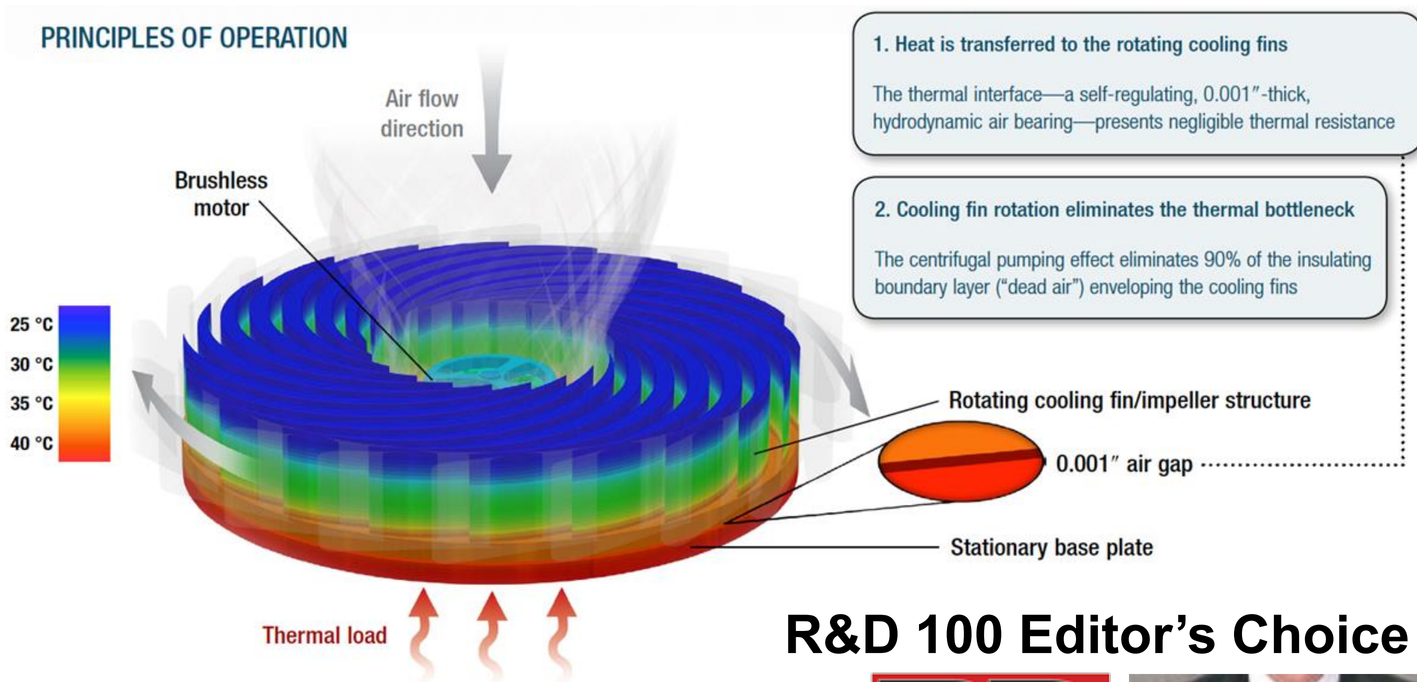
SAND2013-10015P

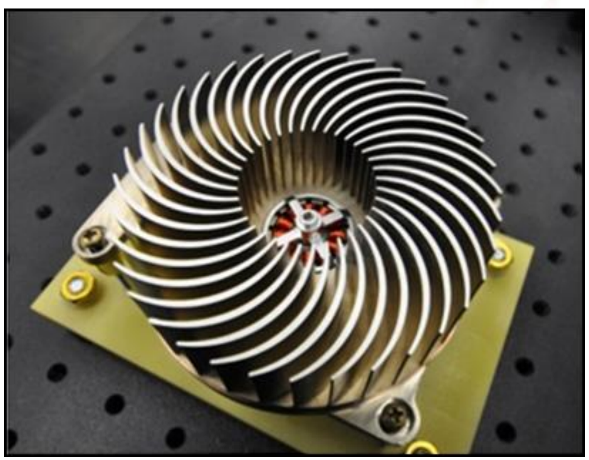
Combustion Research Facility

Sandia Cooler Engineering

Program Overview

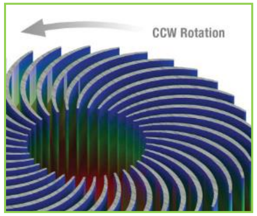
The Sandia Cooler is a breakthrough in air-cooled HX originally conceived for electronics cooling



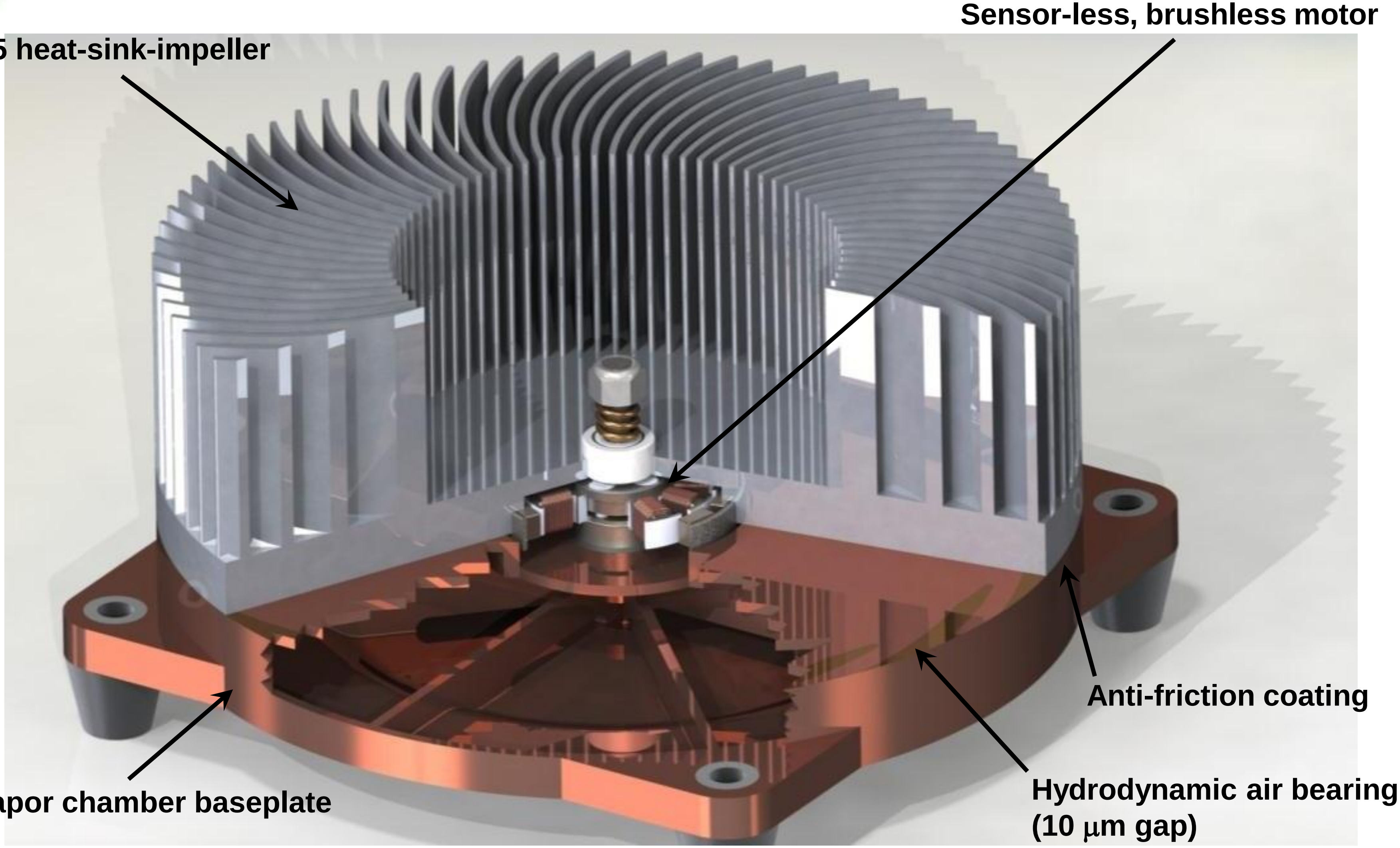


R&D 100 Editor's Choice Winner

2012 R&D 100 50th ANNIVERSARY



Latest CPU cooler design represents the culmination of several years of development



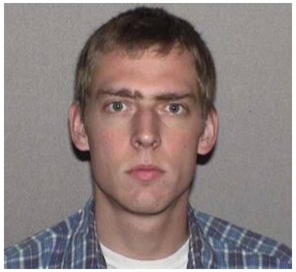
V5 objective: fully matured radial air bearing heat exchanger technology, tech transfer ready

V5 performance goals: R = 0.1 C/W at 3000 rpm, very low noise, 5 W power consumption

The team includes 14 scientists, engineers and technologists

Dr. Imane Khalil
Project Manager

Mark Zimmerman
BSEE: Motor control development



Mike Leick
MSME: Motor control and anti-friction coatings

Dr. Jeff Koplow
PhD Chem: Inventor, technical advisor, axial flow R&D lead

Dr. Wayne Staats
PhD ME: CFD for radial and axial flow impeller design

Ryan Gorman
EE: Motor control development

Nathan Spencer
MSME: Structural dynamics

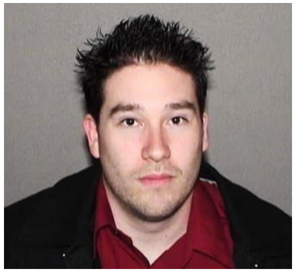
Justin Vanness
MSME: Motor control development

Terry Johnson
MSME: Radial flow project lead and system engineer

Kent Smith
Mech. Tech.: Fabrication and mechanical design

Dr. Patricia Gharaghozloo
PhD ME: CFD/Heat Transfer

Dr. Marco Arienti
PhD ME: CFD/Heat Transfer



Daniel Matthew
BSME: Impeller fabrication and mechanical design

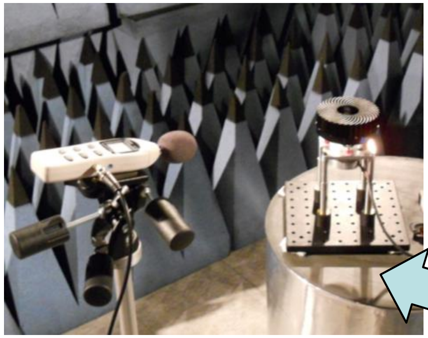
Dr. Ethan Hecht
PhD ChE: Impeller performance characterization

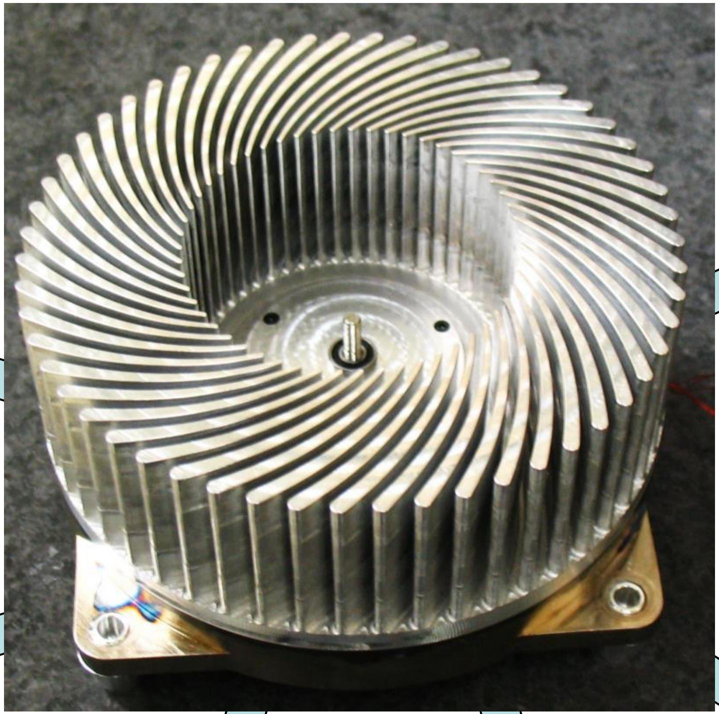
Dr. Arthur Kariya
PhD ME: Heat pipe design

EXPERIMENTAL PERFORMANCE CHARACTERIZATION

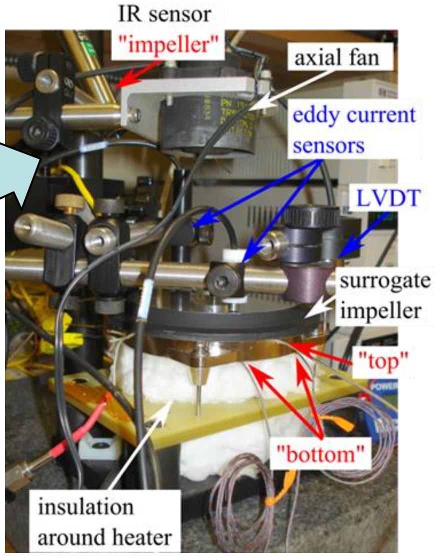
Test stands have been developed to evaluate all aspects of the Sandia Cooler

Acoustic

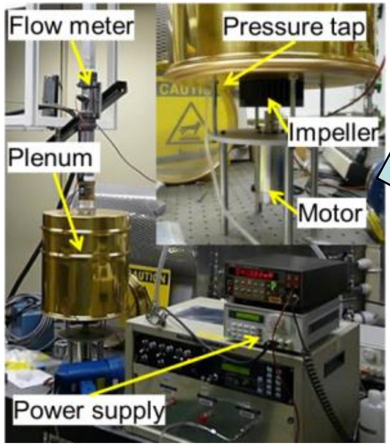




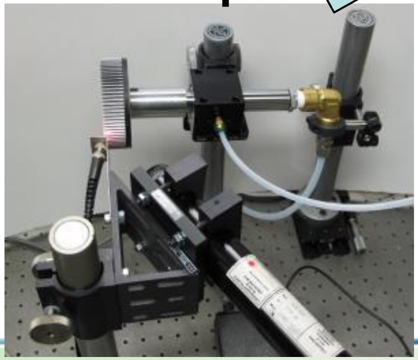
Thermal Resistance



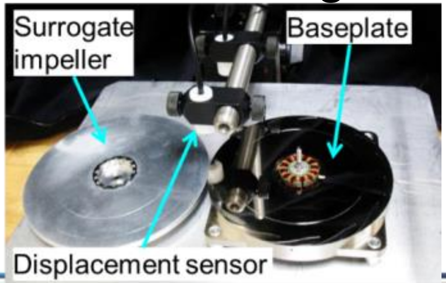
Pressure-Flow

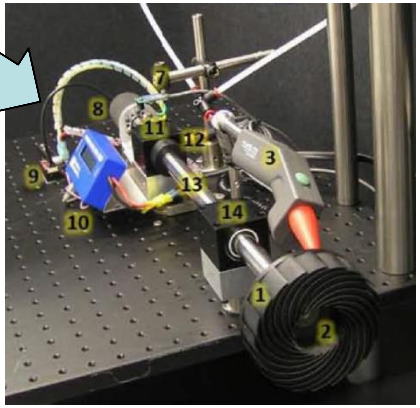


Torque



Air Bearing





Three different impeller geometries have been extensively characterized

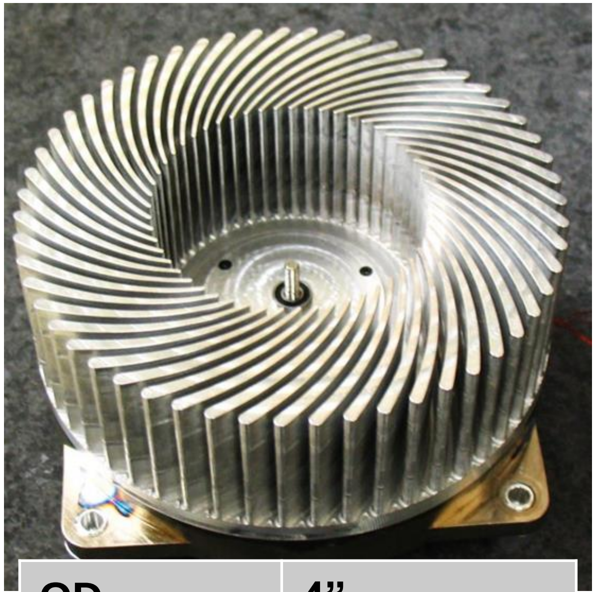
V4

OD	4.0”
ID	1.5”
Fin Height	1.0”
# Fins	36
Shape	Intersecting arcs
Fin Width	

V5

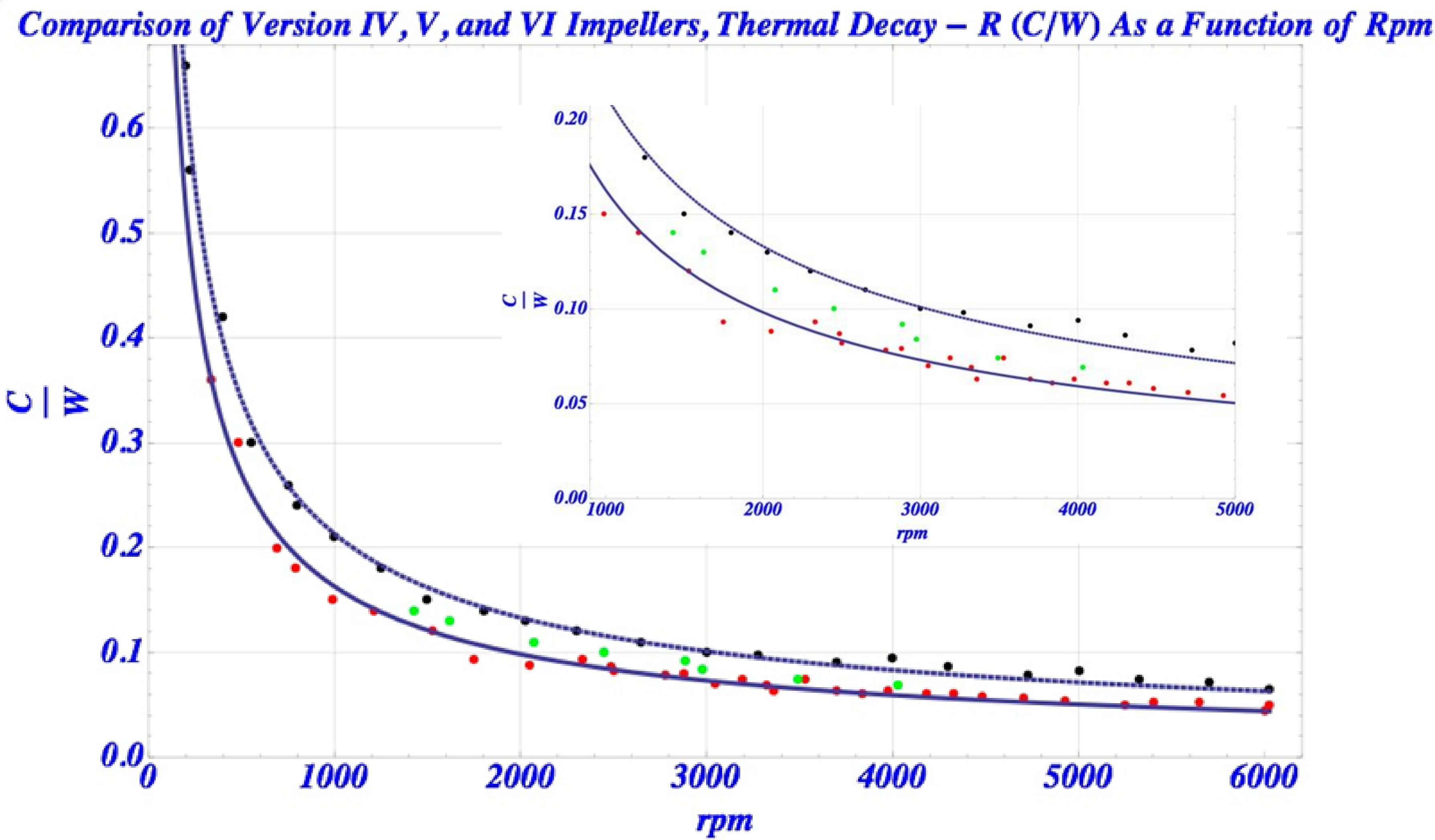
OD	4.0”
ID	2.0”
Fin Height	0.95”
# Fins	80
Shape	Arcs
Fin Width	0.030” uniform

V6

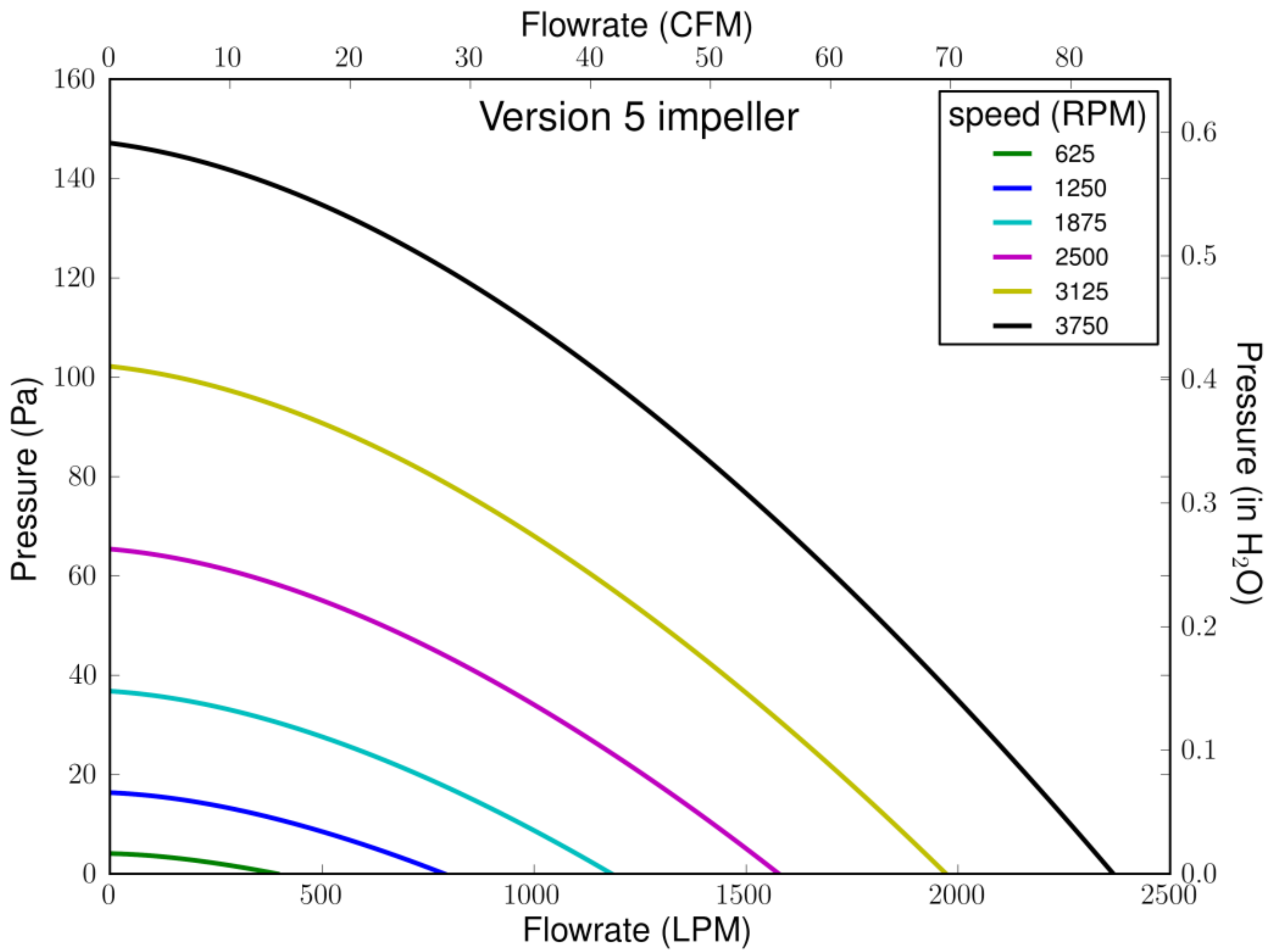


OD	4”
ID	2.0”
Fin Height	1.18”
# Fins	55
Shape	Log spiral
Fin Width	

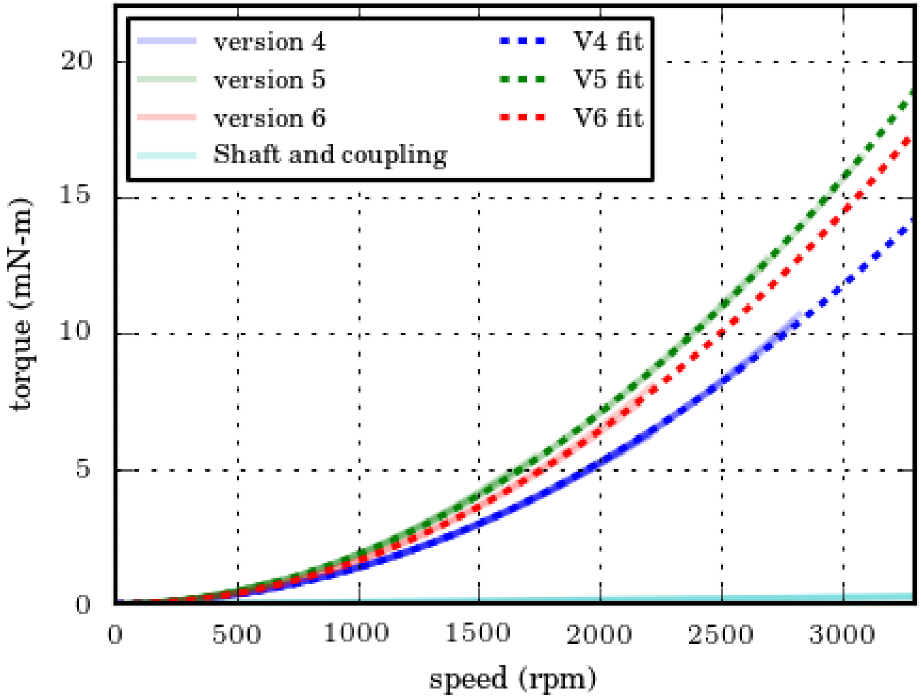
V5 impeller has the lowest thermal resistance tested



Pressure-flow curves were measured for several 4” impellers; V5 performed best



Impeller torque measured vs. speed; power consumption includes impeller and air gap torque



Impeller power:

$$P = \tau \times \omega$$

@2500 rpm V5 P = 3W

Air gap power:

$$P = \tau \times \omega$$

$$\tau = \frac{\pi * \mu * \omega \left(r_o{}^4 - r_i{}^4\right)}{2 * h}$$

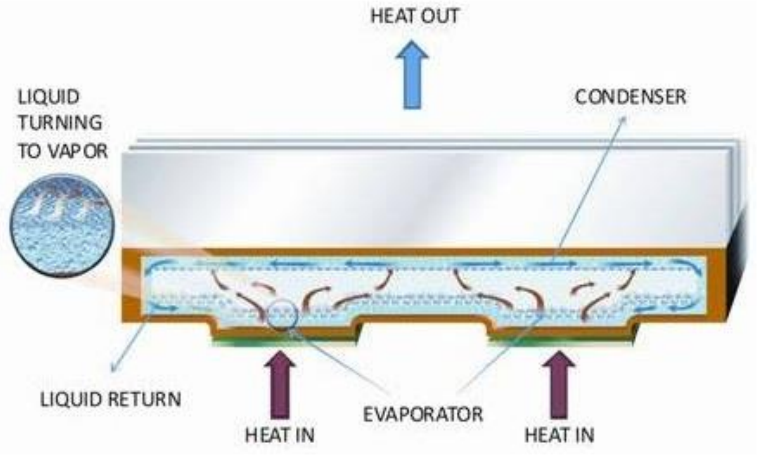
For a 10 micron air gap P = 1.3W

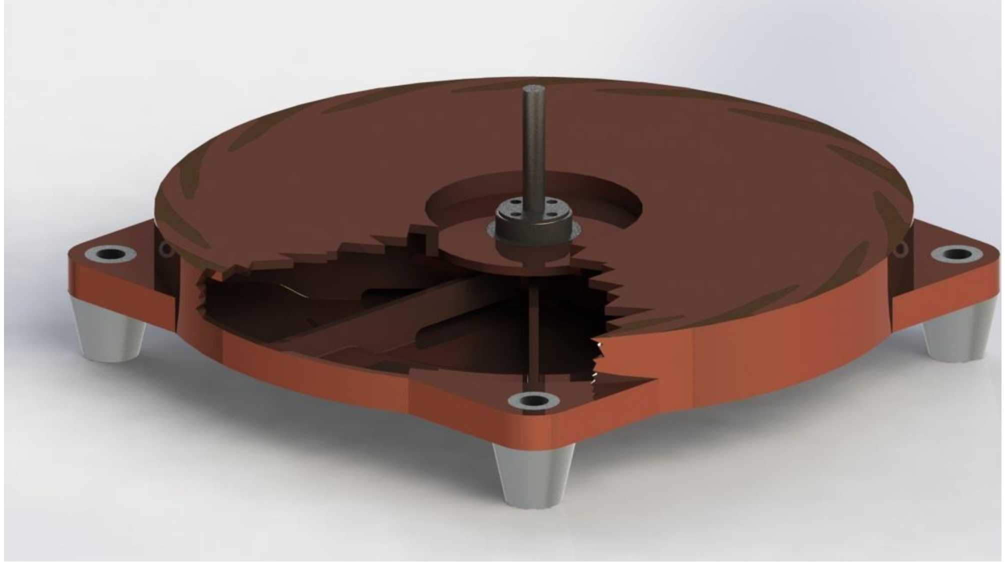
Total power:

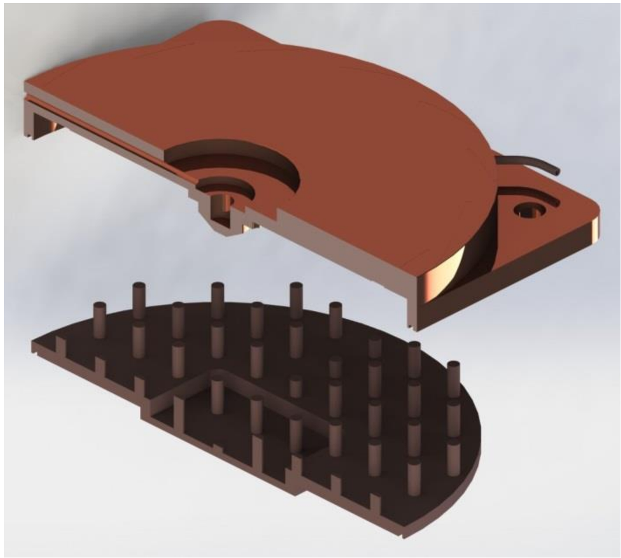
P_{mech} = 3W+1.3W = **4.3W**

Baseplate: Vapor Chamber Incorporation

- Spreading resistance of solid baseplate was unacceptably high
- Vapor chamber solution from Thermacore



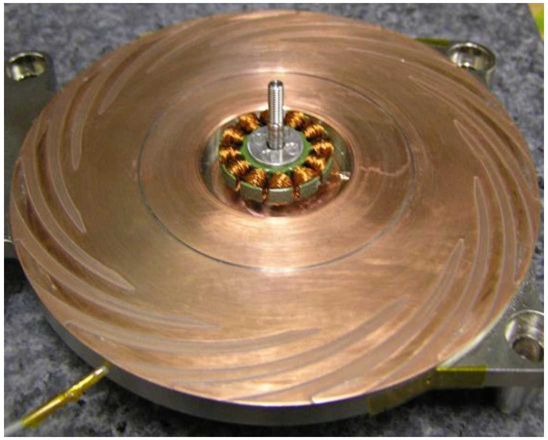




Air bearing design was improved through experiment and analysis

Original Design

Greater lift than needed
Significant pre-load for 10 μm gap
Groove area and depth larger than required



Parameters	
Ø_Impeller	101.6 mm
Groove Depth	81 μm
λ, r_Inner/r_Outer	0.75
α, Groove Angle	15°
k, # of Grooves	15
g, ridge width/groove width	1.0

V5 Design

Good stiffness with less thermal resistance
Less sensitivity to impeller speed
Groove area still larger than required

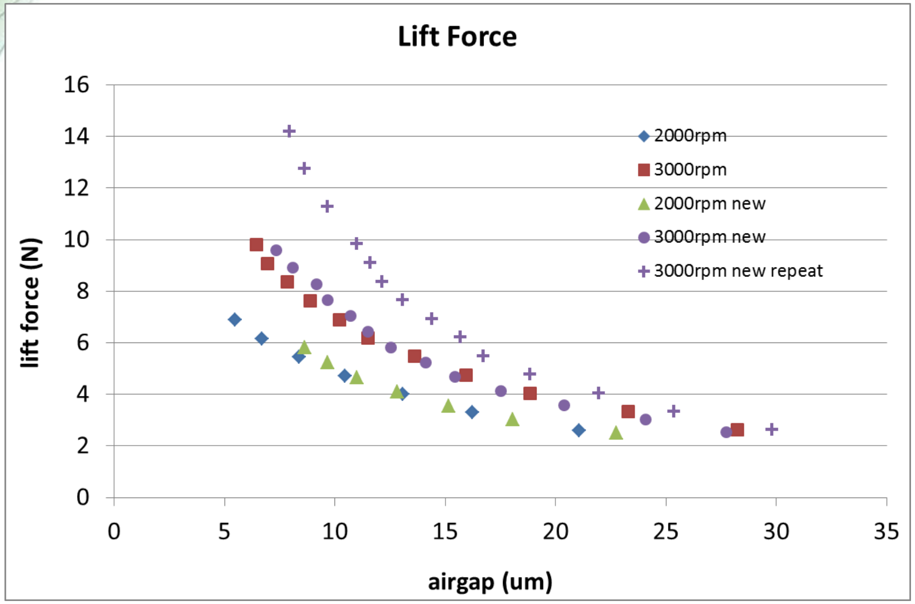
Parameters	
Ø_Impeller	101.6 mm
Groove Depth	25 μm
λ, r_Inner/r_Outer	0.9
α, Groove Angle	15°
k, # of Grooves	15
g, ridge width/groove width	1.0

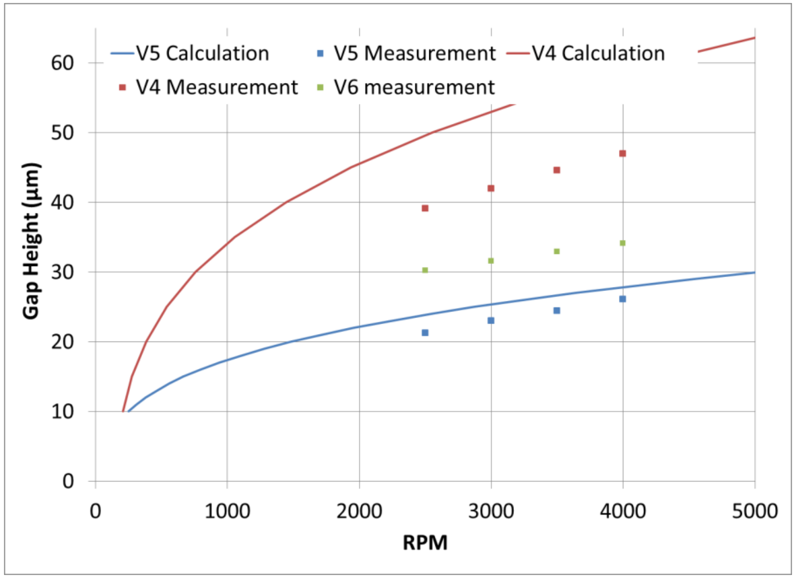
Final Design

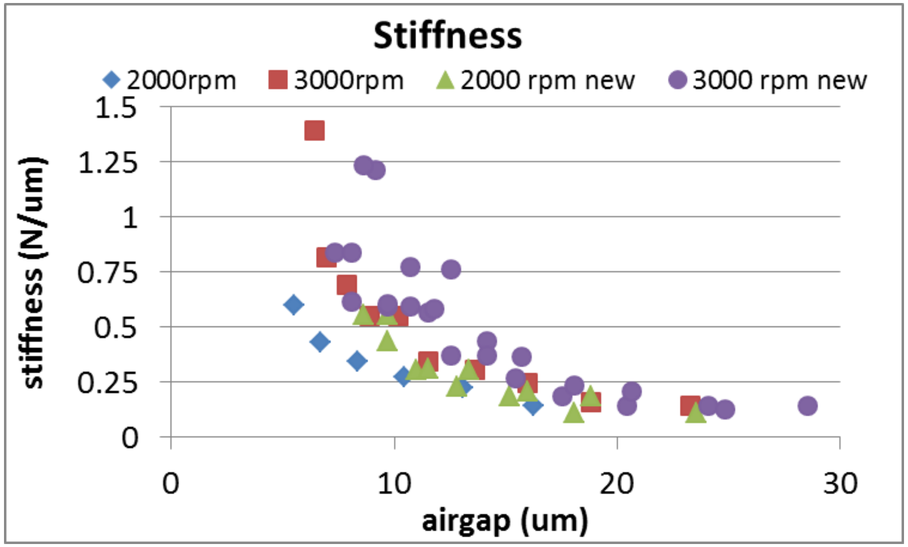
Maximum stiffness at a 10 μm gap
Minimal pre-load
Minimum thermal resistance

Parameters	
Ø_Impeller	101.6 mm
Groove Depth	35 μm
λ, r_Inner/r_Outer	0.9
α, Groove Angle	12°
k, # of Grooves	15
g, ridge width/groove width	1.4

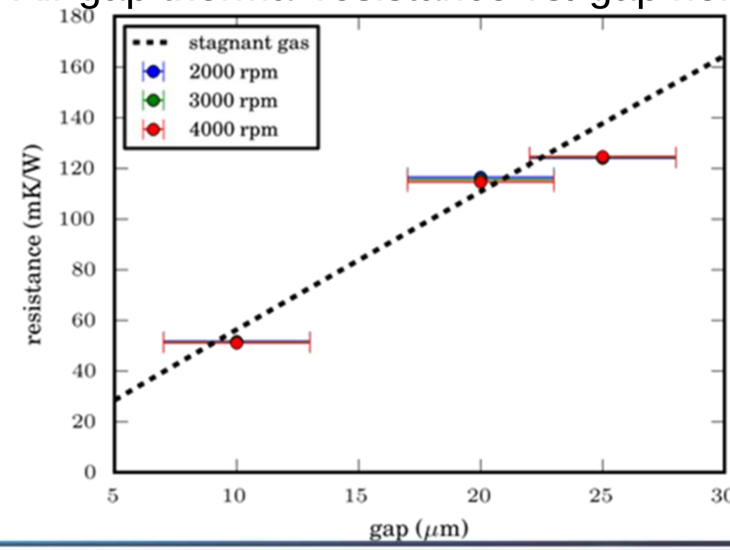
Air bearing provides stiff, low friction interface but thermal resistance is significant



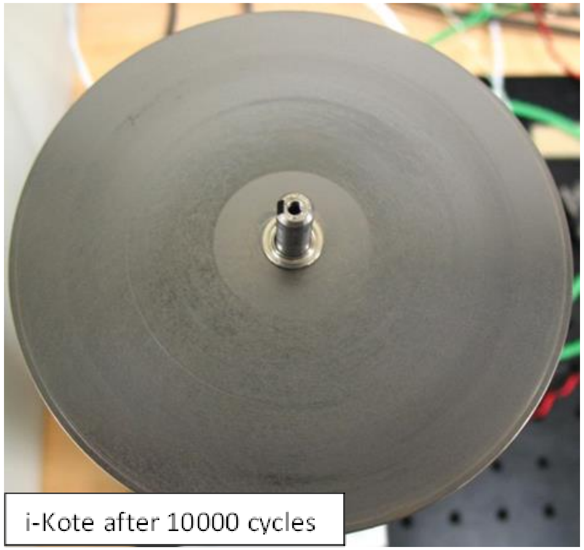


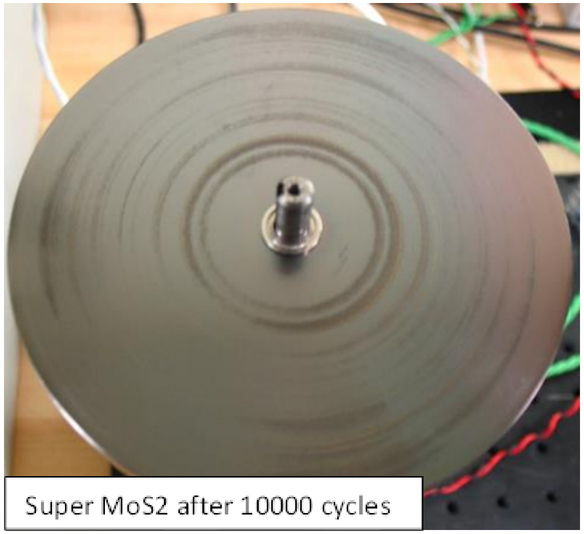


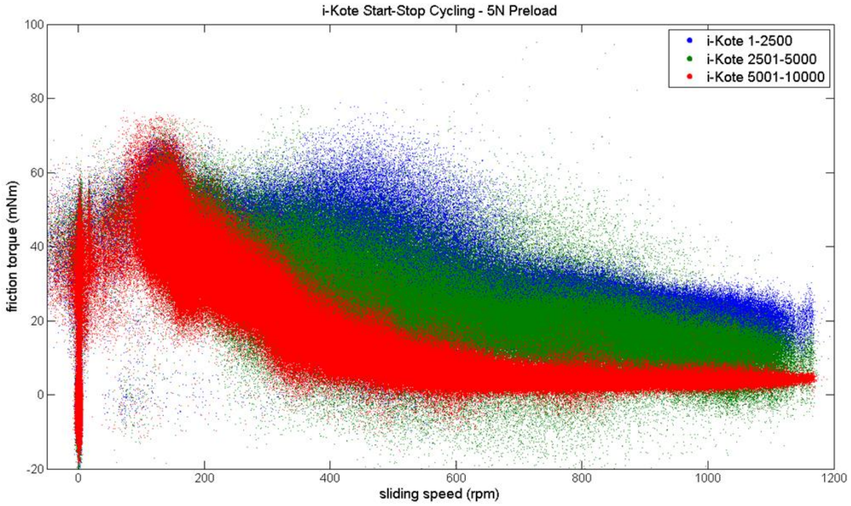
Air gap thermal resistance vs. gap height

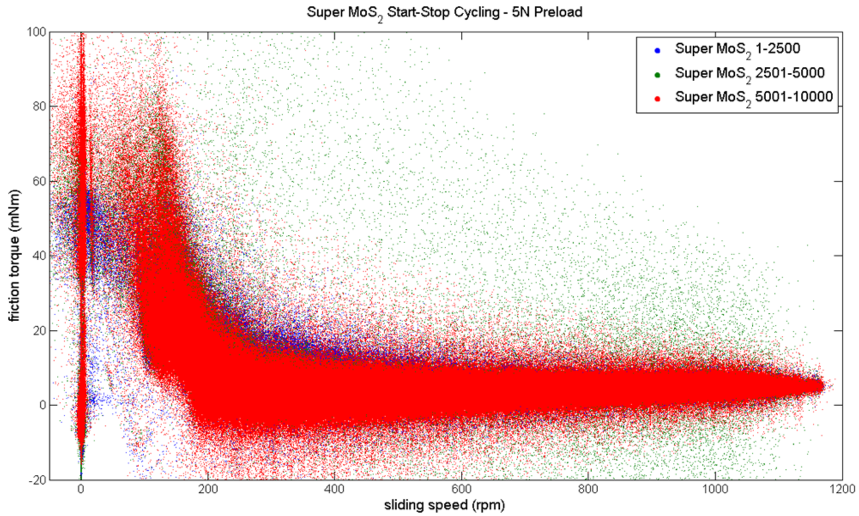


Two anti-friction coatings perform well out to 15,000 start/stop cycles


i-Kote after 10000 cycles

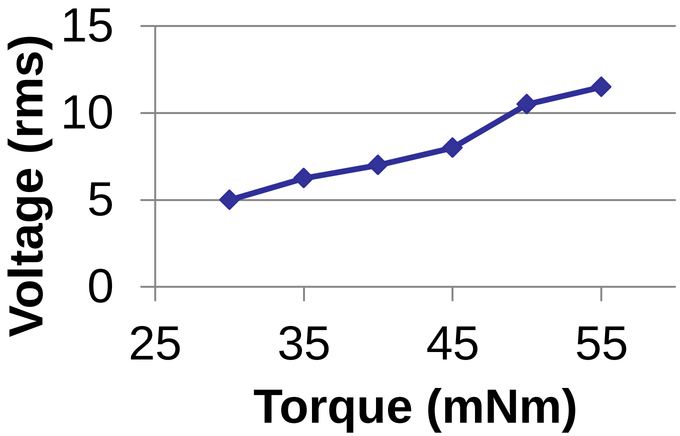

Super MoS2 after 10000 cycles

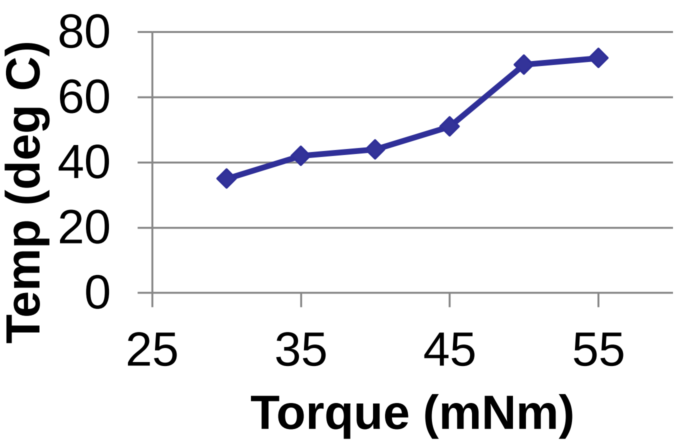


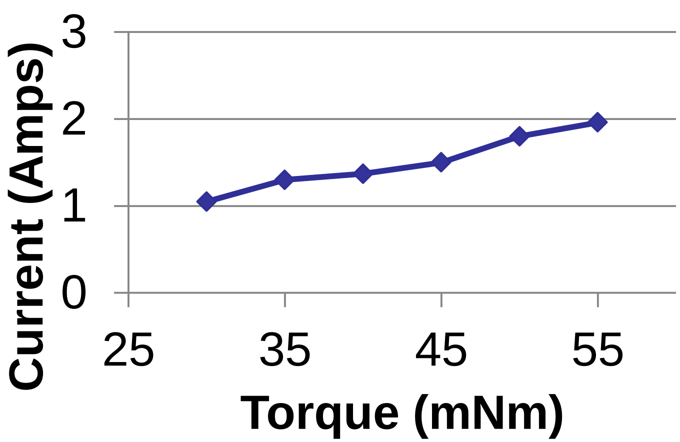


Motor can overcome start-up torque with reduced contact area

- Motor can produce up to 55 mNm
  - potential higher but experienced voltage saturation from amplifiers
- 3-phase motor with 34 gauge windings ramped from 0 to 300 rpm in 1.5 seconds

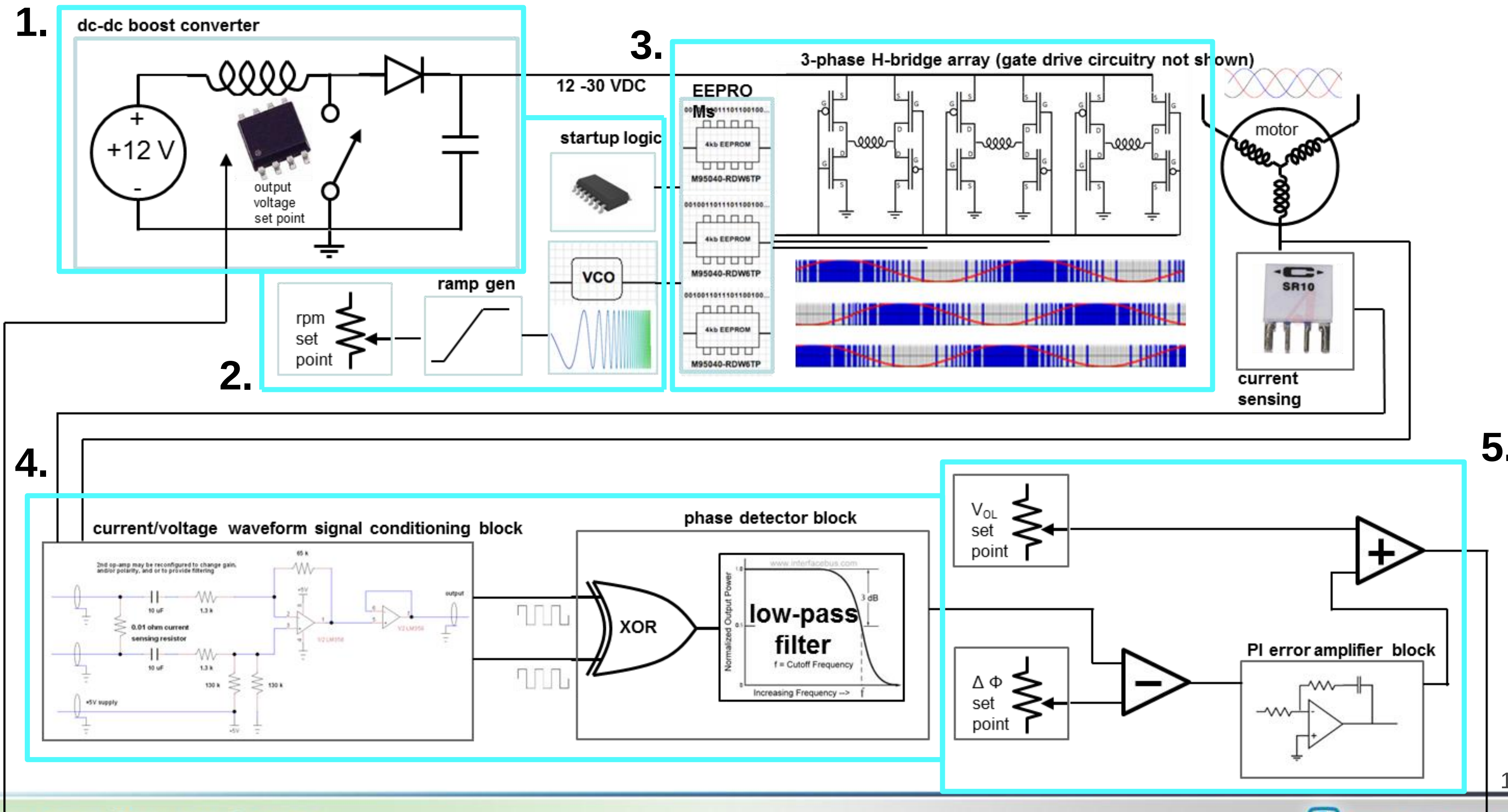






Custom motor controller in final stages of development

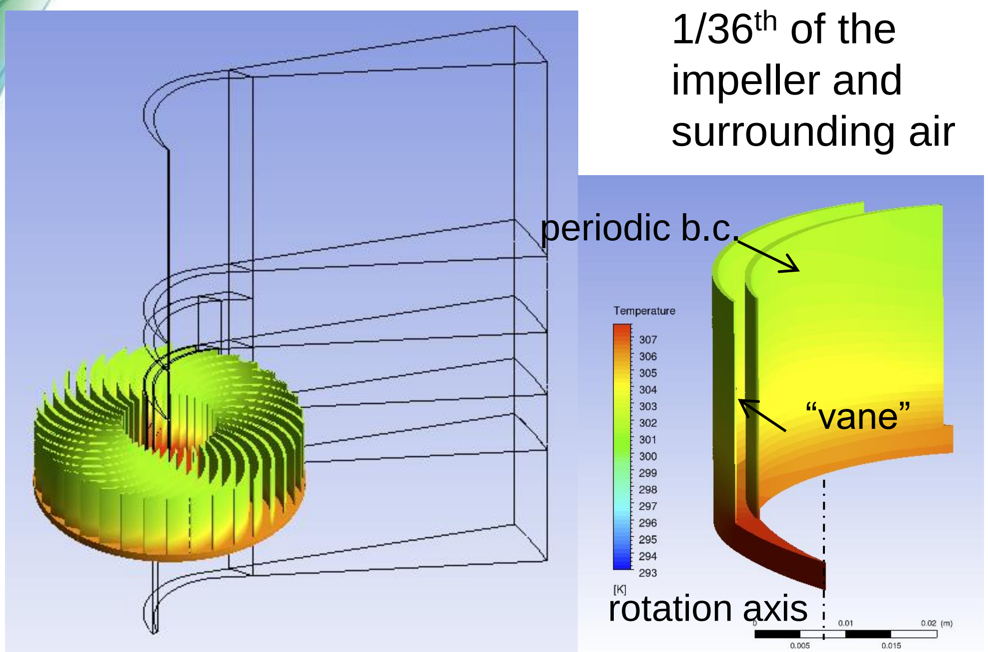
Five primary blocks: 1. Boost converter – prototyping complete, 2. VCO – final tuning for dynamic range complete, 3. H-Bridge – final tuning for efficiency and min heat loss complete, 4. Phase Detect – prototyping complete, 5. PI Loop – ongoing development



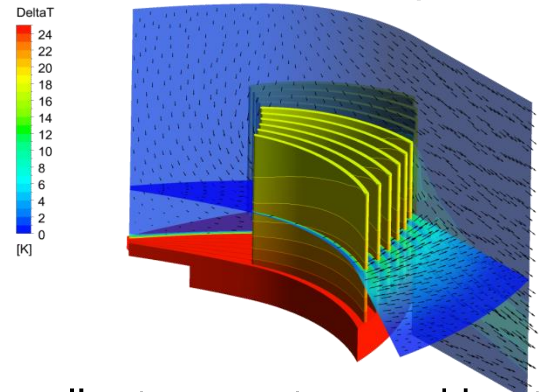
MODELING AND ANALYSIS

Computational fluid dynamics (CFD) models tell us a lot about the cooler performance

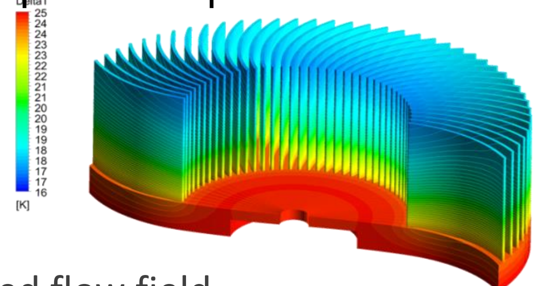
Example: V4 with 36 blades



Flow field and air temperature



Impeller temperature and heat flux

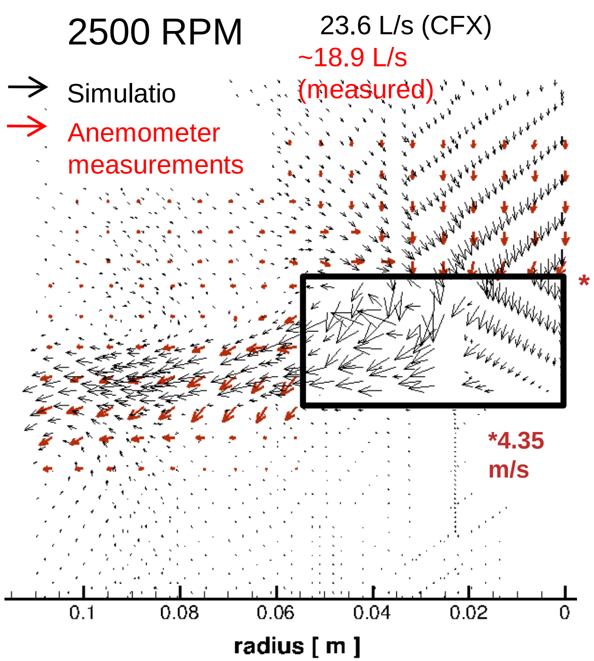


- ANSYS CFX V14.0
- Conjugate heat transfer (solid and fluid computation)
- Rotational reference frame for impeller
- Periodic boundary conditions take advantage of symmetry
- Reynolds-Averaged Navier Stokes (RANS) equations for flow field
  - Shear Stress Transport model

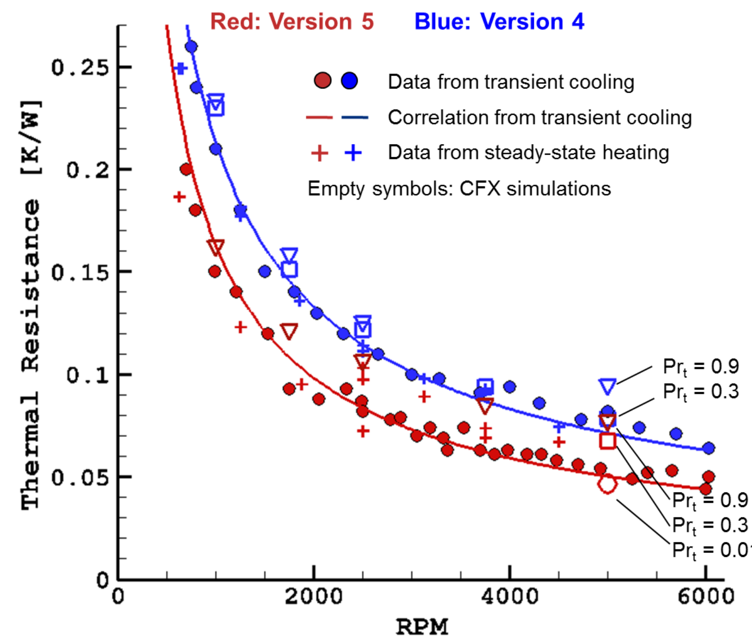
- Detailed flow field
- Temperature distribution in air
- Torque and power consumption
- Heat transfer coefficient
- Temperature distribution within solid regions
- Fin efficiency
- Where solid material is efficiently being used

CFD models have been experimentally validated

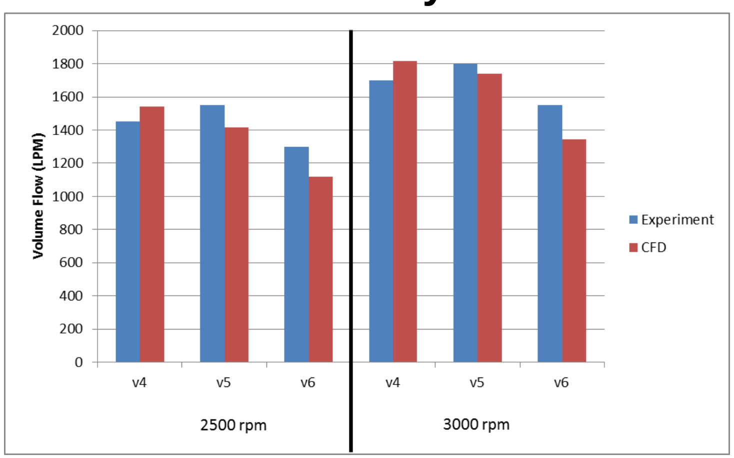
Flow Field



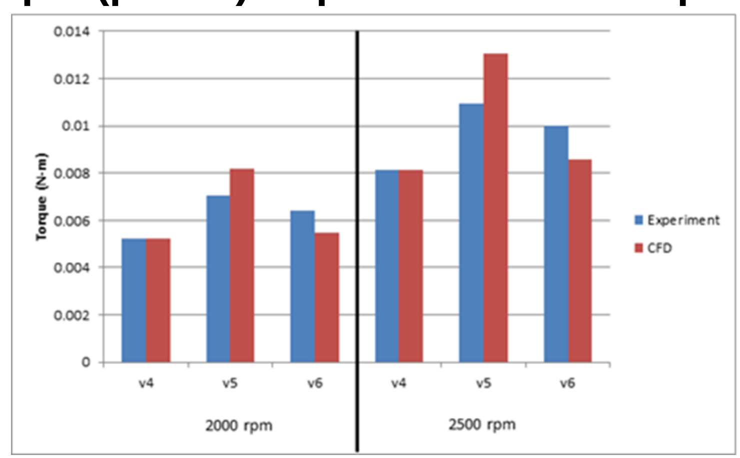
Thermal Resistance



Free delivery rates



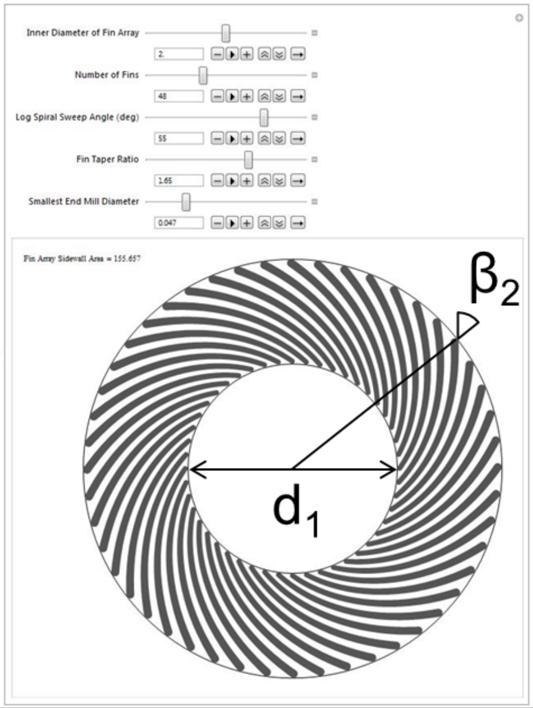
Torque (power) required to rotate impellers



CFD and design models have been used to carry out impeller parameter and scaling studies

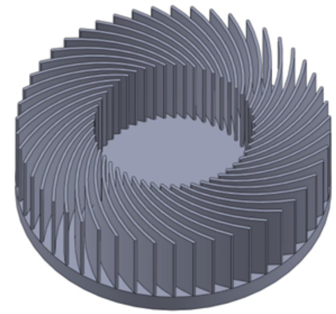
1. Generate Equations

Preliminary information (e.g. surface area) determined from Mathematica model



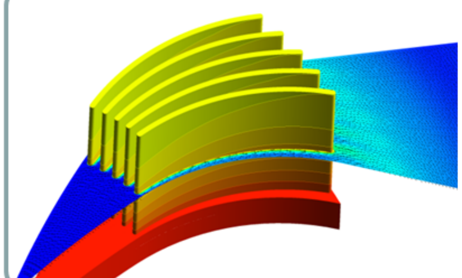
1. Inner diameter (d_1)
2. Blade angle (β)
3. Number of fins (n)
4. Minimum endmill diameter (d_e)
5. Fin Taper Rate (power law dependence of blade width on radius)

2. Create Geometry



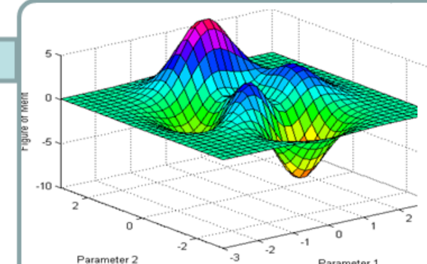
3D geometry generated from vector equations in SolidWorks

3. Simulate



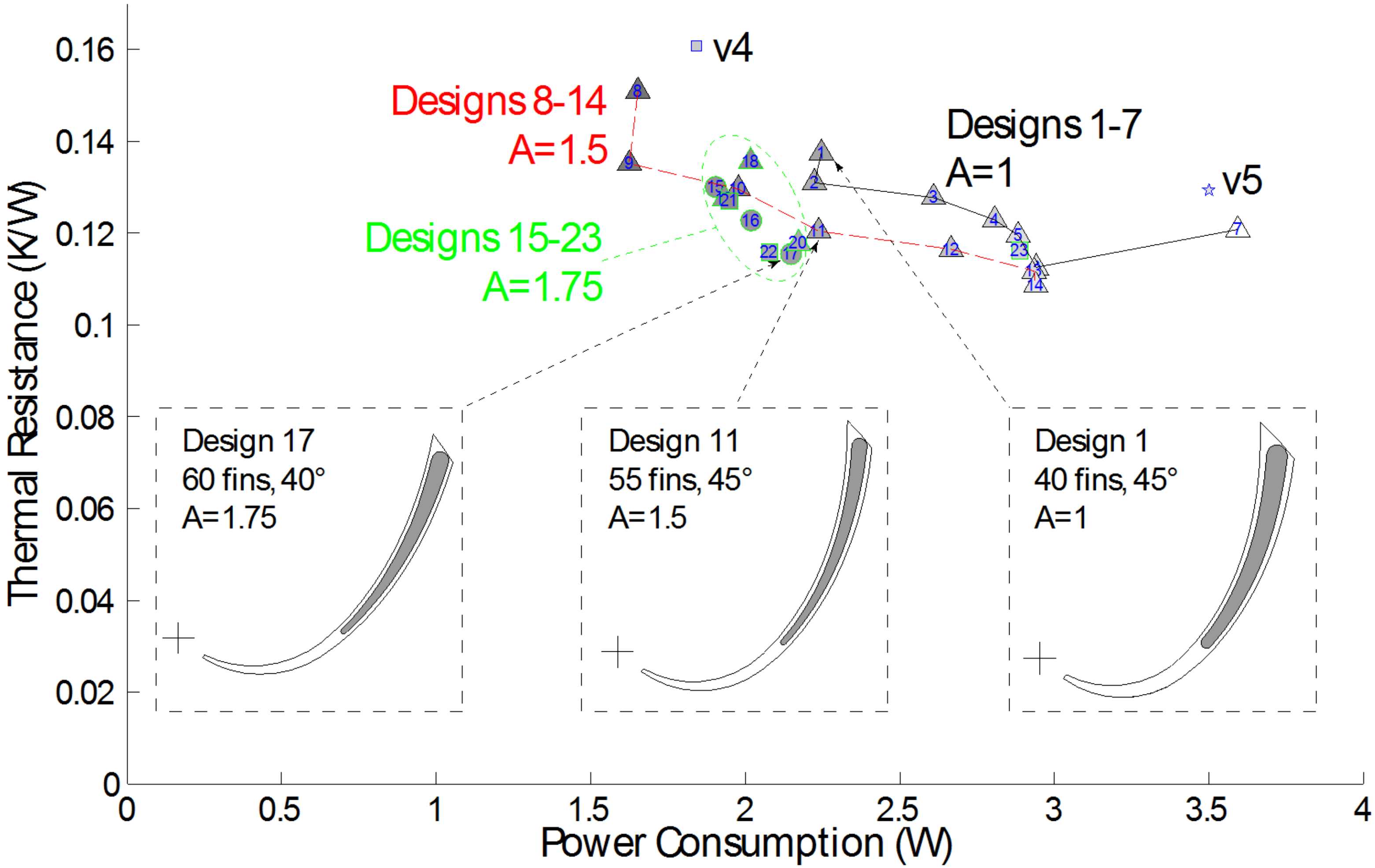
Flow field and conjugate heat transfer simulated in ANSYS CFX

4. Evaluate



Determine sensitivity of thermal resistance to input parameters

40 different permutations of the impeller geometry were modeled to find an improved design



Initial scaling study shows thermal resistance vs. motor power tradeoffs

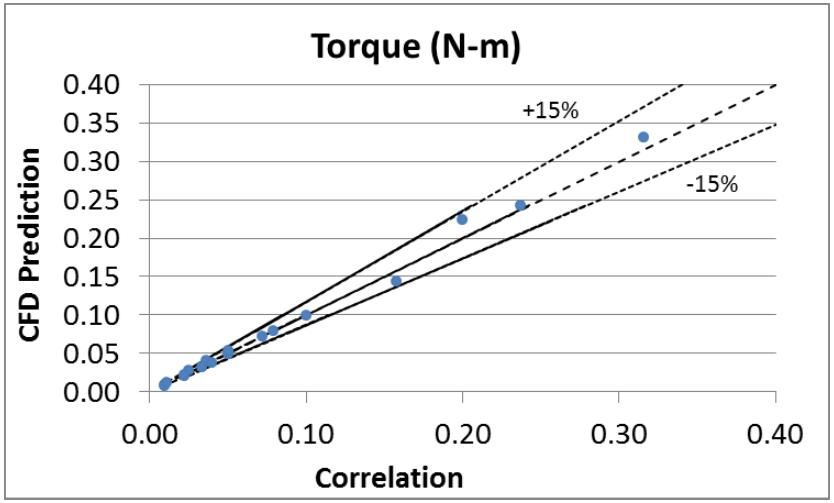
CFD results for scale-up of V6 impeller

Height (cm)	Diameter (cm)	Speed (rpm)	R (K/W)	Torque (J)	Mass Flow (kg/s)	Power (W)
3	10	2500	0.097	0.0092	0.026	2.4
3	10	2500	0.118	0.0085	0.023	2.2
1.5	15	1666	0.119	0.012	0.021	2.1
1.5	15	2500	0.082	0.028	0.034	7.4
1.5	15	3000	0.071	0.041	0.041	12.9
1.5	15	5000	0.047	0.099	0.087	52.0
3	15	1666	0.079	0.021	0.038	3.7
3	15	1666	0.082	0.022	0.039	3.8
3	15	2500	0.061	0.050	0.061	13.0
3	15	2500	0.058	0.054	0.060	14.1
3	15	3000	0.054	0.073	0.074	22.8
3	15	5000	0.030	0.223	0.13	117.0
4.5	15	1666	0.051	0.033	0.059	5.7
3	20	1250	0.053	0.038	0.061	5.0
3	20	2500	0.030	0.143	0.15	37.5
4.5	20	2500	0.028	0.243	0.17	63.5
6	20	1250	0.031	0.079	0.11	10.4
6	20	2500	0.022	0.331	0.23	86.6
6	20	5000	0.017	1.353	0.48	708.5

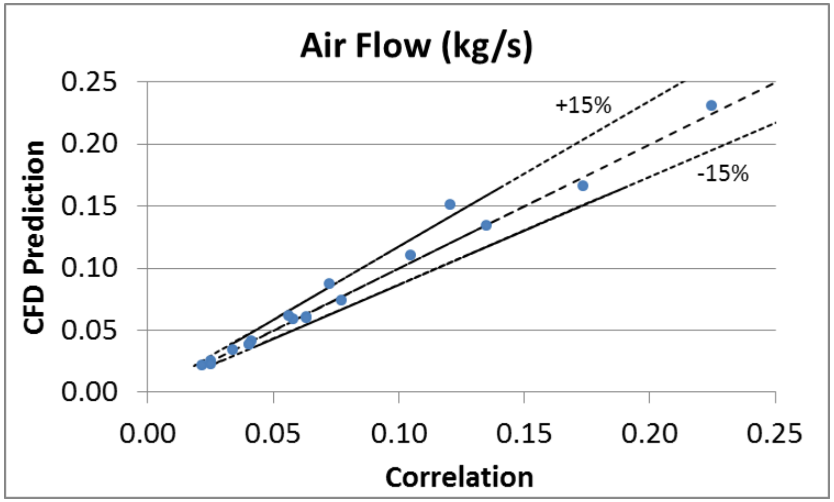
- V6 geometry: 55 fins, 45˚, 1” inner radius, 3 cm height, 1.5 power law
- Uniform in-plane scaling; 1.5X and 2X
- Independent vertical scaling for some cases; 0.5X, 1X, 1.5X, and 2X
- Speed scaled inversely with diameter based on V6 @2500rpm for some cases

Correlations based on CFD studies predict impeller performance to within ±20%

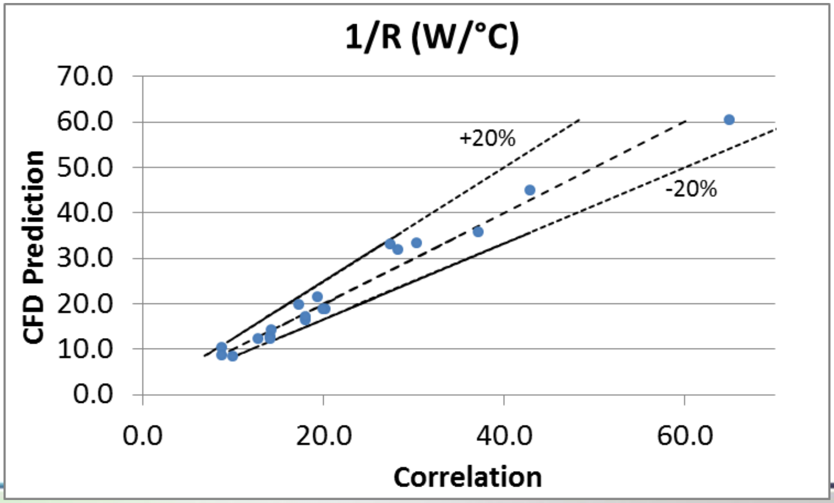
$$\tau = 4.8 \times 10^{-12} h \omega^2 d^4$$



$$Q = 1.16 \times 10^{-7} h^{0.9} \omega^{1.1} d^{2.25}$$



$$\frac{1}{R} = 2.82 \times 10^{-3} h^{0.5} \omega^{0.6} d^{1.8}$$



Note, since *Power* = $\tau \times \omega$:

$$P = 4.8 \times 10^{-12} h \omega^3 d^4$$

ω in rad/s
h in cm
d in cm

NEW PROJECTS IN FY14

We have four new projects for FY14 along with completion of the Demo Units

Continuing Work:

1. Complete 10 CPU Cooler Demonstration Units
2. Development of a Thermoelectric Cooling Device – University of Maryland/Optimized Thermal Systems

New Work:

3. Condenser for Residential Refrigerator – University of Maryland
4. Heat Exchanger for HVAC systems – ORNL
5. Heat Exchanger for Residential Heat Pumps – UTRC

Ten Demonstration Units will be completed by January 2014

Most components are complete and ready to assemble

Impellers:

- 5 are complete
  - machined, coated, motor rotor installed
- 6 more have been machined, not coated

Vapor chamber baseplates:

- 9 are complete, 1 more in progress
  - Machined and coated

Shafts:

- 10 are complete

Motor Stators:

- 3 wound and ready