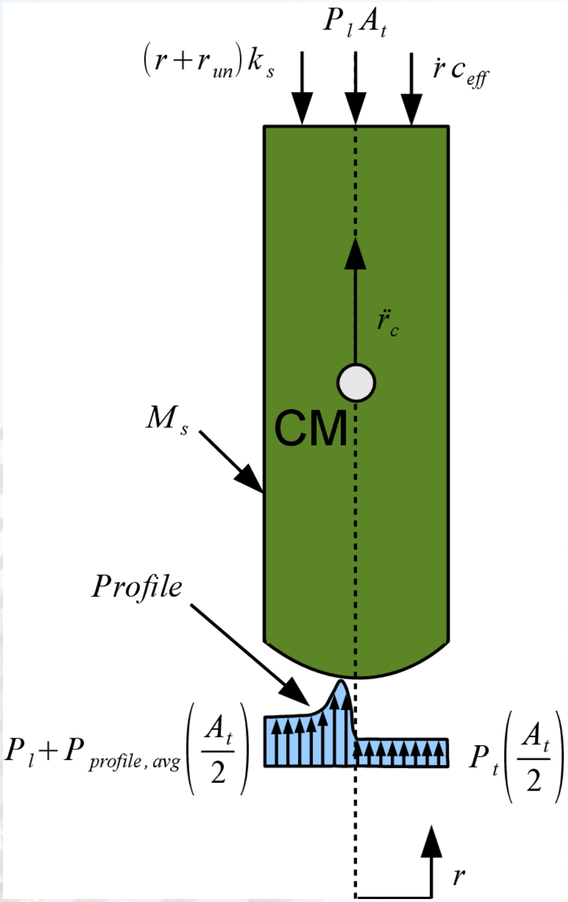
R134a Results – 40Ton



Tip Seal Design

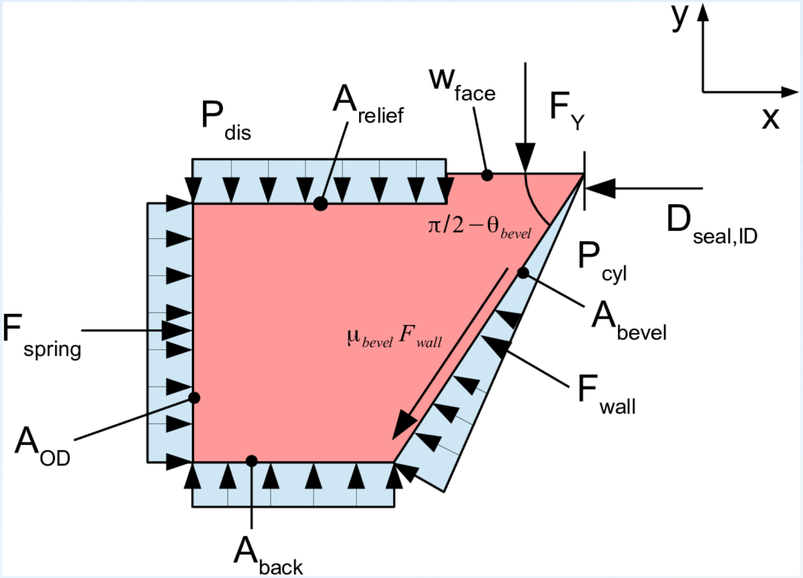
- Used modeling tool developed for tip seals, presented in Bradshaw (2013)
- Matched net tip seal force on cylinder bore per unit axial length relative to the 6th generation compressor
- Symmetric radius tip

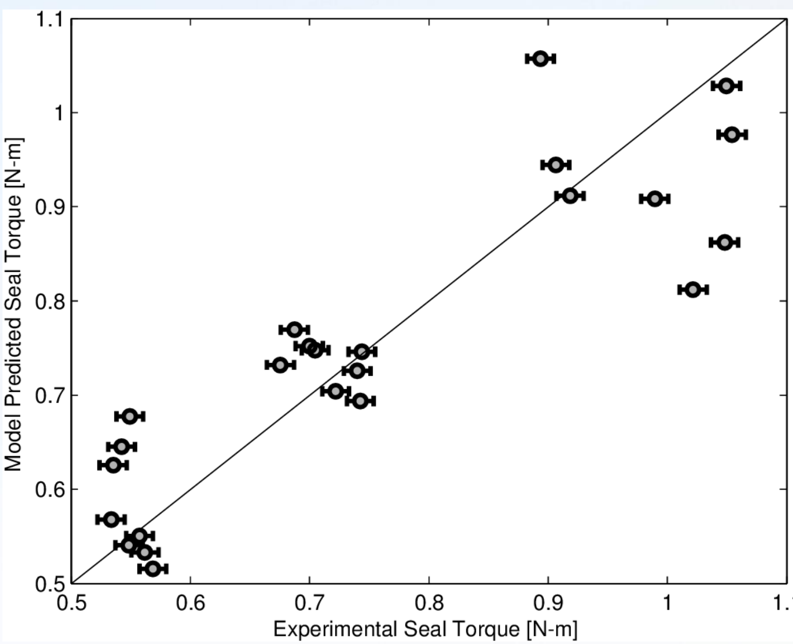
	Avg. Force	Force/Length	Mass	Tip Radius
	N	N/m	g	mm
6th Gen.	15.1	297	9.5	2.36
40 Ton	74.7	297	39.7	3.81



Free body diagram of tip seal used in design tool, from Bradshaw (2013)

Spool Seal Design





- Design tool developed and validated to evaluate bevel seal designs, presented in Bradshaw et al. (2015)
- Utilized side seal design tool to match a similar face load per unit volume as 6th generation compressor

	V_{disp}	Model Torque	Ratio
	cm^3	N-cm	N/cm^2
6th Gen.	54.7	37	1.94
40 Ton	1024	1100	1.56

Results Summary – Best Efficiency Estimates

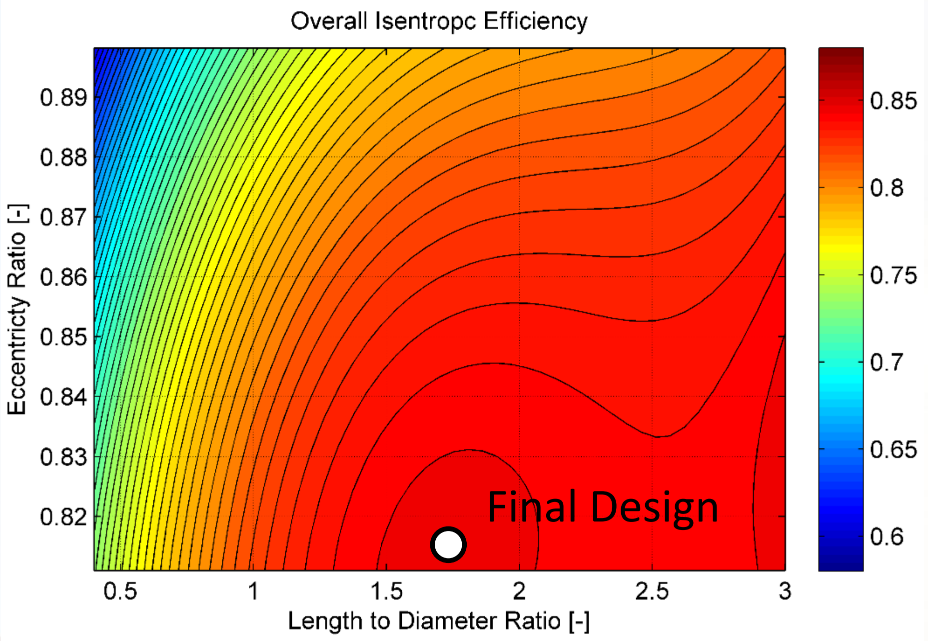
Cooling	Fluid	Shaft Eff.*	Motor Eff.	Total Eff.**
TonsR	-	%	%	%
5	134a	77.5%	90.0	67.8%
20	134a	83.9%	91.0	73.5%
40	134a	84.6%	92.0	75.8%
80	134a	87.2%	92.0	78.2%

*Model predicted values

**Efficiency uncertainty estimated at roughly 5% points

40 Ton – R134a Design Details

Ecc. Ratio	-	0.811
L/D	-	1.82
Rotor Dia.	in (cm)	4.411 (11.2)
Cylinder Dia.	in (cm)	5.440 (13.8)
Cylinder Length	in (cm)	9.890 (25.1)
Vane Width	in (cm)	1.090 (2.77)
Isentropic Eff.	%	84.6
Volumetric Eff.	%	92.6
Discharge Temp	F (C)	128 (53.3)
Massflow	lbm/min (kg/min)	118 (53.5)
Shaft Power	kW	23.5
% Carryover Vol.	%	1.76



- Final design from model requires detailed design work to account for
  - Strength and resilience
  - Manufacturability
  - Location of parts (i.e. ability to assemble)

Conclusions

- A rotating spool compressor is described
- The spool compressor model and key sub-models are described
- A methodology for compressor design using modeling tools is presented
- The methodology has shown great success between the 5th and 6th generation and now been applied to a 7th generation

Works Cited

- Bradshaw, C.R., Groll, E.A., 2015. Development of a Loss Pareto for a Rotating Spool Compressor Using High-Speed Pressure Measurements and Friction Analysis. *Applied Thermal Engineering.*(accepted without revision).
- Bradshaw, C.R., Groll, E.A., 2013. A Comprehensive Model of a Novel Rotating Spool Compressor. *International Journal of Refrigeration.* **36**(7), 2007 - 2013.
- Bradshaw, C.R., Orosz, J., Kemp, G., Groll, E.A., 2014. Influence of Volumetric Displacement and Aspect Ratio on the Performance Metrics of the Rotating Spool Compressor. *In: Proceedings of the International Compressor Engineering Conference, Purdue University, West Lafayette, IN, USA*. No. 1177.
- Bradshaw, C.R., 2013. Spool Compressor Tip Seal Design Considerations. *In: 8th International Conference on Compressors and their Systems, City University, London, UK*. 341 - 351.
- Orosz, J., Kemp, G., Bradshaw, C.R., Groll, E.A., 2014. An Update on the 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. 1378.

Questions?

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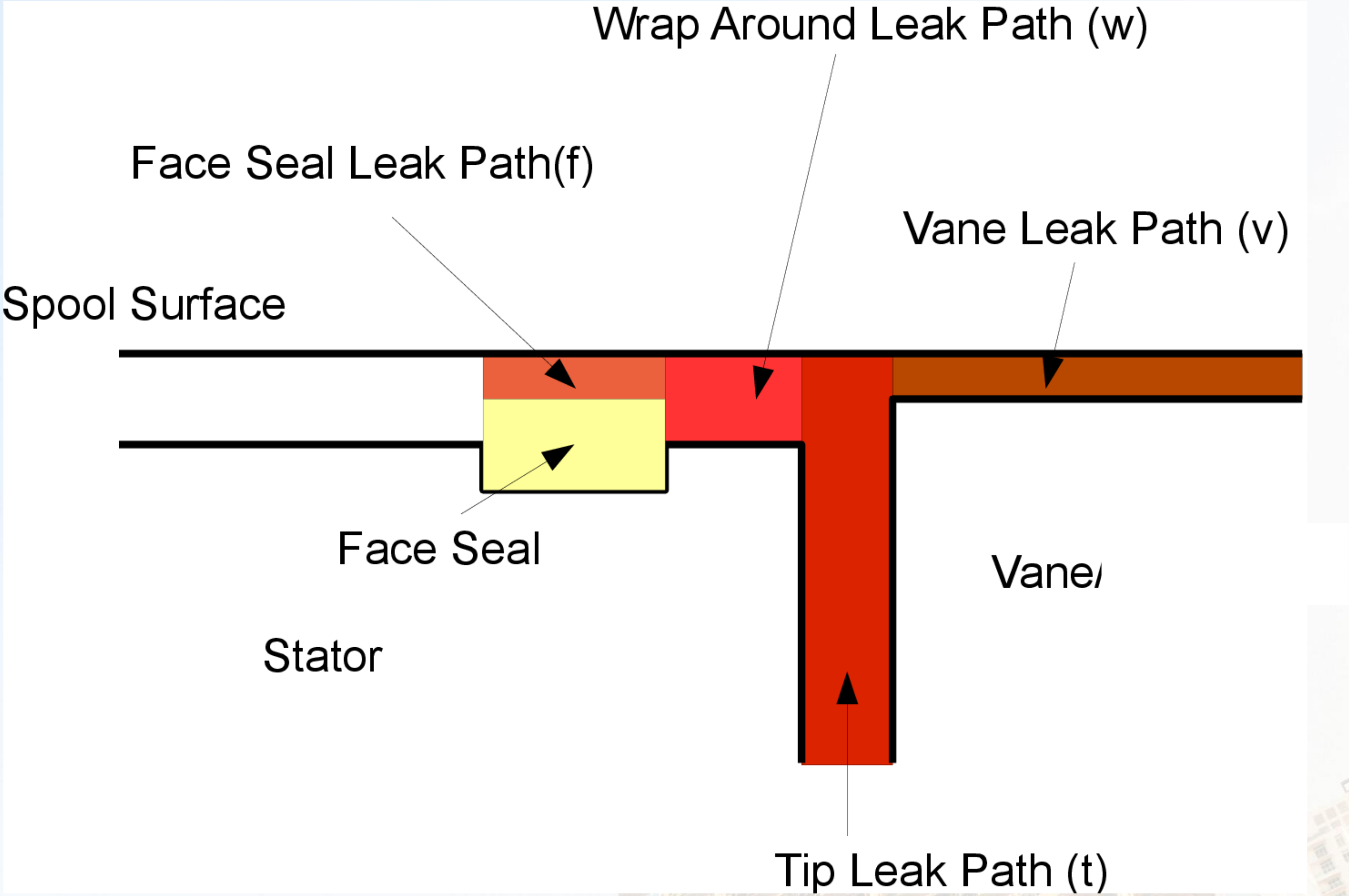
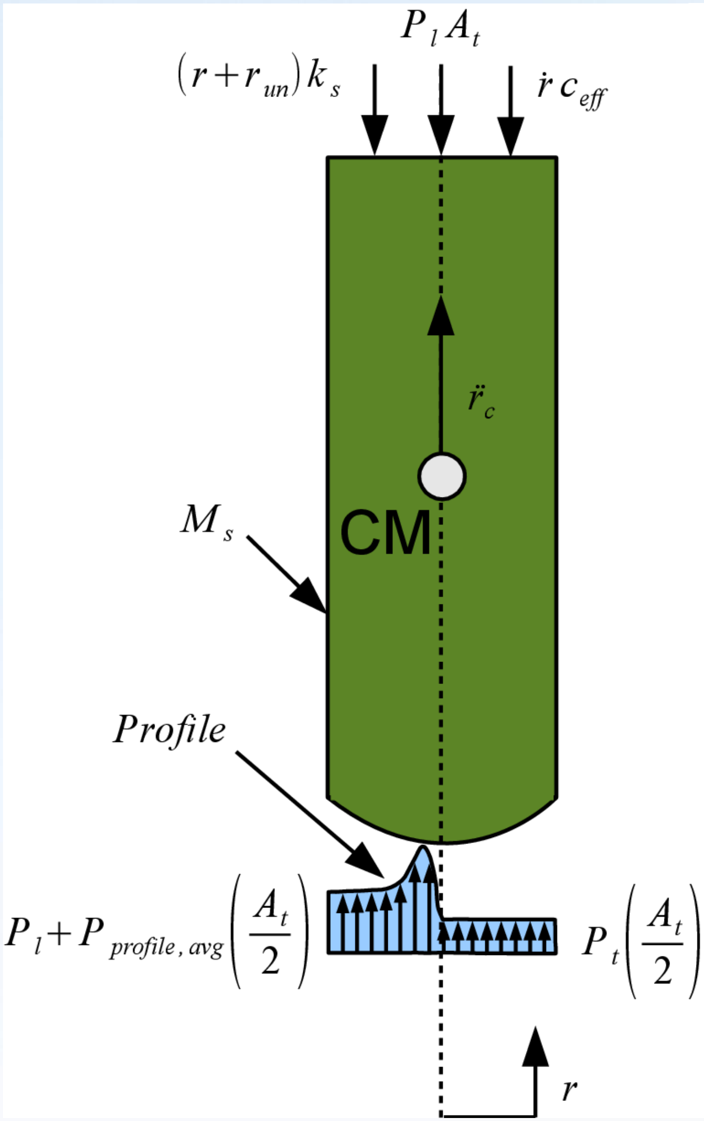
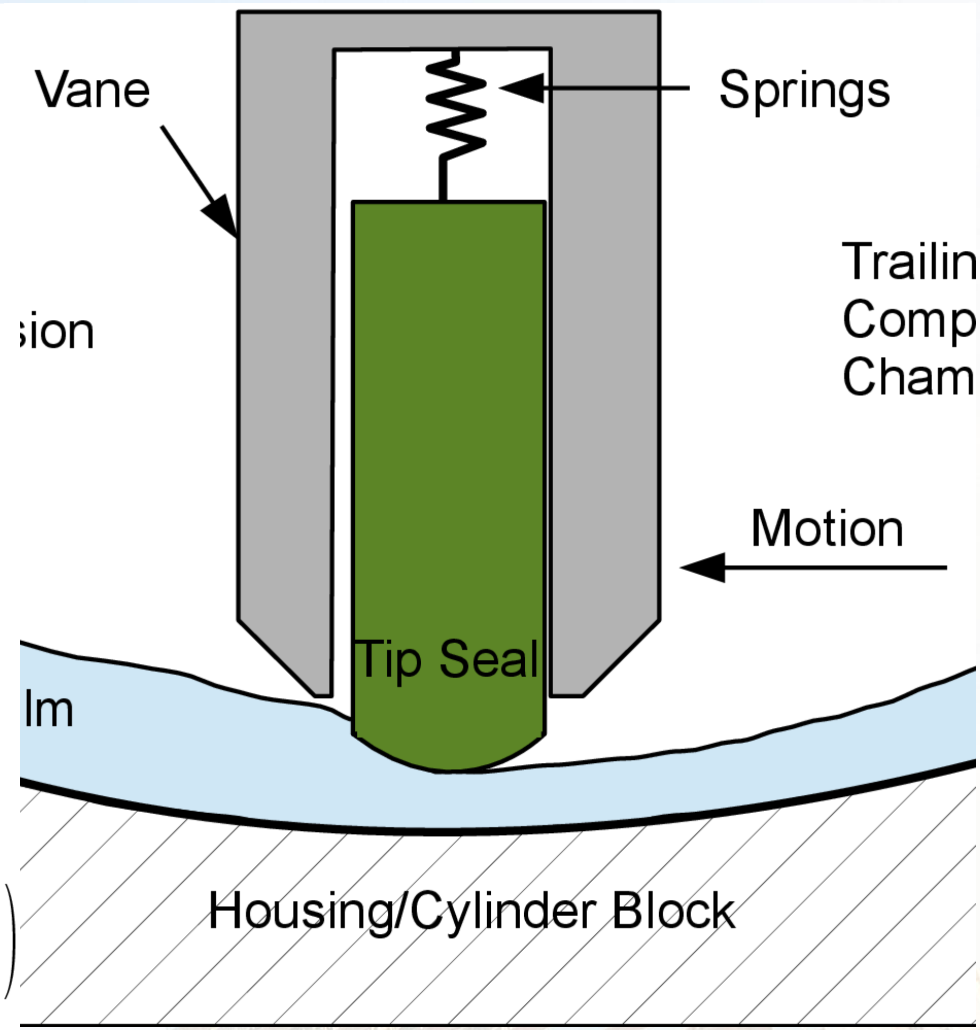
Radial View of Leak Paths




$P_l A_t$
$(r+r_{un})k_s$
$\dot{r}c_{eff}$
$\ddot{r}_c$
CM
M_s
Profile
$P_l+P_{profile,avg}\left(\frac{A_t}{2}\right)$
$P_t\left(\frac{A_t}{2}\right)$
r

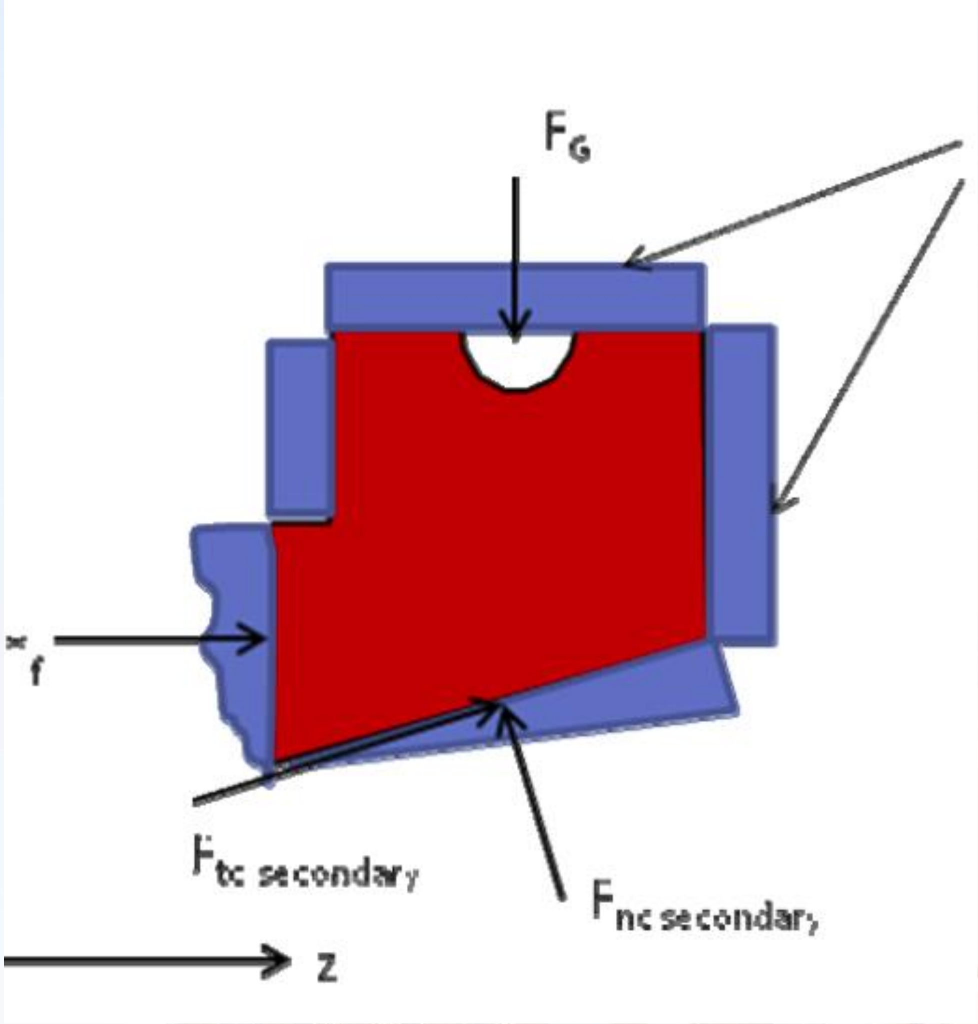

Vane
Springs
Motion
Tip Seal
Housing/Cylinder Block

Spool Seal Modeling

A bevel seal model has been developed which includes oil shear and dry friction components

Model includes:

- Static force balance on spool seal
- Fluid film pressure reaction of the oil/refrigerant mixture on the sealing surfaces



Heat Transfer

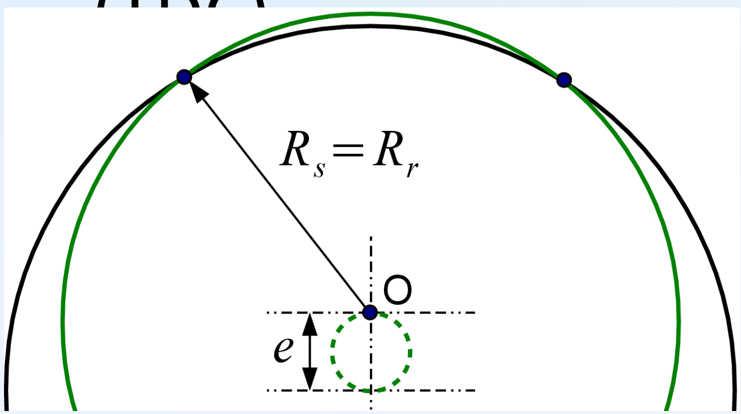
- Modified Dittus-Boelter for use in spiral heat exchangers

$$h_c = 0.023\frac{k}{D_h}Re^{0.8}Pr^{0.4}\left[1.0 + 1.77\left(\frac{D_h}{r_{aver}}\right)\right]$$

- r_{aver} is rotor radii and hydraulic diameters are a function of crank angle for each chamber
- Reynolds and Prandtl number are a function of crank angle

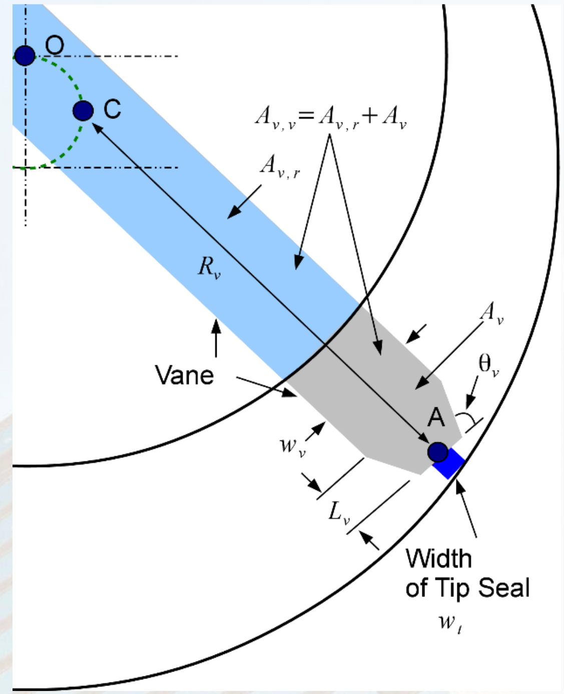
Additional Frictional Losses

- Top Dead Center (TDC)



- Modeled as hydrodynamic oil shear between rotor and cylinder housing

- Vane



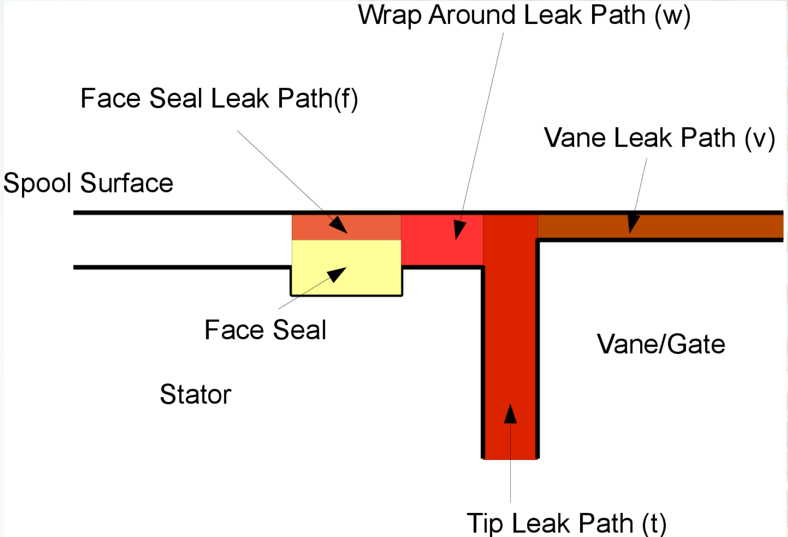
- Modeled as Coulomb friction

Manufacturing Tolerances in Model

- 0.0001”/” (1 micron/cm) on TDC and wrap-around gaps
- Minimum TDC clearance is 0.0005” (~1.25 micron), so formulation is as follows:

$$g_{TDC} = 0.0005 + 0.0001 * \mathrm{h}_{stator}$$

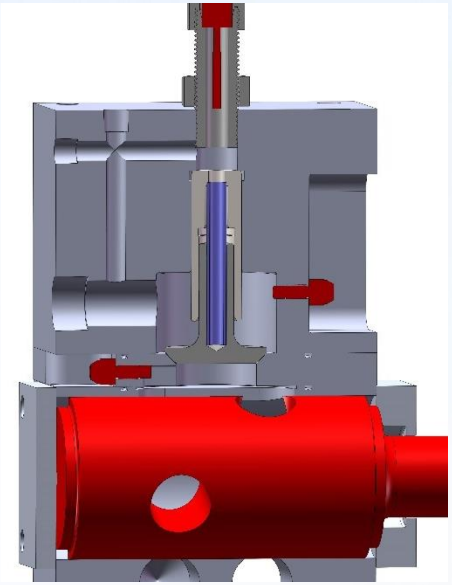
- Same minimum and relationship holds for the wrap-around

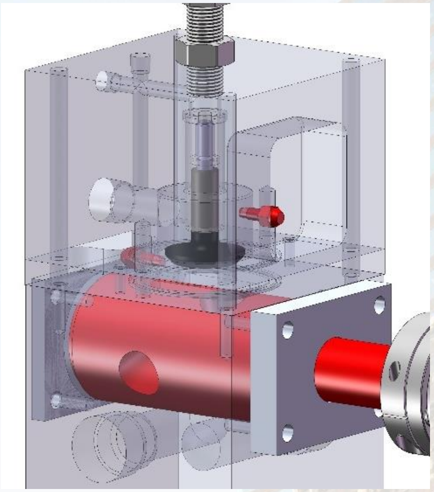


Valve Design

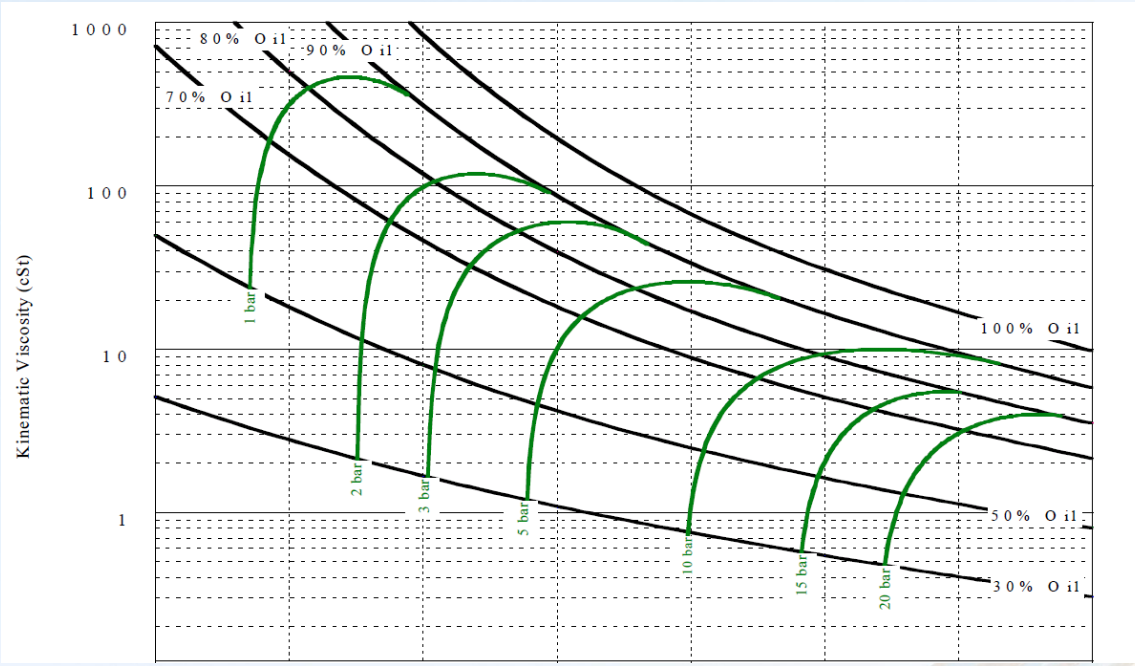
- Modified 6th Gen. valve design for improved mechanical reliability
  - Modified valve not pictured
- Scaled design to appropriate port size for 40 Ton compressor
- Ran 1 pcs. in reliability test mechanism

Compressor poppet valves, 6th Gen (left) and 40 Ton (right)





Oil Considerations



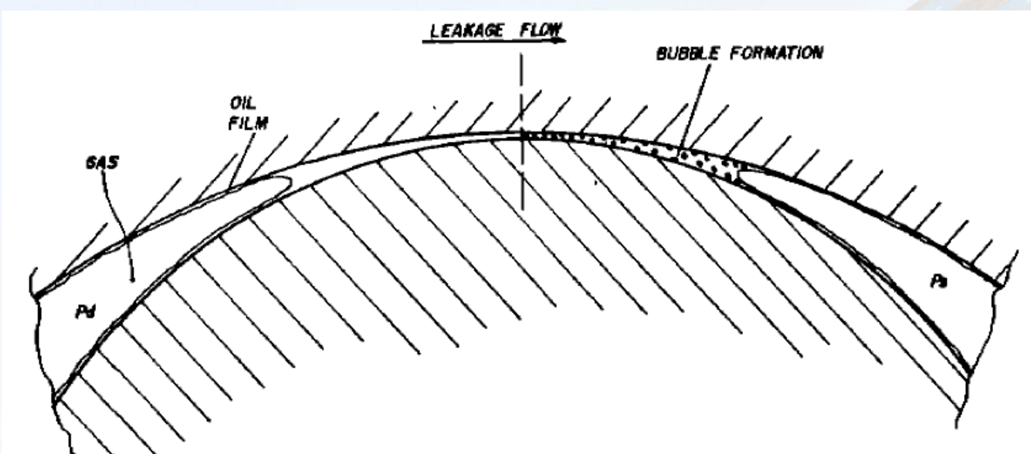
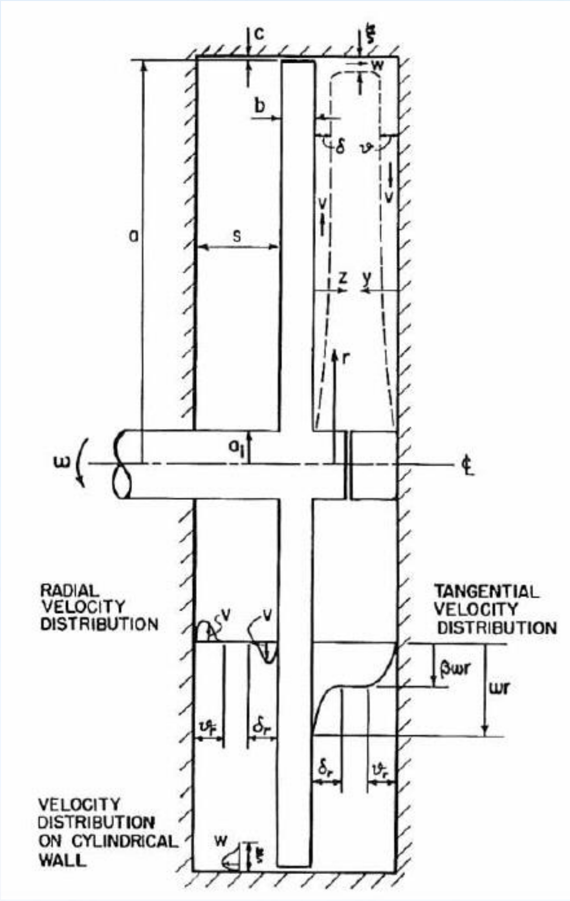
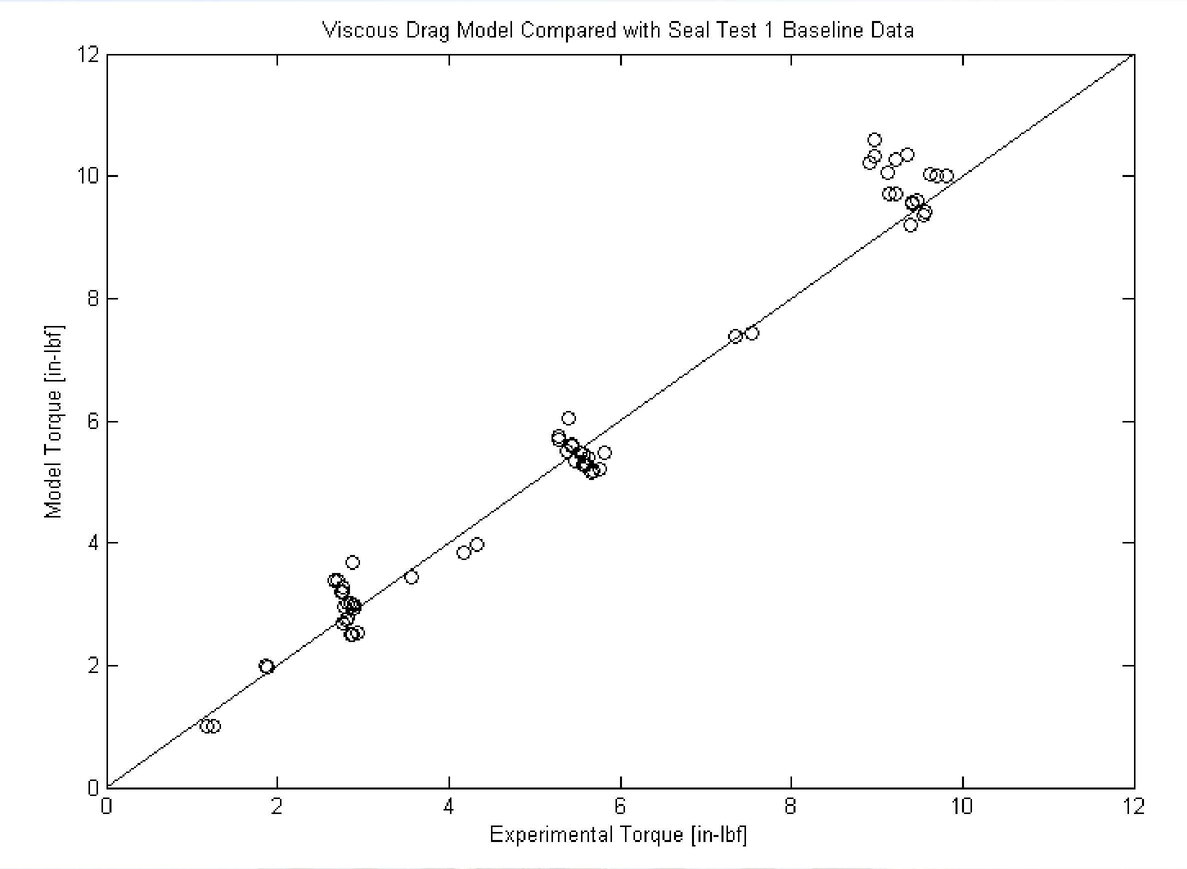


FIG. 3 – LEAKAGE FLOW PATTERN

- Oil properties accounted for in friction analysis in model
  - Drag
  - Solubility influence on viscosity
- Leakage influence is accounted for to a certain degree and continues to be an area of exploration

Viscous Drag

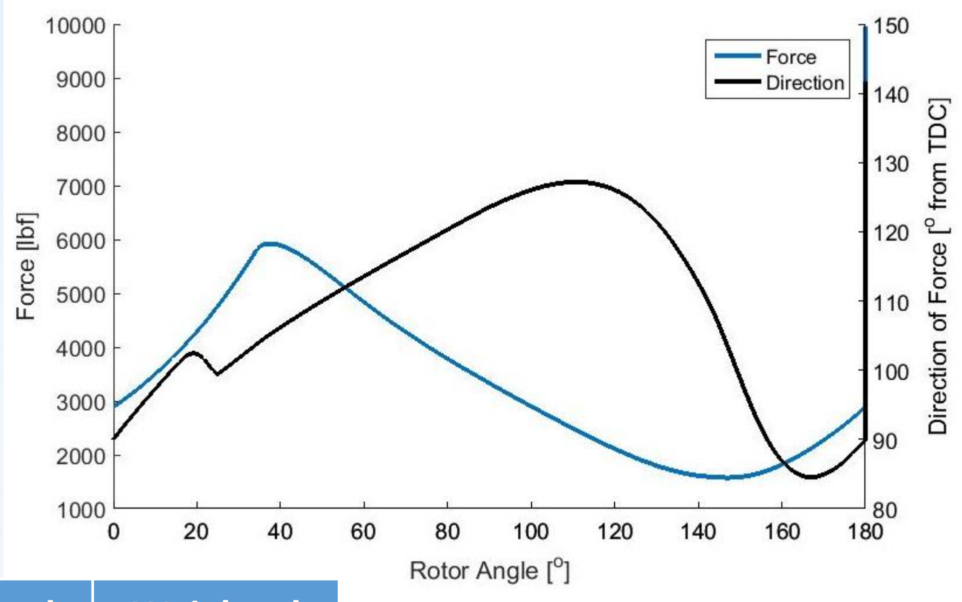




Drag of an Enclosed Disk from Daily and Nece, 1960

Bearing Loads

- Bearing loads calculated at various head pressures
- Calculated bearing



T_{evap}(C)	T_{cond}(C)	ARI Weight	Max Load (kN)	Weighted Load(kN)
3.33	53	0.01	33.9	0.34
3.33	43.3	0.42	27.8	11.7
3.33	33.9	0.45	25.1	11.3
3.33	28.3	0.12	24.5	2.93

40 Ton Compressor Simulated Bearing Loads at 38 F Evaporating and 110 F Condensing