

FLYWHEEL ENERGY STORAGE SYSTEM MECHANICAL STATOR DESIGN

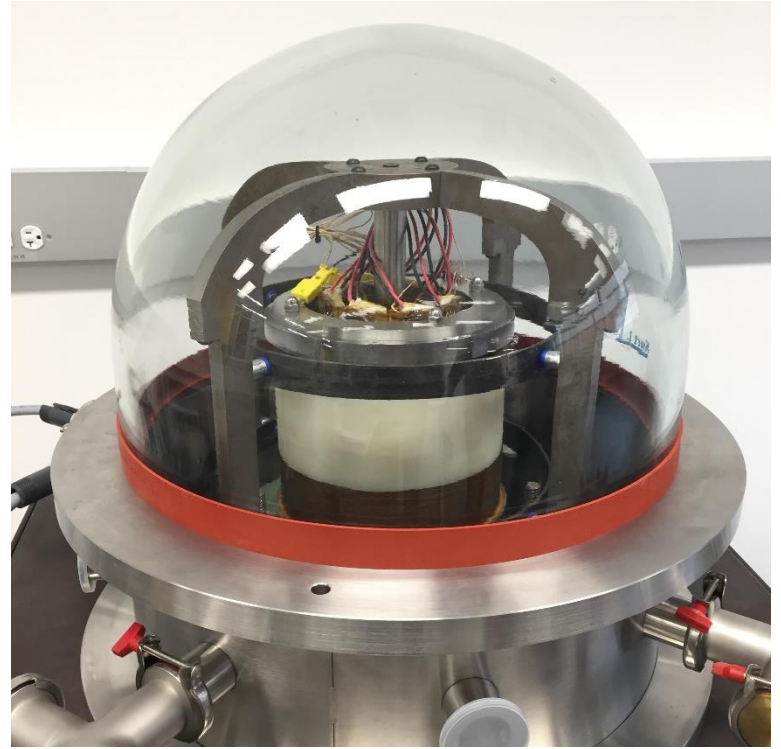


COOPER ATKINSON, NICHOLAS BACHUS, KYLE PETERSEN

University of Idaho
College of Engineering

UI FLYWHEEL

- Utilizes field regulated reluctance machine to accelerate and decelerate the rotor. No windings on rotor
- Friction losses are minimized by using magnetic levitation and containment in vacuum chamber



UI FLYWHEEL BACKGROUND

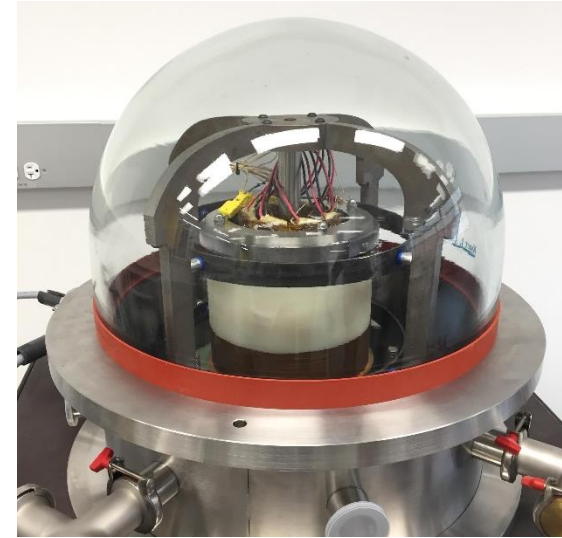
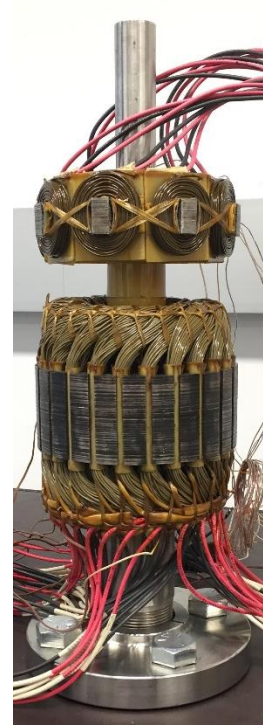
NASA Steckler Space Grant funded research project to develop a flywheel energy storage system for lunar and terrestrial use.

- Phase I – proof of concept
- Phase II – manufacturing of low speed system (1000 rpm)
- Phase III – design and manufacturing of high speed system (25000 – 30000 rpm)



PHASE III

- M.E. and E.E. Team – High Speed
 - Develop a modular analytical model for the high speed field reluctance machine stator
- C.S. Team – Low Speed
 - Design and test a control system for the safe acceleration and stabilization of the flywheel

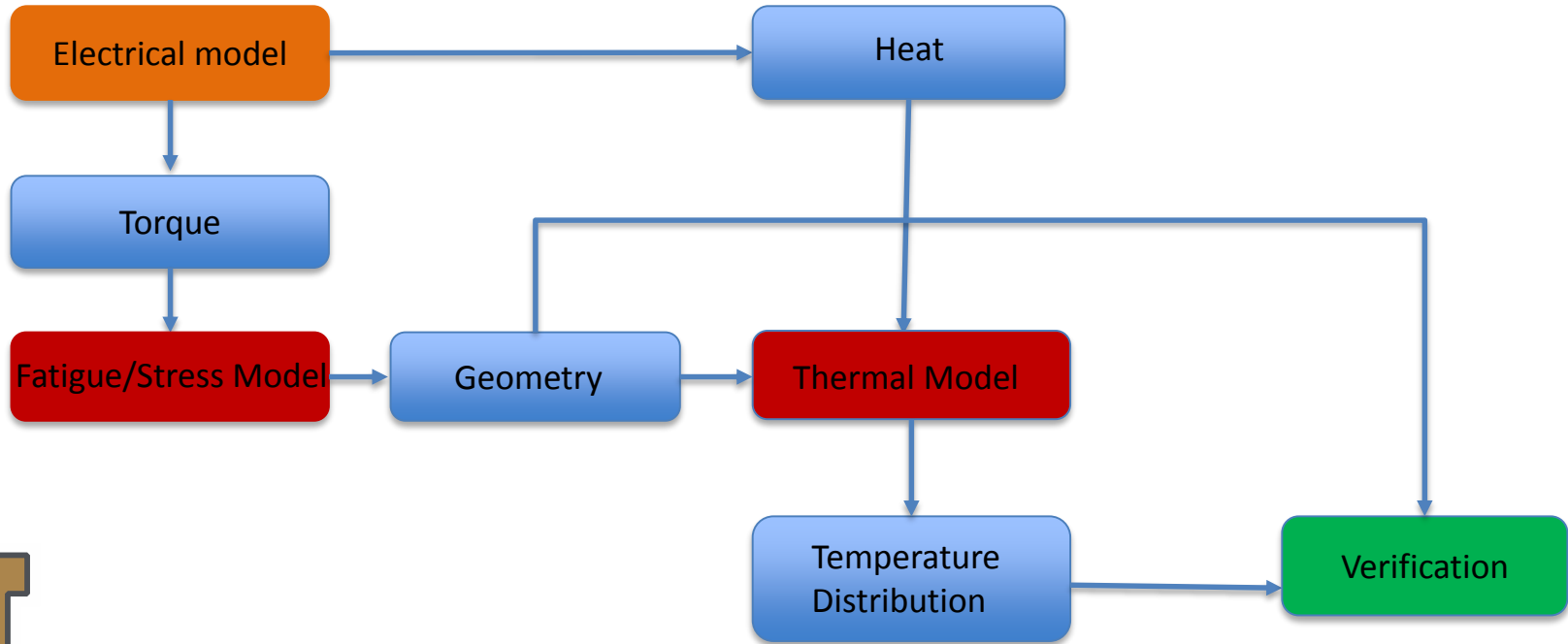


PROJECT

- Develop analytical modular model for the high speed flywheel stator
- Inputs – Torque and Heat
- Outputs – Geometry and Temperature Distribution



MODEL OUTLINE



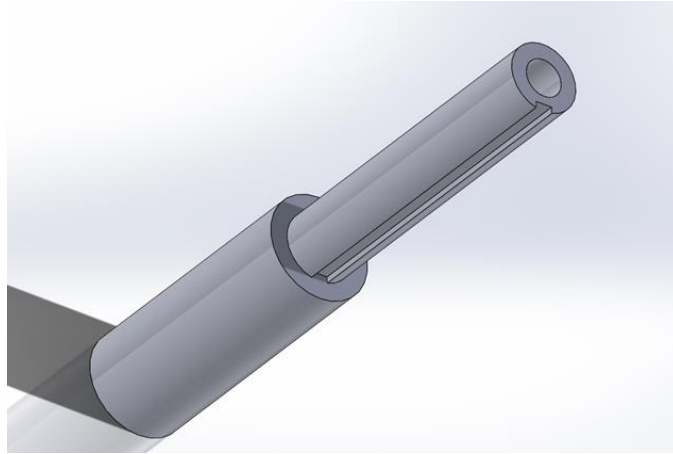
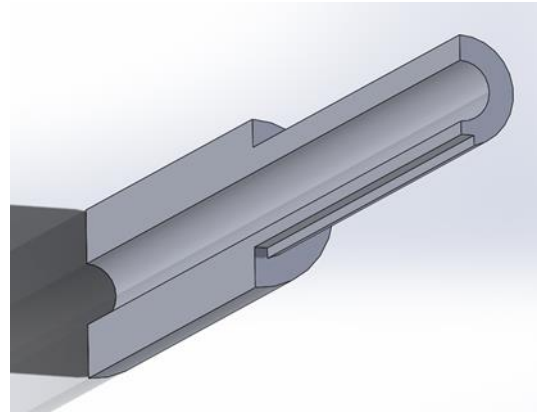
STRESS & FATIGUE ANALYSIS



FATIGUE ANALYSIS

Stator had the following important aspects

- Stepped Shaft
- Through-bore for coolant
- Keyway for windings
- Stainless steel shaft
- Iron laminations
- Torque loading



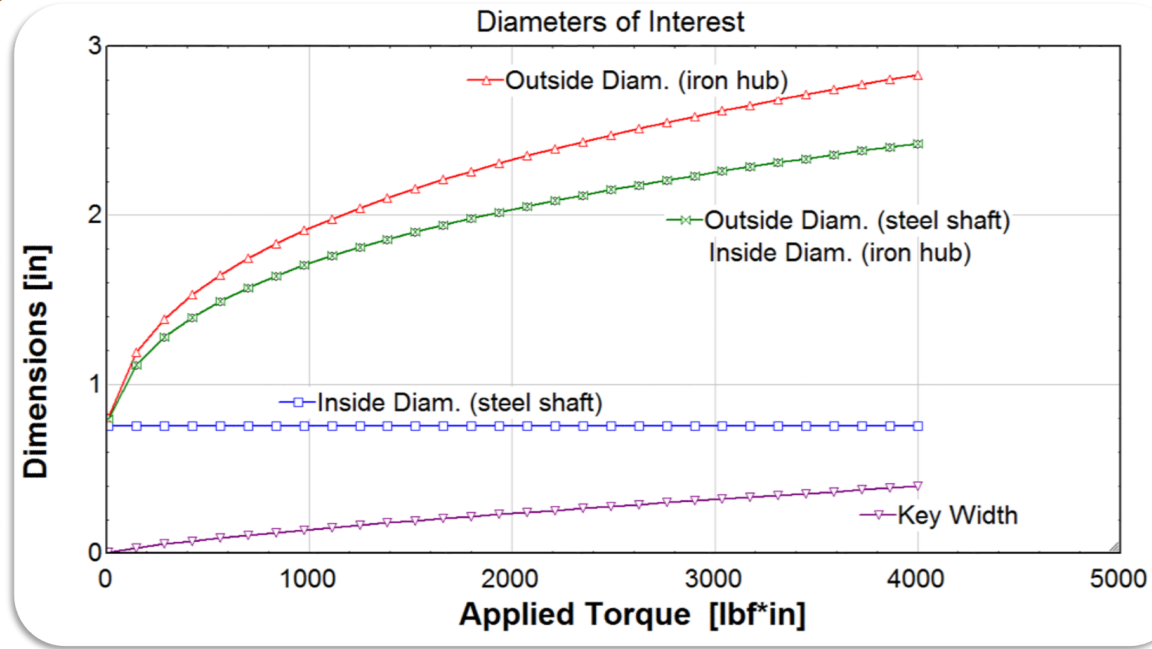
STRESS ANALYSIS

$$\text{Bending: } \sigma = \frac{My}{I} = \frac{\left(\frac{2T}{d_o}\right)\left(\frac{L}{2}\right)\left(\frac{d_o}{2}\right)}{\frac{\pi}{64}(d_o^4 - d_i^4)}$$

$$\text{Torsional Shear: } \tau = \frac{Tr}{J} = \frac{T\left(\frac{d_o}{2}\right)}{\frac{\pi}{32}(d_o^4 - d_i^4)}$$

$$\text{Principle Stress: } \sigma_e = \frac{\sigma_x + \sigma_y}{2} \pm \sqrt{\left(\frac{\sigma_x + \sigma_y}{2}\right)^2 + \tau_{xy}^2}$$

$$\text{Fatigue Limit: } S_e = \frac{(0.5)(S_u)(K_{size})(K_{surface})(K_{load})}{K_{notch}}$$



KEYWAY ANALYSIS

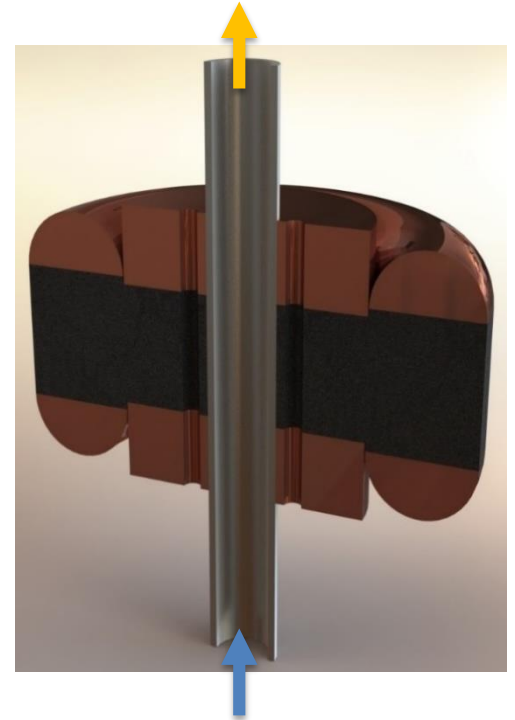
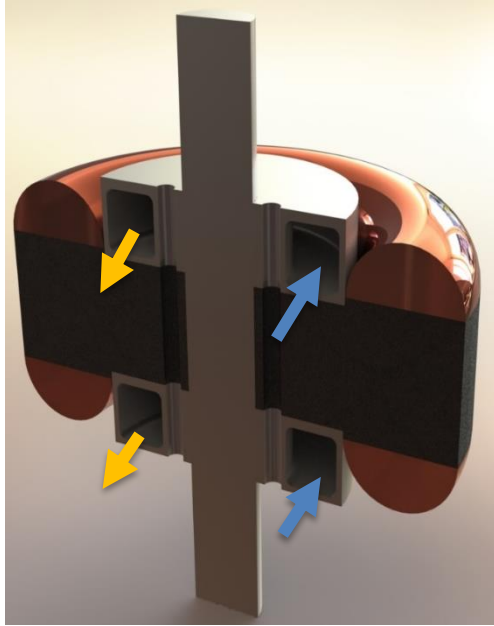
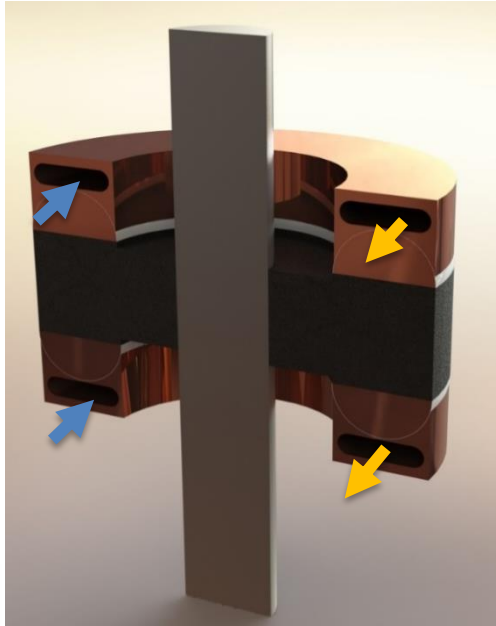
- Decided to use a keyway for attaching the iron stator laminations to the shaft
 - Easier assembly/disassembly than a press fit
 - Won't slip under high load
 - Could design key to be the point of failure to protect stator shaft and laminations



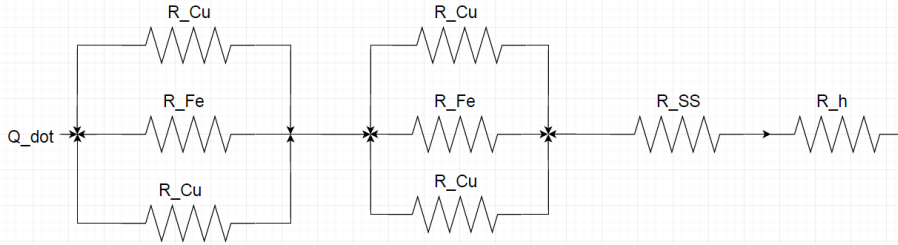
HEAT TRANSFER



EARLY DESIGNS



THERMAL CIRCUIT

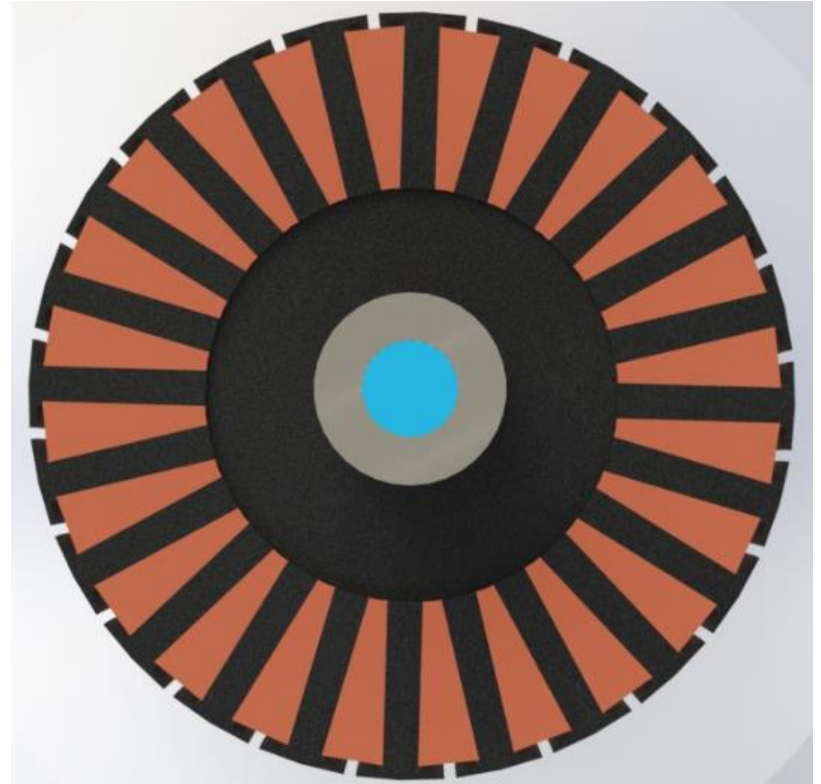


$$Q = \frac{\Delta T}{R}$$

$$R_{cond} = \frac{\ln\left(\frac{r_2}{r_1}\right)}{2\pi LK}$$

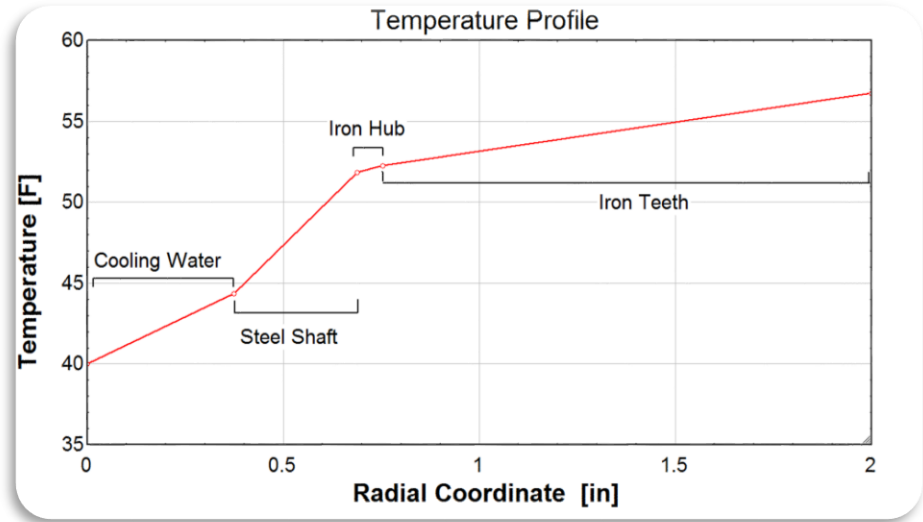
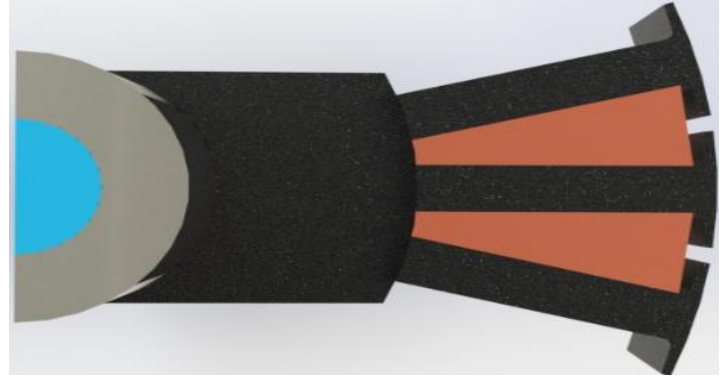
$$R_{conv} = \frac{1}{hA}$$

$$R_{total} = \frac{R_{cu}R_{fe}}{2R_{fe} + R_{cu}} + \frac{R_{cu}R_{fe}}{2R_{fe} + R_{cu}} + R_{ss} + \frac{1}{hA}$$

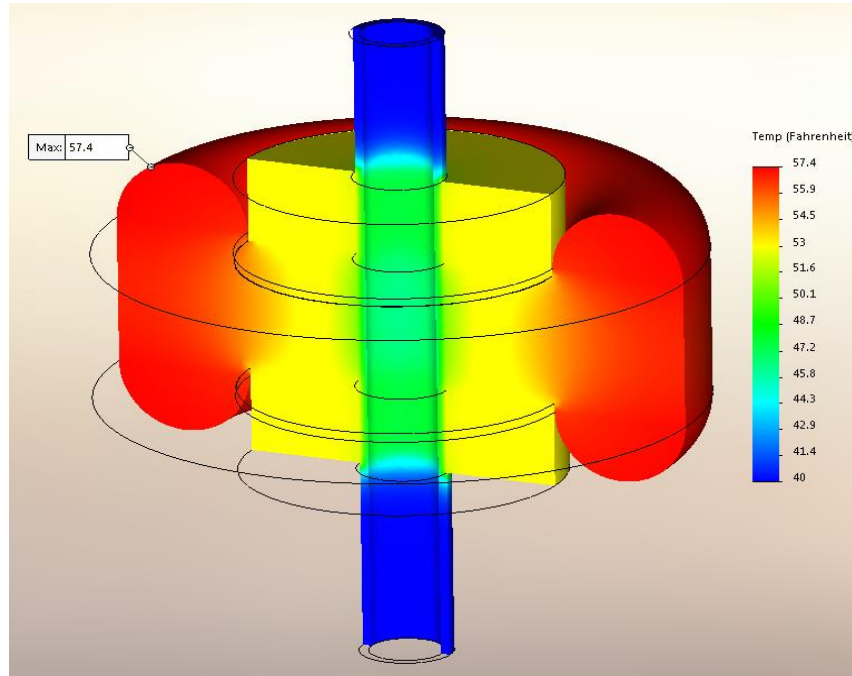


OUTPUTS

- Predicts the temperature distribution across the coolant, stator shaft, iron hub, and iron teeth
- Max temperature must be below 200°C (390°F) to ensure integrity of coil wire insulation
- Used independent Finite Element Analyses to verify thermal model



THERMAL VERIFICATION



FINAL DELIVERABLES

- Modular analytical model
 - Fatigue calculations for geometry to ensure long life (~2 million cycles)
 - Thermal performance of cooling system to minimize temperature distribution (Must operate below 200°C)
 - Thermal fatigue stress
 - Keyway dimensions
 - Base bolt pattern
- Detailed procedure for using model and verifying calculations.
- Machining suggestions for future teams



MOVING FORWARD

- Once rotor and field regulated reluctance machine parameters are determined, the analytical model will be used to determine stator geometry
- Develop manufacturing plan
- High speed system can then be built



QUESTIONS?



THERMAL FATIGUE

Thermal Strain

$$\varepsilon = \alpha \Delta T$$

“Seen” Strain

$$\varepsilon_{seen} = \varepsilon_{steel} - \varepsilon_{iron}$$

“Seen” Strain to radial stress

$$\varepsilon_{seen} = \frac{\sigma_r}{E_{material}}$$

Hoop stress outside Steel Surface

$$\sigma_h = \frac{2(r_o^2)\sigma_r}{(r_o - r_i)^2}$$

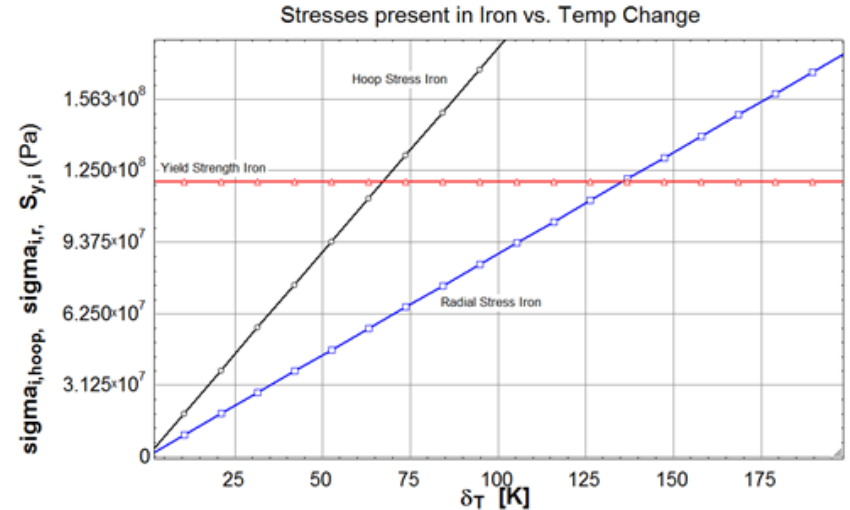
Hoop stress inside iron surface

$$\sigma_h = \frac{\sigma_r(r_o + r_i)^2}{(r_o - r_i)^2}$$

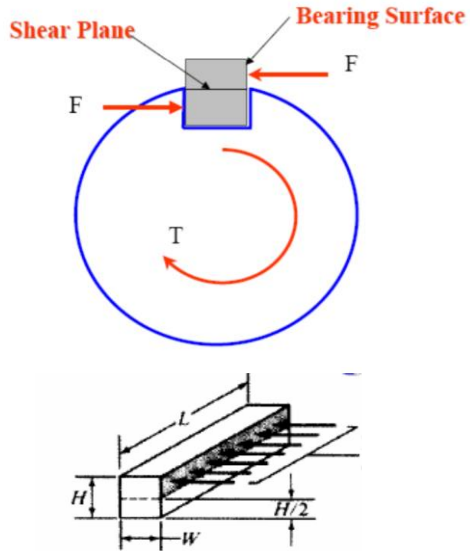


THERMAL FATIGUE RESULTS

- Increase in temperature => Increase in stress
- Increase in diameter => Decrease in hoop stress
- Tolerances affect calculations
 - Thermal expansion on order of 1/1000ths of inches



KEYWAY ANALYSIS



Minimum key
length - shear

$$L_s = \frac{4TF_s}{S_y DW}$$

Minimum key
length - bearing

$$L_s = \frac{4TF_s}{KS_y DH}$$

Where

$$1.0 < K < 1.5$$

Shear Stress in Key

$$\tau_{avg} = \frac{2T}{DWL} = \frac{0.5S_y}{F_s}$$

For shaft calculations,
assume stress
concentration factors:

Bending

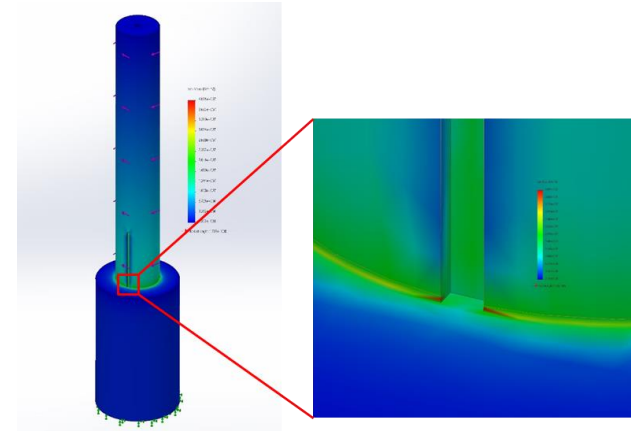
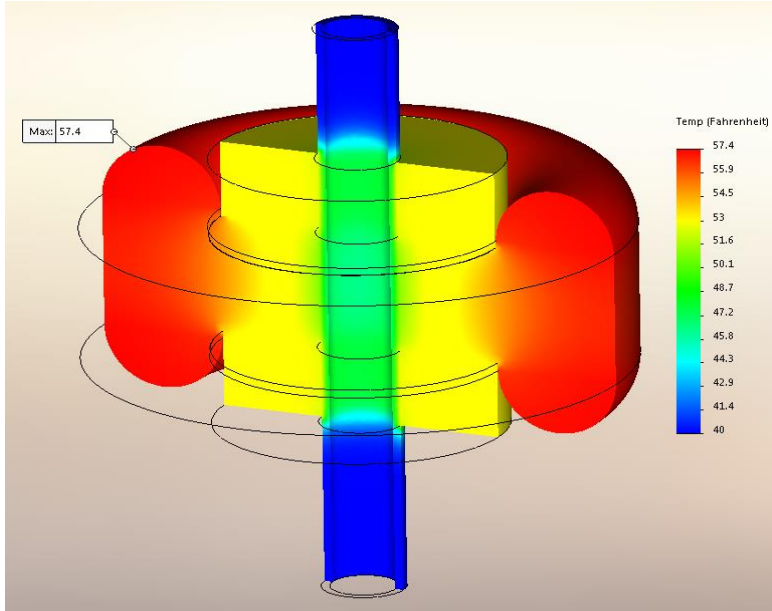
$$K_t = 2.2$$

Torsion

$$K_{ts} = 2.6$$



VERIFICATION SIMULATIONS



Inputs

Torque (Nm)	Torque (lbf*in)	cycles
11.29848293	100	$2 \cdot 10^6$
11.29848293	100	$2 \cdot 10^6$
56.49241467	500	$2 \cdot 10^6$
56.49241467	500	$2 \cdot 10^6$
84.738622	750	$2 \cdot 10^6$
84.738622	750	$2 \cdot 10^6$
225.9696587	2000	$2 \cdot 10^6$
225.9696587	2000	$2 \cdot 10^6$

Verification

Fatigue Limit (MPa)	Max Stress in Shaft (MPa)
53.05	16.88
53.05	16.89
53.05	32.77
53.05	36.17
53.05	40.31
53.05	41.33
53.05	57.4
53.05	49.83

