NOVEL CONCEPTS FOR THE COMPRESSION OF LARGE VOLUMES OF CARBON DIOXIDE – PHASE III

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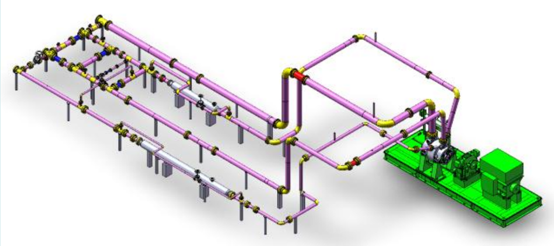
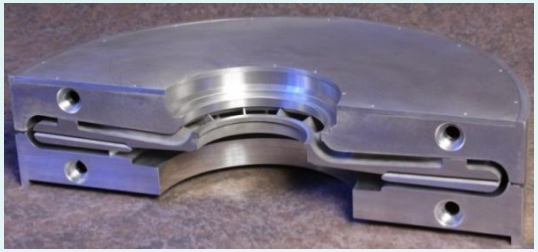


Project Funded by DOE NETL

DOE PM: Travis Shultz



DRESSER-RAND

11 Divisions

- Engine Emissions
- Fuels & Lubricants
- Automation
- Aerospace Electronics
- Space Science
- Nuclear Waste
- Applied Physics
- Applied Power
- Chemistry
- Electronics

- **Mechanical Engineering**
  - Rotating Machinery Group

- 1200 Acres
- 2 million Ft^2

- 3200 Employees
- 1200 Engineers
- 170 Buildings

Project Motivation

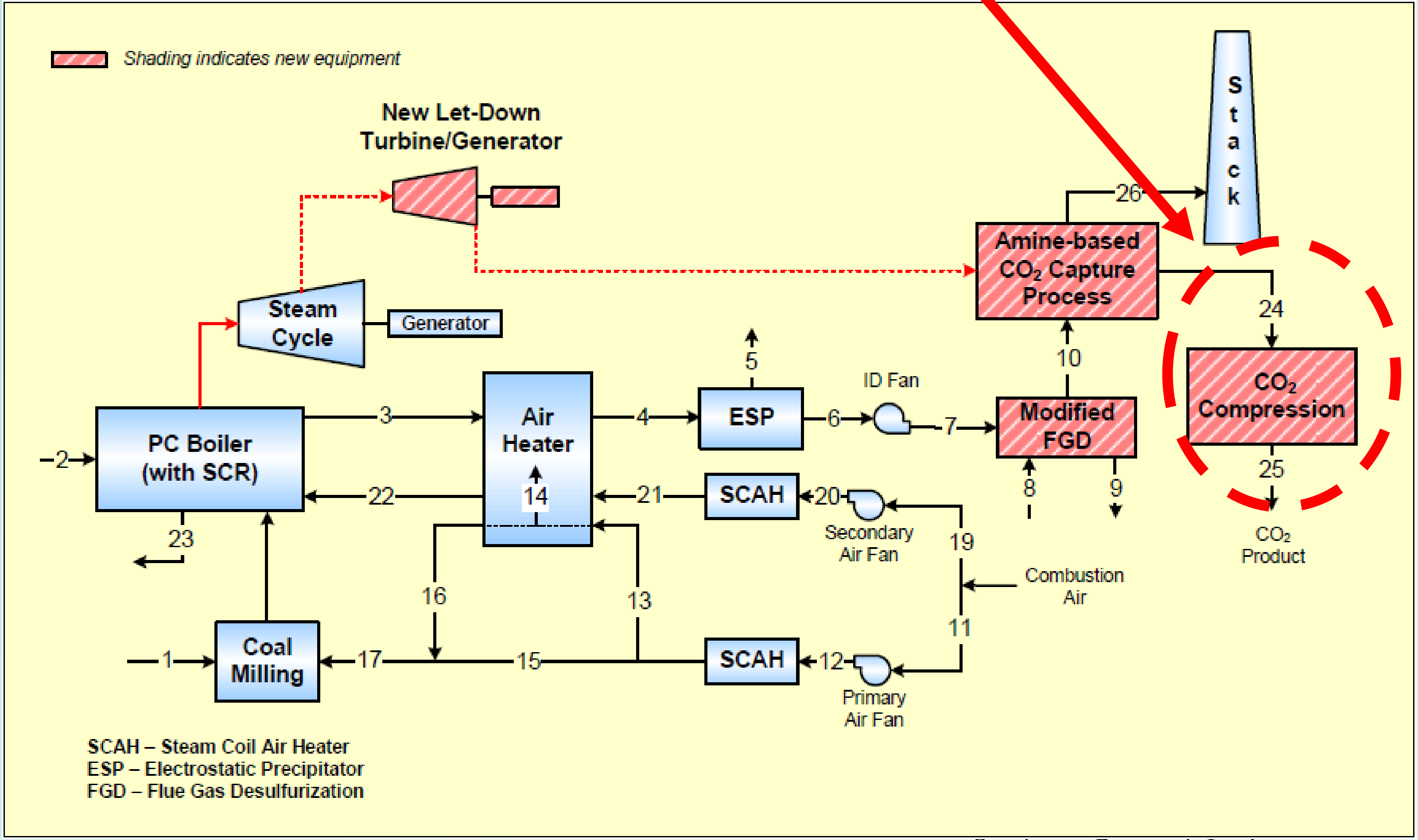
- CO_2 capture has a significant compression penalty - as high as 8 to 12%.
- Final pressure around 1,500 to 2,200 psia for pipeline transport or re-injection.
- Based on a 400 MW coal plant, the typical flow rate is ~600,000 to 700,000 lbm/hr.
- Project goal: Double-digit reduction of compression power for CO_2 capture.
- Many thermodynamic processes studied.
- Several challenges with the application discussed.

Project Overview

- Phase I (Completed in 2007)
  - Perform thermodynamic study to identify optimal compression schemes
- Phase II (Completed in 2010)
  - Test Rig testing of two concepts:
    - Isothermal compression (complete)
    - Liquid CO_2 pumping (complete)
- Phase III (Kicked off 2nd Qtr 2011)
  - Pilot scale compression plant
  - 55,000 lbm/hr

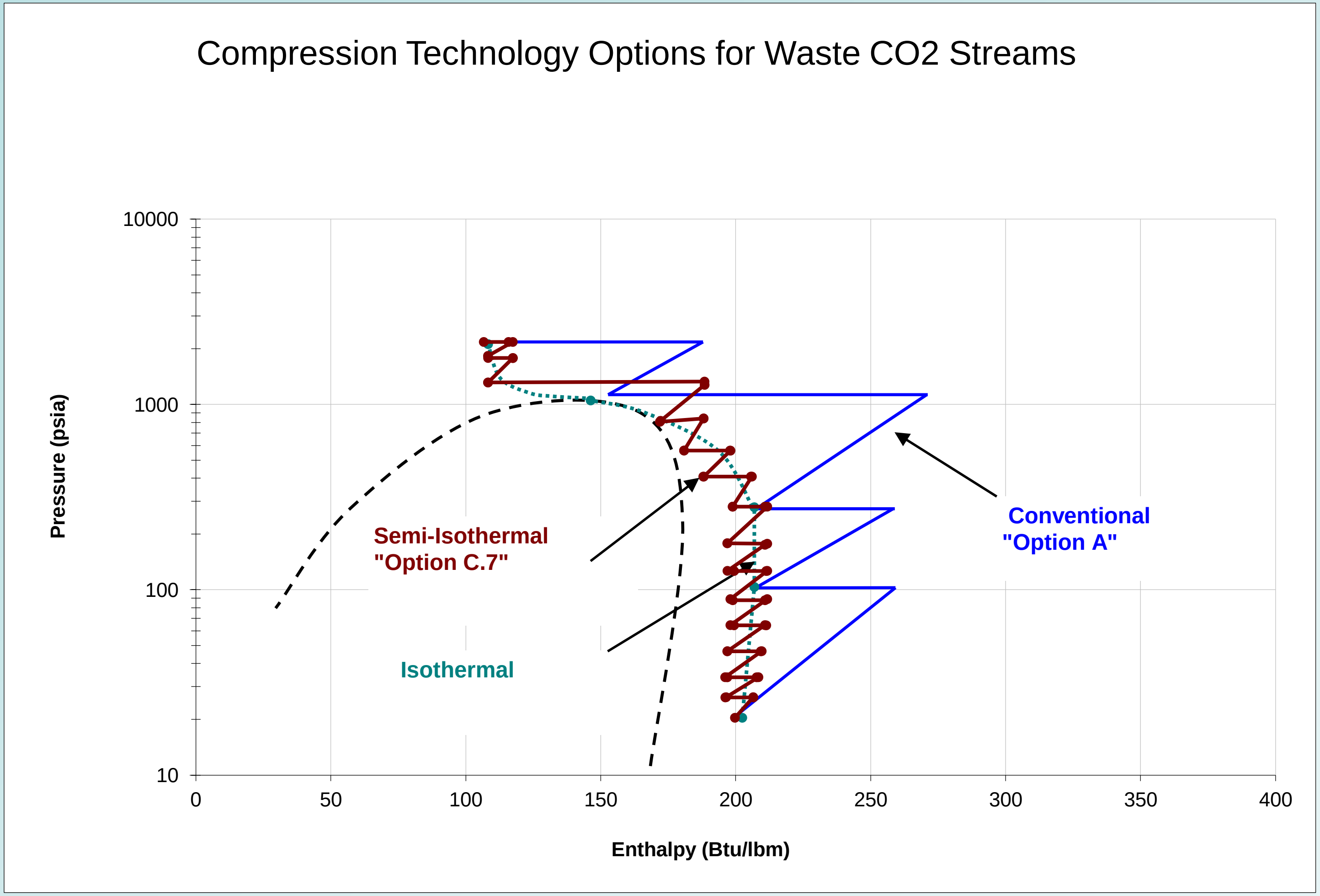
DOE PC Reference Case

- Only CO_2 stream considered



DOE/NETL report 401/110907

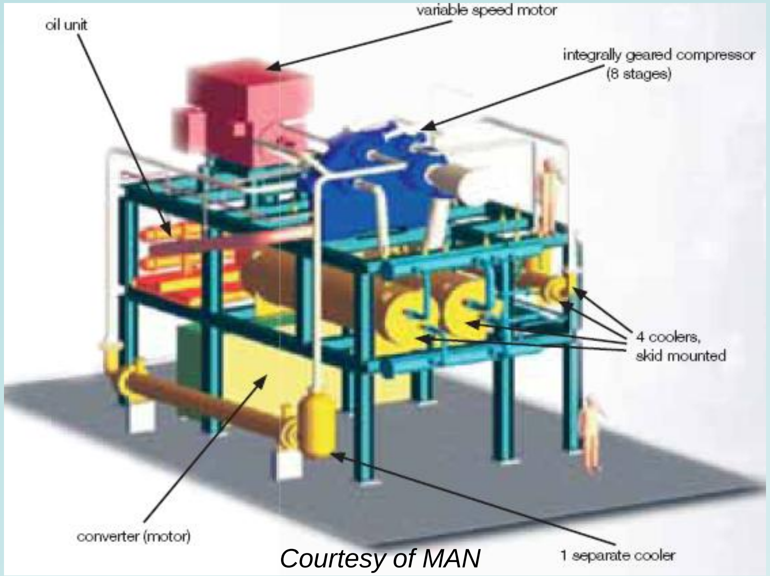
Proposed Solution for Optimal Performance



Optimal solution utilizes inter-stage cooling

Challenges: High Reliability

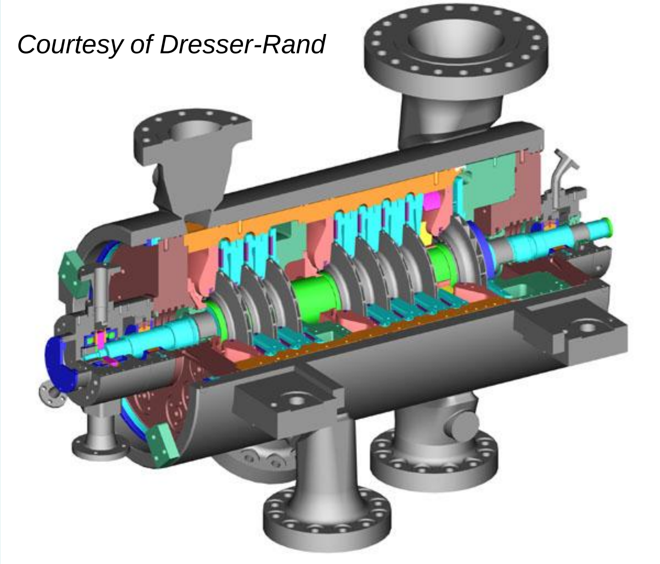
Integrally Geared Isothermal Compressor



Courtesy of MAN

- Integrally geared can achieve near isothermal compression
- Can contain up to 12 bearings, 10 gas seals plus gearbox
- Typically driven by electric motor
- Impellers spin at different rates
  - Maintain optimum flow coef.

Single-Shaft Multi-stage Centrifugal Compressor



Courtesy of Dresser-Rand

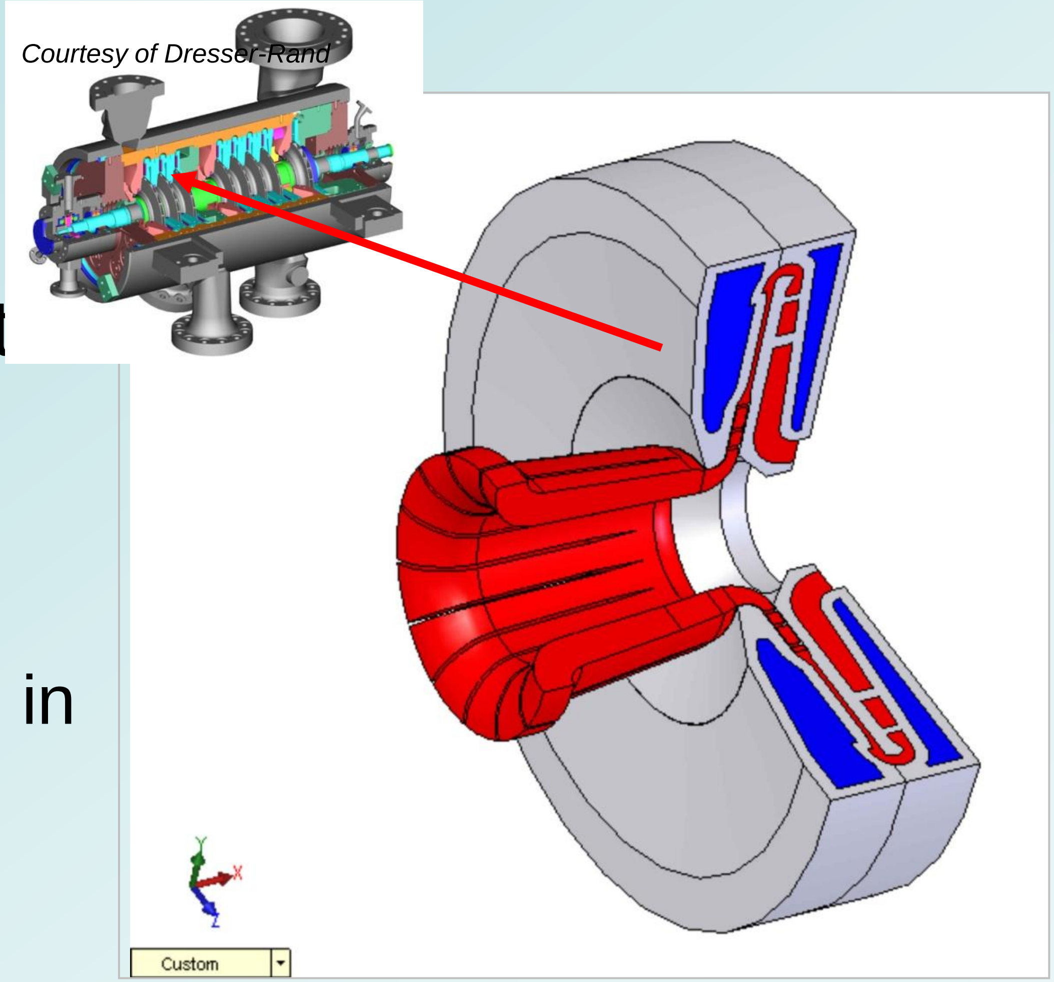
- Multi-stage centrifugal proven reliable and used in many critical service applications currently (oil refining, LNG production, etc.)
- Fewer bearings and seals
  - (4 brgs & seals for 2 body train)
- Can be direct driven by steam turbine

Phase 2 Project Goals

- Develop internally cooled compressor stage that:
  - Provides performance of an integrally geared compressor
  - Has the reliability of a in-line centrifugal compressor
  - Reduces the overall footprint of the package
  - Has less pressure drop than a external intercooler
- Perform qualification testing of a refrigerated liquid CO_2 pump

Internally Cooled Compressor Concept

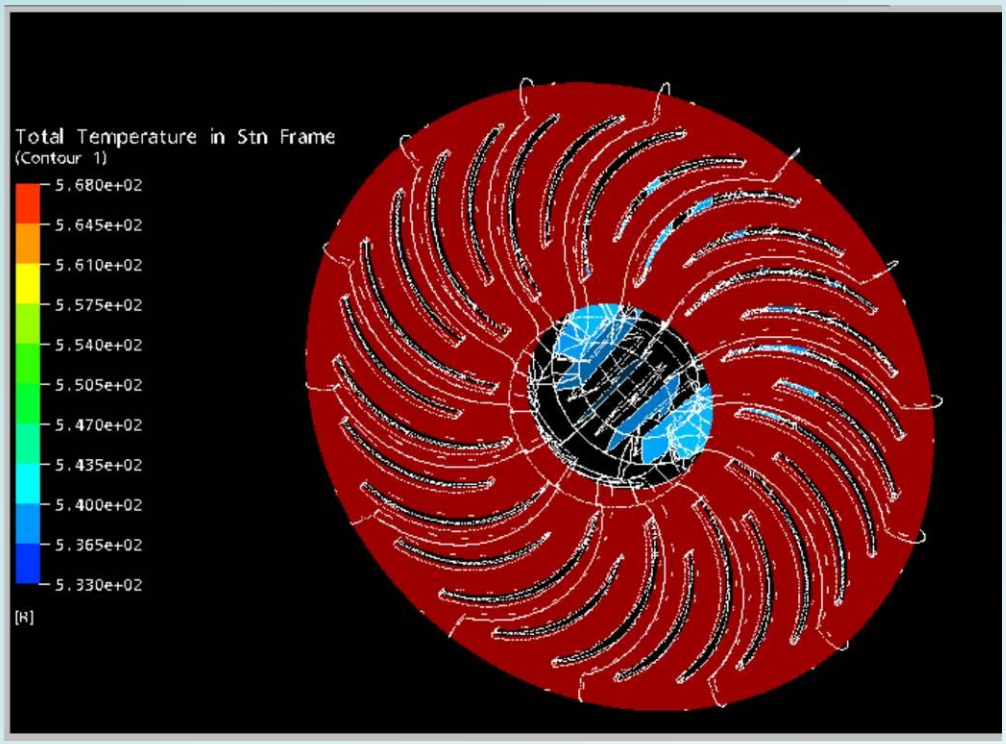
- Investigate an internally-cooled compressor concept
  – Red - CO_2 flow path through compressor stage
  – Blue - Liquid cooling in the diaphragm
  – Grey - Solid



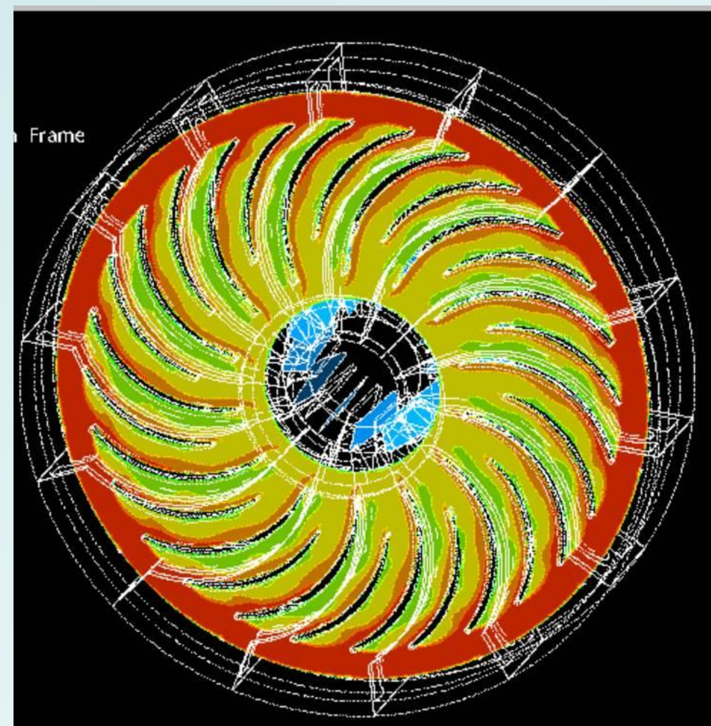
Conjugate Heat Transfer CFD Model

- Predicted temperature in return channel with and without internal cooling.

Without Heat Transfer

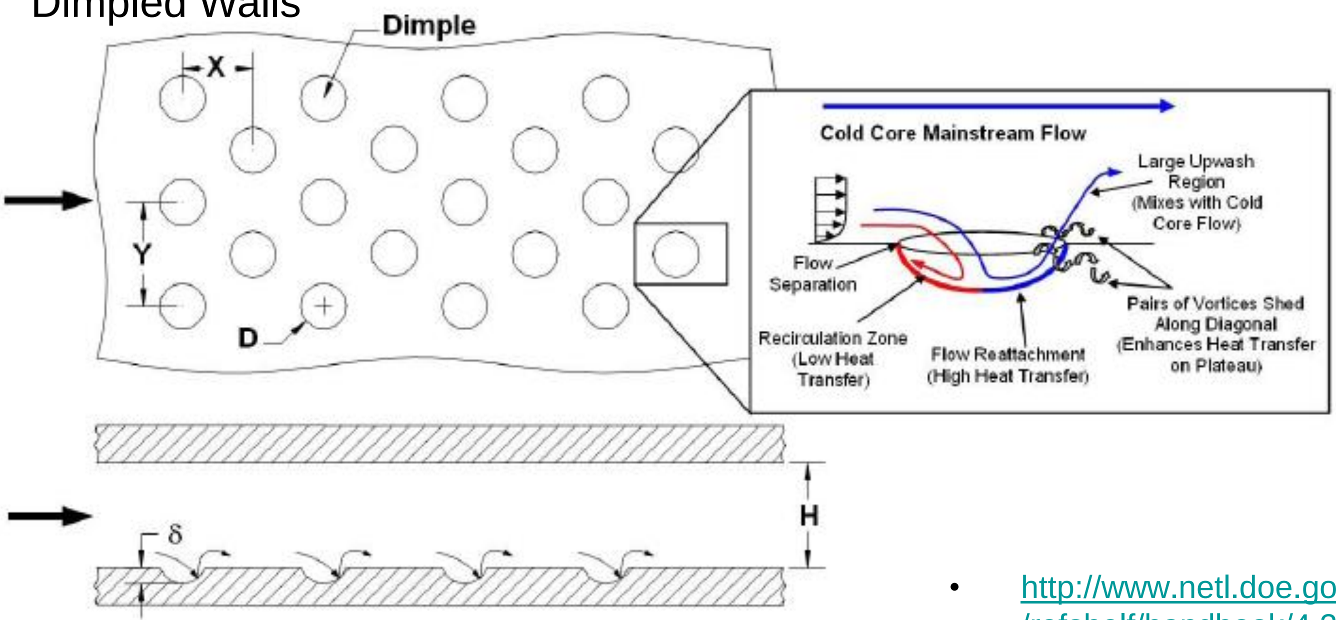


With Heat Transfer



Heat Transfer Enhancement

Dimpled Walls



- http://www.netl.doe.gov/technologies/coalpower/turbines/refshelf/handbook/4.2.2.2.pdf

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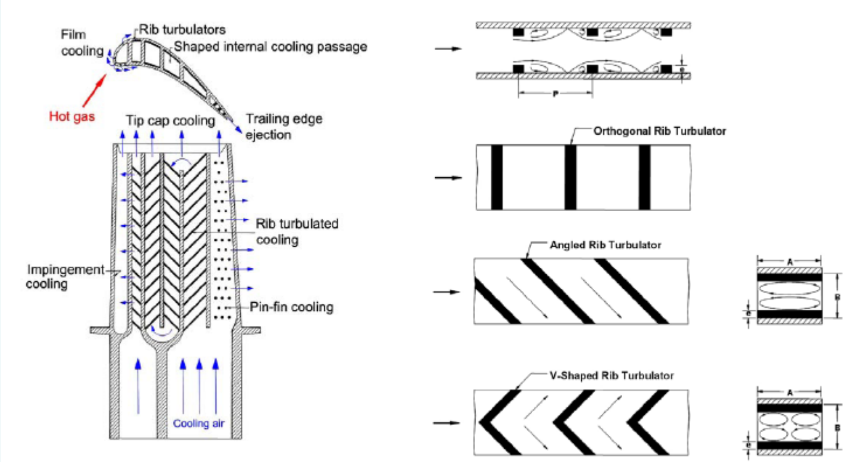
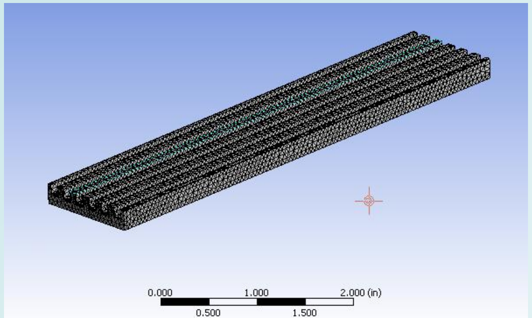
Ribs on Walls



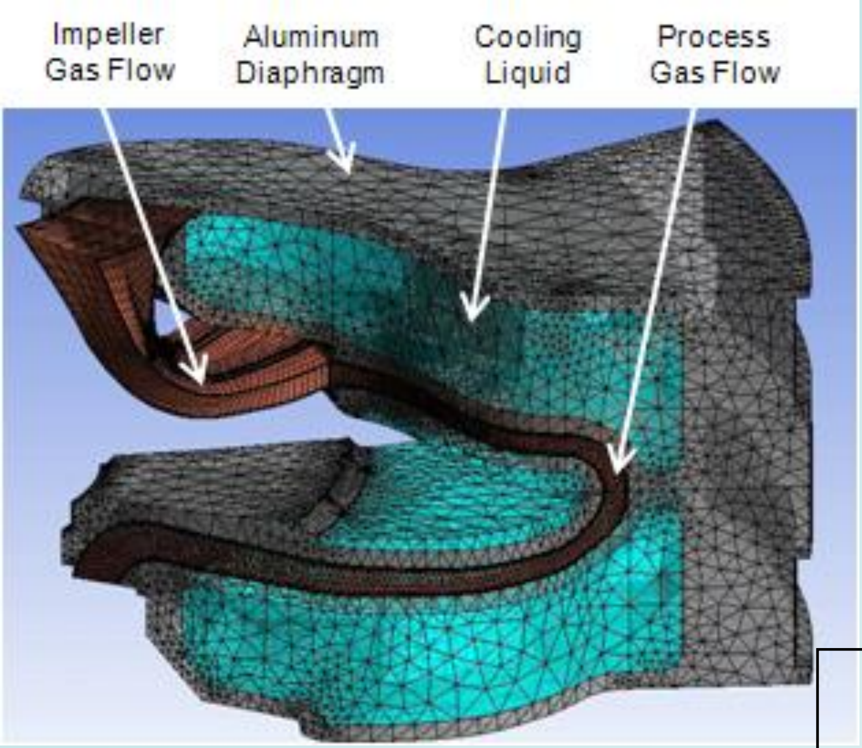
Grooved Airfoil Surface



Benefits of a Cooled Diaphragm

- Provides similar performance of an integrally geared compressor
- Has the reliability of a in-line centrifugal compressor
- Reduces the overall footprint of the package
- Has less pressure drop than an external intercooler
- In some applications, a cooled diaphragm can eliminate the need for an external cooler
  - Use straight through vs. back-to-back
  - Reduce number of compressor bodies
- Compressor fouling can be reduced by lowering the gas temperature below the polymerization point (e.g. ethylene)

Conjugate Heat Transfer CFD Model



Grid from Full Conjugate Heat Transfer (2-fluid) Section Model

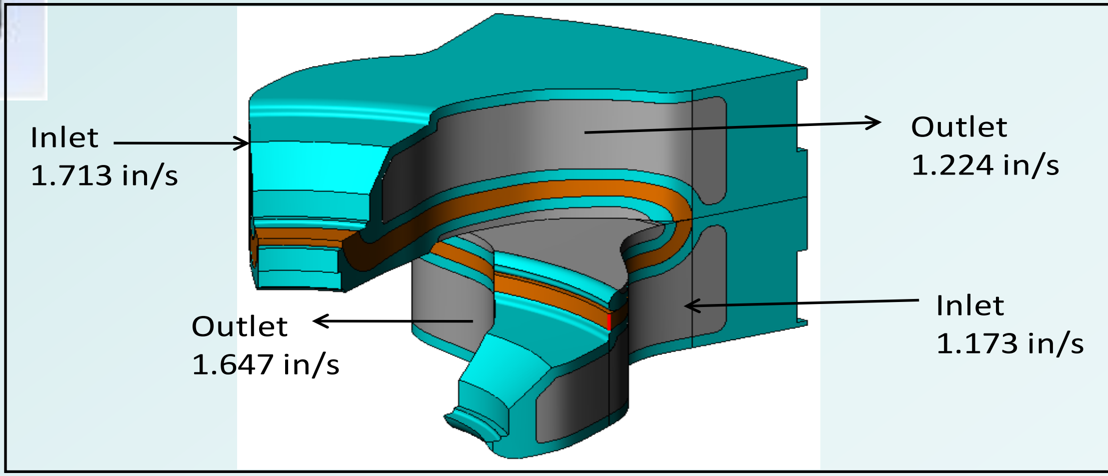
Models Used:

1. **Heat transfer coefficients on liquid interface**
2. **Full conjugate heat transfer model**

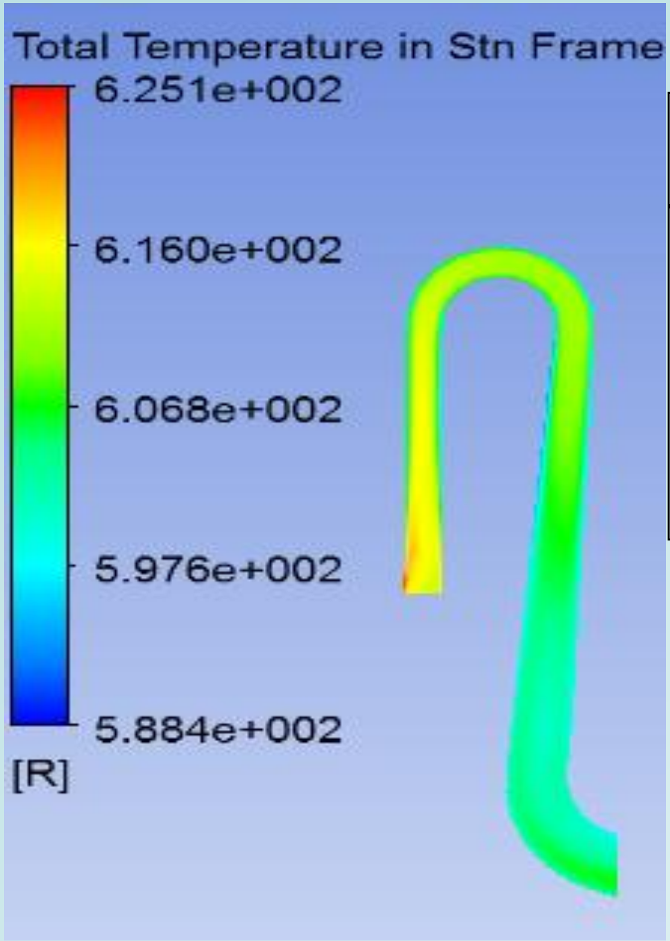
Adiabatic Results (No cooling)

	OEM Data	Model	(%) Difference
Total Pressure Ratio	1.550	1.648	6.3
Total Temperature Ratio	1.136	1.139	0.3
Gas Power [HP]	102.0	104.3	2.3

Flow Boundary Conditions for Cooling Fluid



CFD Results of Adiabatic and Conjugate Heat Transfer Models



Model	Quantity	Impeller Ratio	Stage Ratio
Adiabatic	Total Pressure	1.773	1.670
	Total Temperature	1.142	1.142
Diabatic with Heat Transfer Coefficients	Total Pressure	1.764	1.671
	Total Temperature	1.141	1.116
Diabatic with Full Conjugate Heat Transfer	Total Pressure	1.767	1.678
	Total Temperature	1.141	1.117

Good correlation between model using heat transfer coefficients on the liquid interface and the full two-fluid model

Analysis of Design Configurations

- *Adiabatic* – No heat transfer from CO_2, serves as the baseline for other cases.
- *Smooth wall (SW) heat transfer* – Smooth walls on both the water and CO_2 sides, i.e., no convection coefficient augmentation geometry used.
- *Smooth wall heat transfer at 9,155 rpm* – Same smooth wall geometry, as previous case; however, operated with a reduced stage pressure ratio to simulate a slower speed.
- *Smooth wall with higher radius ratio* – In order to increase heat exchanger effectiveness, surface area was increased by using a longer diffuser.
- *Ribbed water side walls and dimpled CO_2 side walls* – A convection coefficient augmentation case.
- *Ribbed water side walls, dimpled CO_2 side walls, and grooved airfoils* – The second convection coefficient augmentation case.

Cooled Diaphragm Benefits

- Sample compression from 15 to 250 psi straight through compressor
- Different heat transfer technologies explored

Geometry	RPM	Radius Ratio	# Stages	HX Effectiveness	Updated Gas Power Savings
Adiabatic Reference	12850	1.5	5	NA	0%
Smooth Wall	12850	1.5	5	0.15	7.0%
Smooth Wall	12850	1.8	5	0.197	8.6%
Ribs and Dimples	12850	1.5	5	0.25	1.2%
Ribs, Dimples, and Grooves	12850	1.5	5	0.31	-0.93%
Adiabatic Reference	9155	1.5	9	NA	0%
Smooth Wall	9155	1.5	9	0.15	13.3%
Smooth Wall	9155	1.8	9	0.197	15.3%

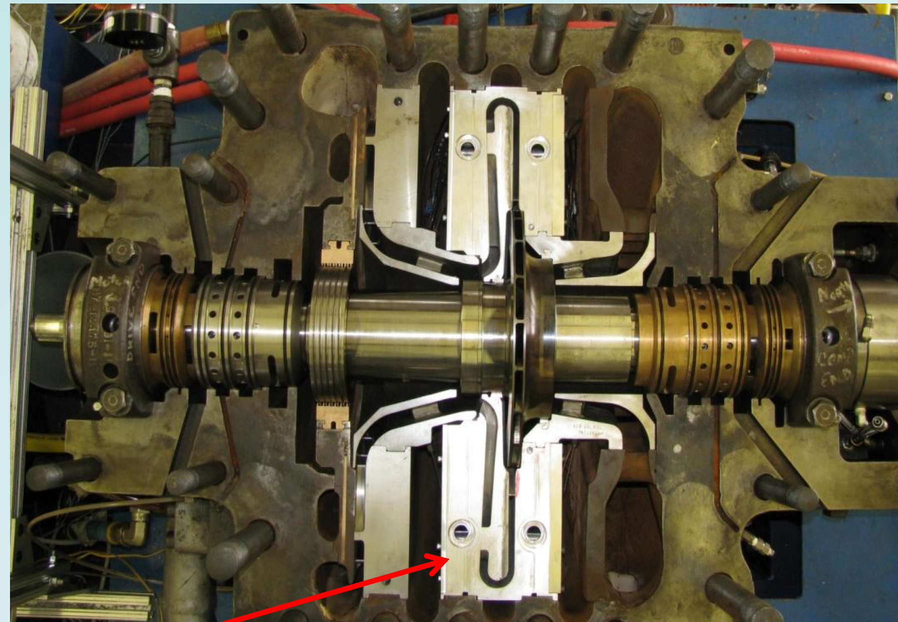
Configurations Considered

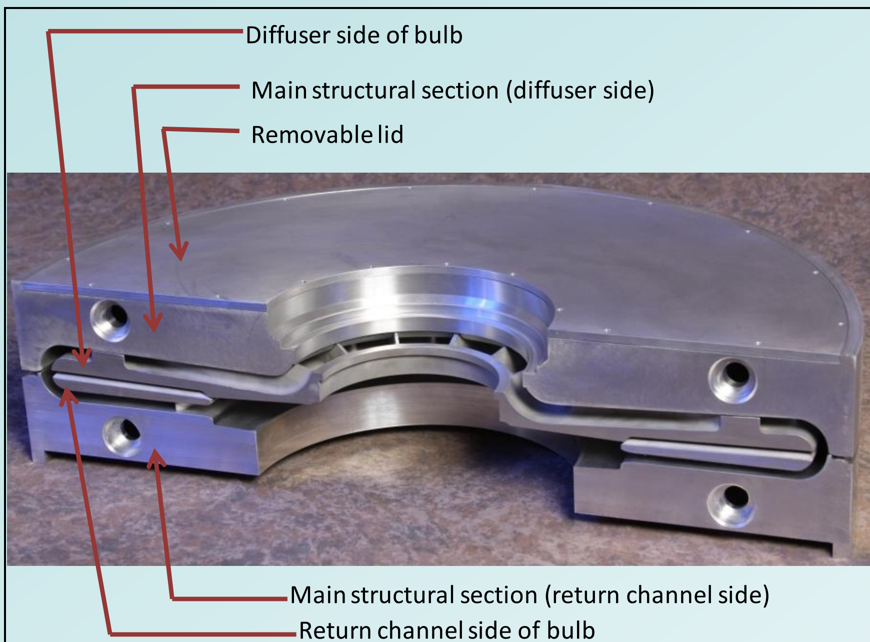
- Single stream inlet Pressure/Temperature = 14.8 psia / 115°F
- Discharge Pressure = 2,150 psia
- Intercooler/Aftercooler Exit Temperature = 115°F
- Liquefaction at 250 psia unless otherwise noted
- The following methods were analyzed for power comparisons:
  - DOE Baseline (efficiencies and refrigeration/liquefaction cycle performance calibrated to match data in [1]
  - Back-to-back LP and HP compressors with uncooled diaphragms
  - Back-to-back LP and HP compressors with cooled diaphragms, 15% and 20% effectiveness, 85°F cooling water
  - Back-to-back LP compression with cooled diaphragm (15% effectiveness, 85°F cooling water), liquefaction (ideal economizer), and pumping
  - Back-to-back LP compression with cooled diaphragm (15% effectiveness, 85°F cooling water), liquefaction (actual economizer), and pumping
  - Back-to-back LP compression with cooled diaphragm (15% effectiveness, 85°F cooling water) up to 425 psia, ideal economizer (removes all superheat), liquefaction, and pumping

[1] Ramezan, et. al., “Carbon Dioxide Capture from Existing Coal-Fired Power Plants,” DOE/NETL-401-110907, National Energy Technology Laboratory, Nov. 2007.

Case Description	Predicted Gas Power [hp/(lbm/min)]	Power Savings	Discharge Temp (°F)	Horsepower Breakdown
DOE Baseline	4.7634	0%	271 Comp -34 Condenser -7 Pump	4.7634 = 2.6603 Compressor + 1.7945 Condenser + 0.3086 Pump
D-R B2B LP and HP (Uncooled Diaphragm)	4.4489	6.6%	384 LP 296 HP	4.4489 = 2.8060 LP Compressor + 1.6429 HP Compressor
D-R B2B LP and HP (Cooled Diaphragm, 15% Effectiveness)	4.2672	10.4%	312 LP 229 HP	4.2672 = 2.7175 LP Compressor + 1.5497 HP Compressor
D-R B2B LP and HP (Cooled Diaphragm, 20% Effectiveness)	4.2083	11.7%	292 LP 210 HP	4.2083 = 2.6896 LP Compressor + 1.5187 HP Compressor
D-R B2B LP (Cooled Diaphragm, 15% Effect., Ideal Economizer), Liquefaction & Pumping	4.4092	7.4%	312 Comp 15 Economizer (gas)* -15 Condenser 8 Pump	4.4092 = 2.7175 Compressor + 1.4283 Condenser + 0.2634 Pump
D-R B2B LP (Cooled Diaphragm, 15% Effectiveness, Actual Economizer), Liquefaction & Pumping	4.4914	5.7%	312 Comp 46 Economizer (gas)* -15 Condenser 8 Pump	4.4914 = 2.7175 Compressor + 1.5105 Condenser + 0.2634 Pump
D-R B2B LP (Cooled Diaphragm, 20% Effectiveness) up to 425 psia, Economizer down to saturation T, Liquefaction & Pumping	4.8628	2.1%	400 Comp 20 Economizer (gas) 19 Condenser 45 Pump 95 Economizer (liquid)	4.8628 = 3.4551 Compressor + 1.1473 Condenser + 0.2605 Pump. Used DOE COP value for liquefaction, may be able to increase COP at higher T

Test Rig Construction

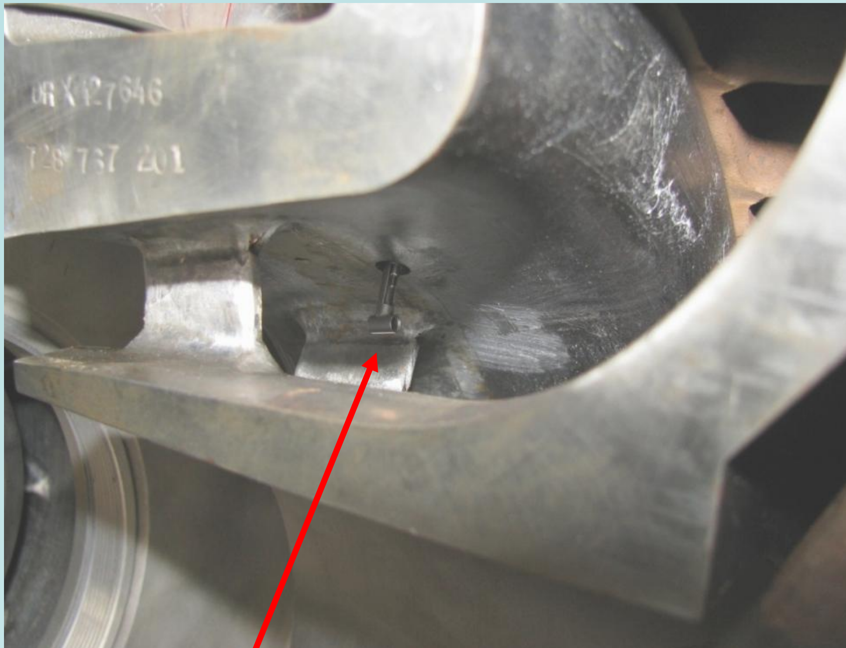


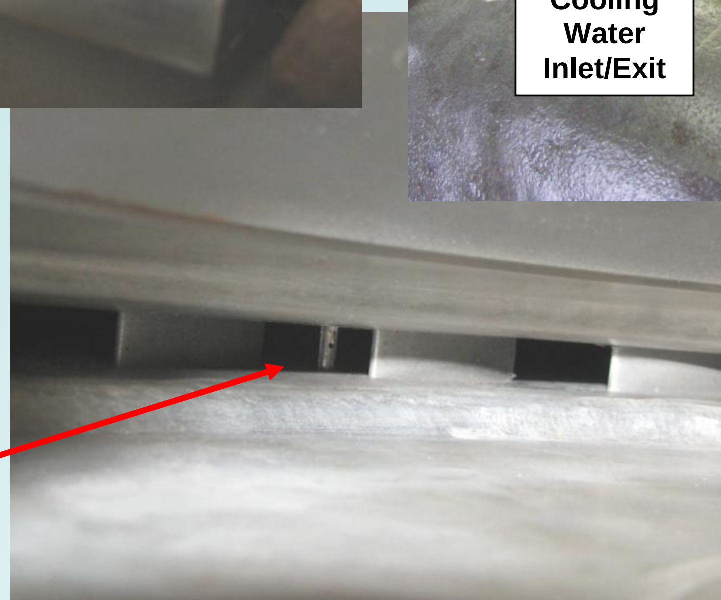


Closed Loop Test Facility

- Driven by 700 hp electric motor through gearbox
- Torque meter installed to measure power
- Loop rated to 300 psi suction and 500 psi discharge
- Test speeds up to 14,300 rpm

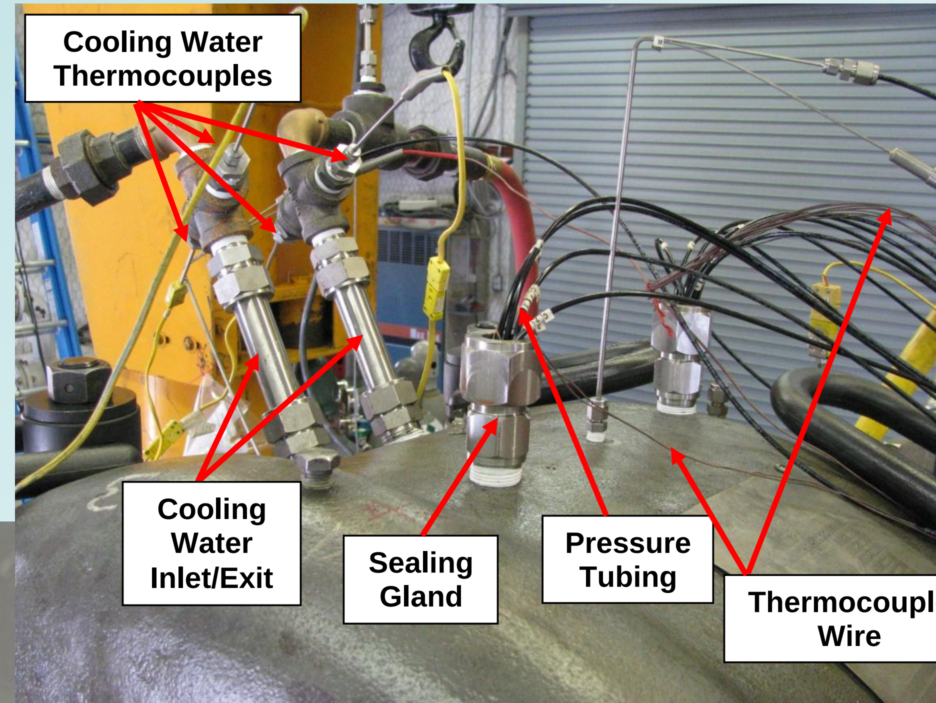
Instrumentation





Combination Kiel Head Pressure/Temperature Probe at Suction and Discharge Bridge-over

Half-Shielded Thermocouple Probe Near Impeller Exit



- 28 Temperature Probes
- 30 Pressure Measurements
- Flow Rate (CO_2 and Cooling)
- Speed
- Shaft Torque
- Axial Thrust
- Gas Samples Taken

Some Definitions

- Heat Exchanger Effectiveness

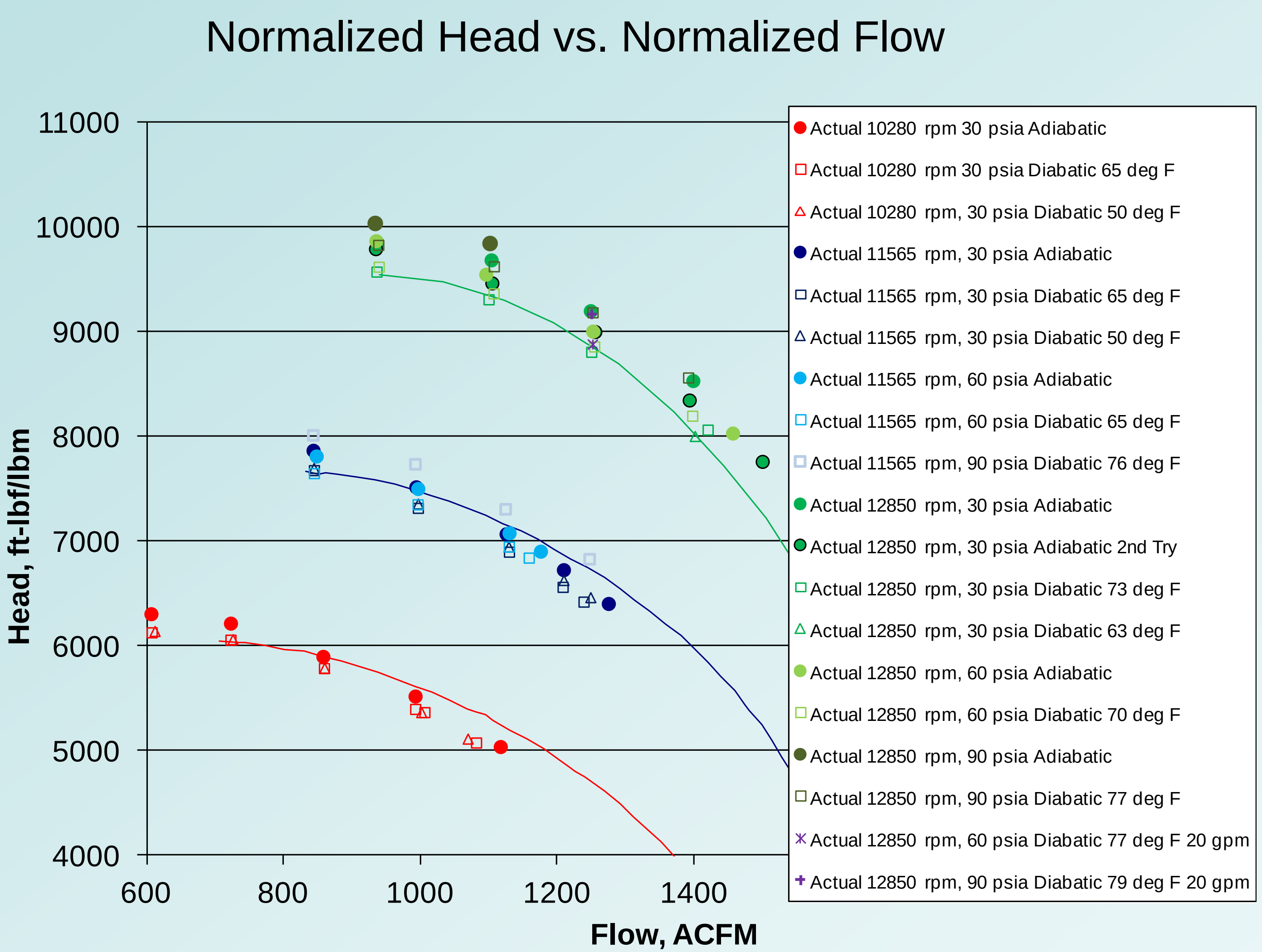
$$\varepsilon = \frac{\dot{Q}}{\dot{Q}_{max}} = \frac{Actual\ Heat\ Transfer\ Rate}{Maximum\ Possible\ Heat\ Transfer\ Rate}$$

where

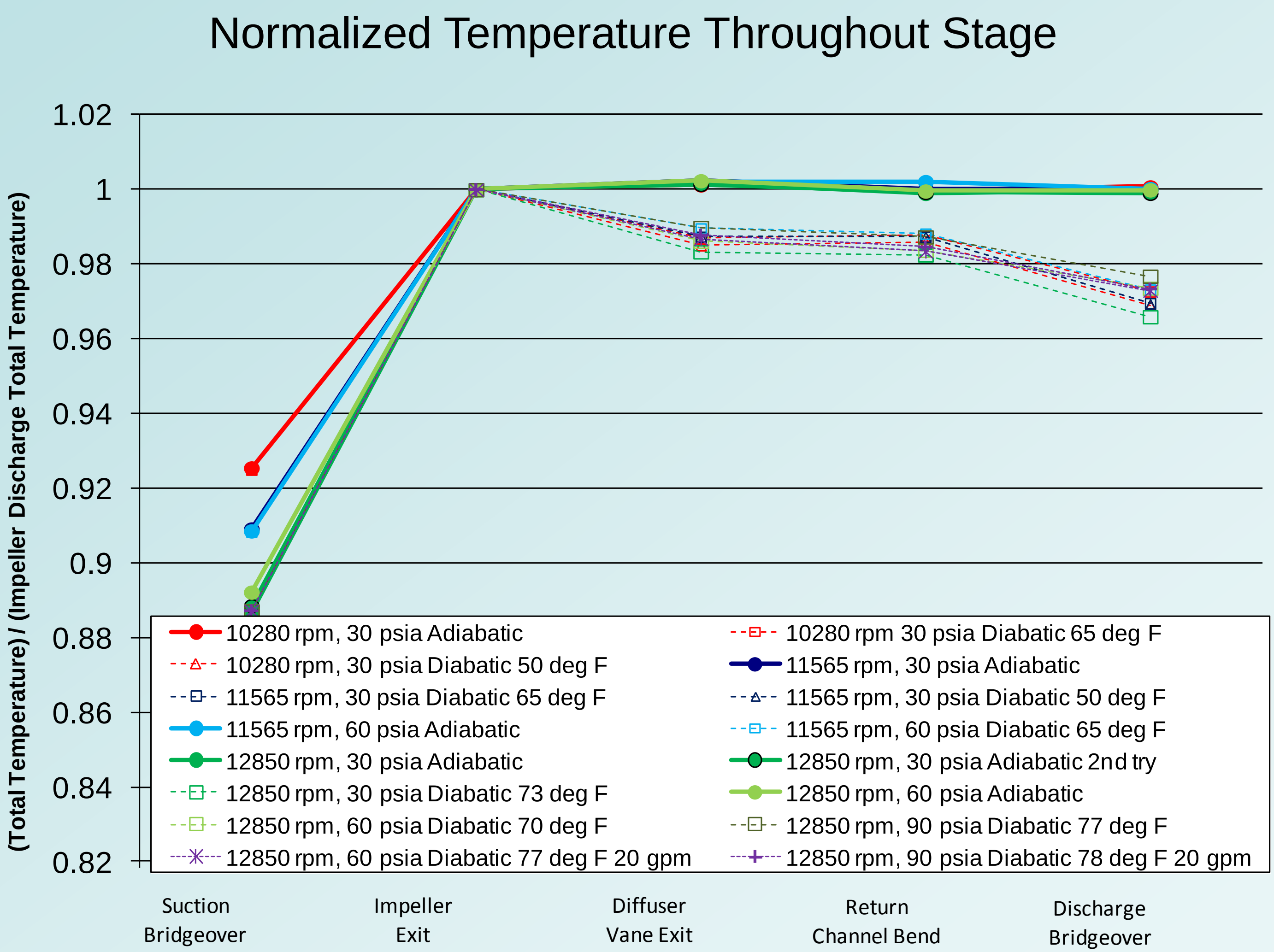
$$\dot{Q} = C_{H2O}\left(T_{H2O,out} - T_{H2O,in}\right) = C_{CO2}\left(T_{CO2,in} - T_{CO2,out}\right)$$

$$\dot{Q}_{max} = C_{min}\left(T_{CO2,in} - T_{H2O,in}\right)$$

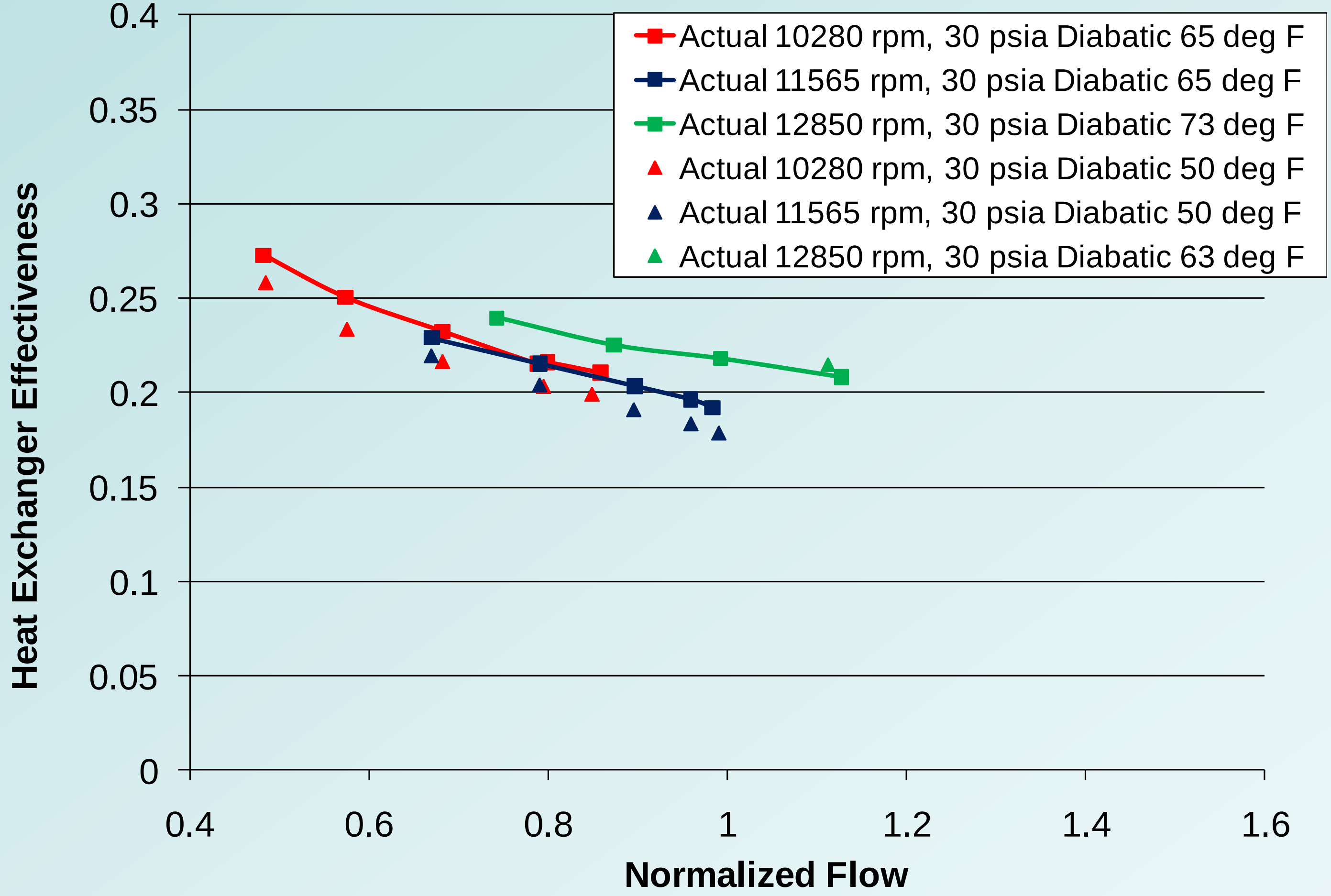
Measured Polytropic Head vs. Flow
30-90 psia (2-6 bar) Suction Pressure



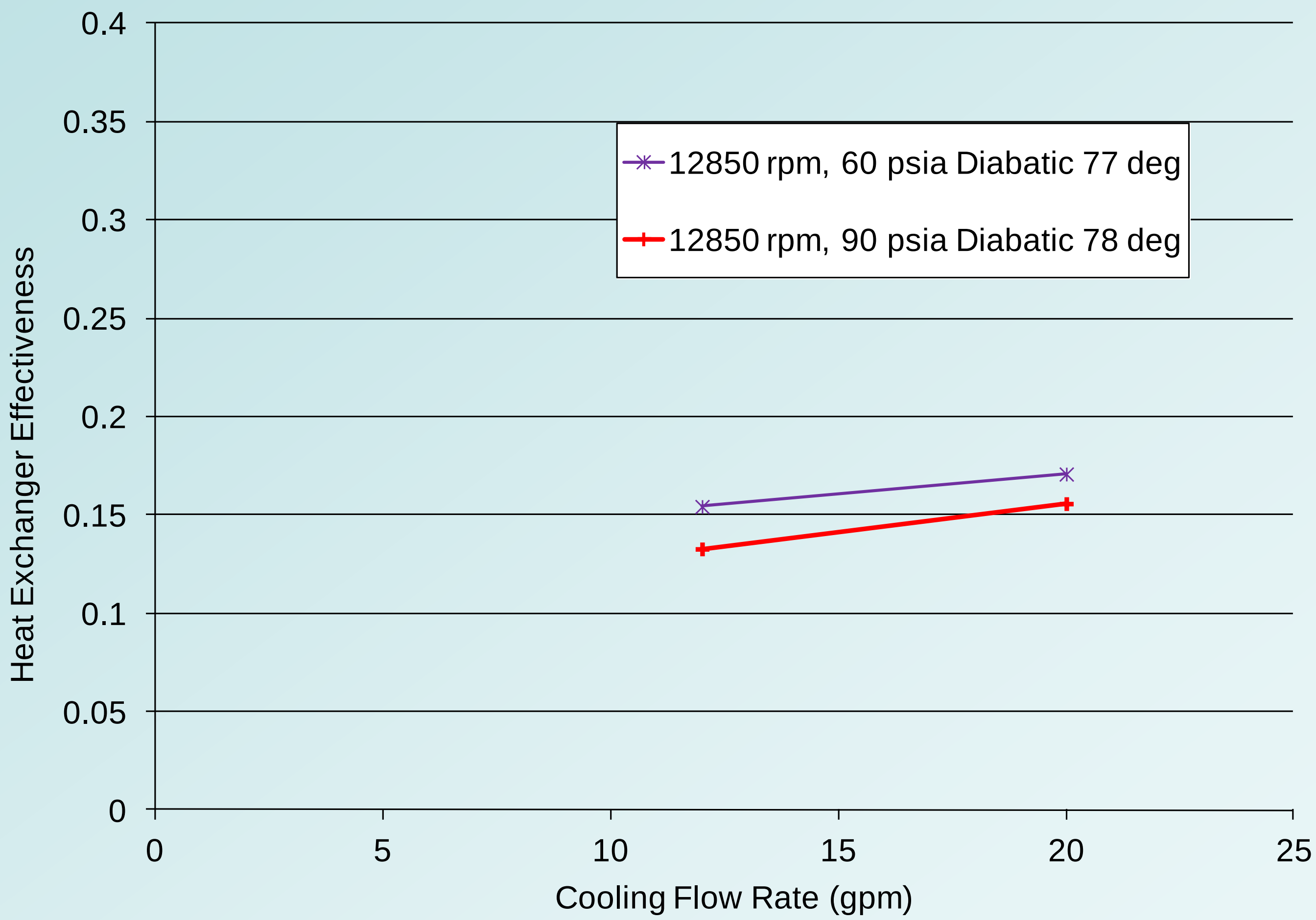
Measured Total Temperature Profiles



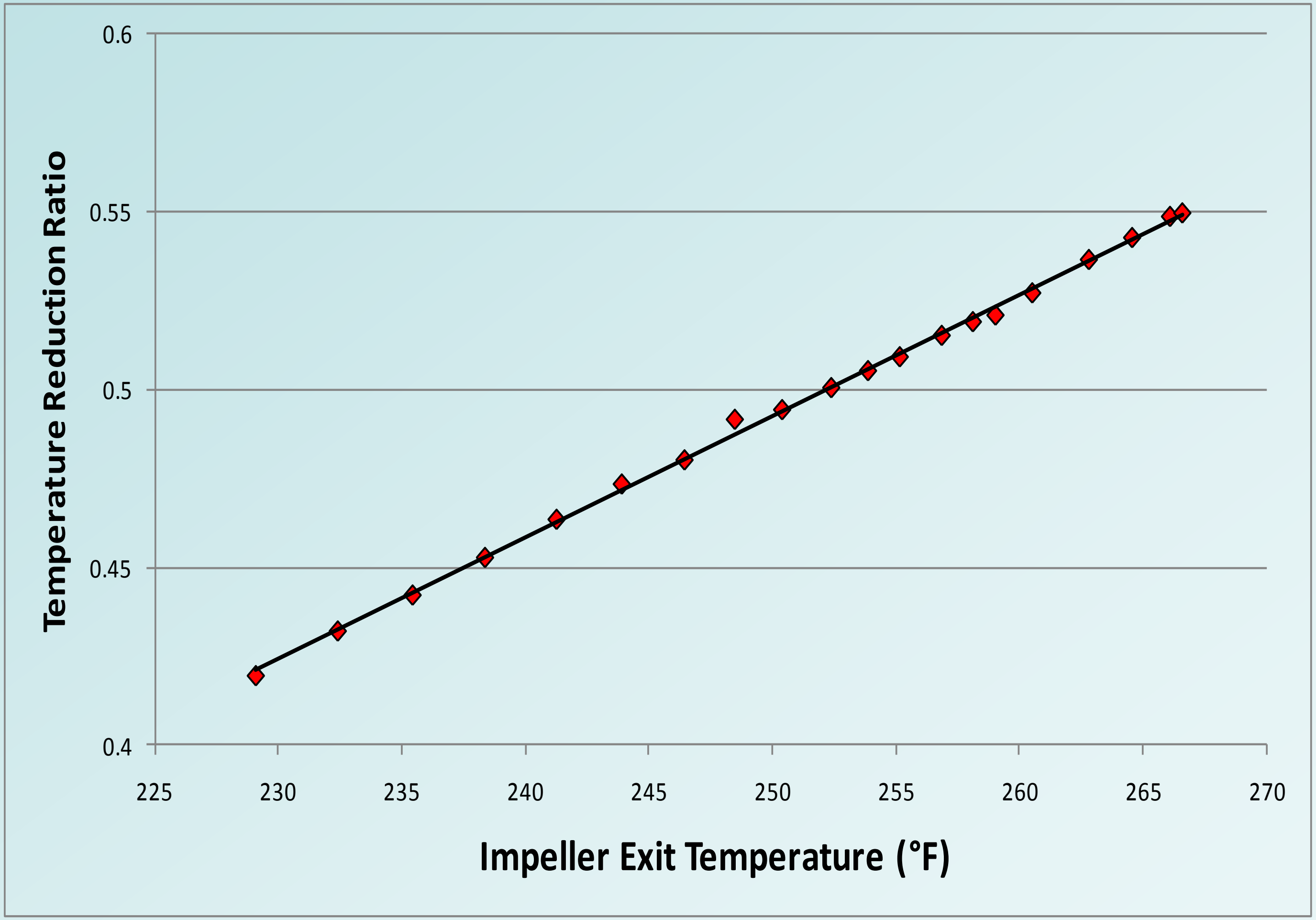
Measured Heat Exchanger Effectiveness vs. Flow at 30 psia Suction Pressure



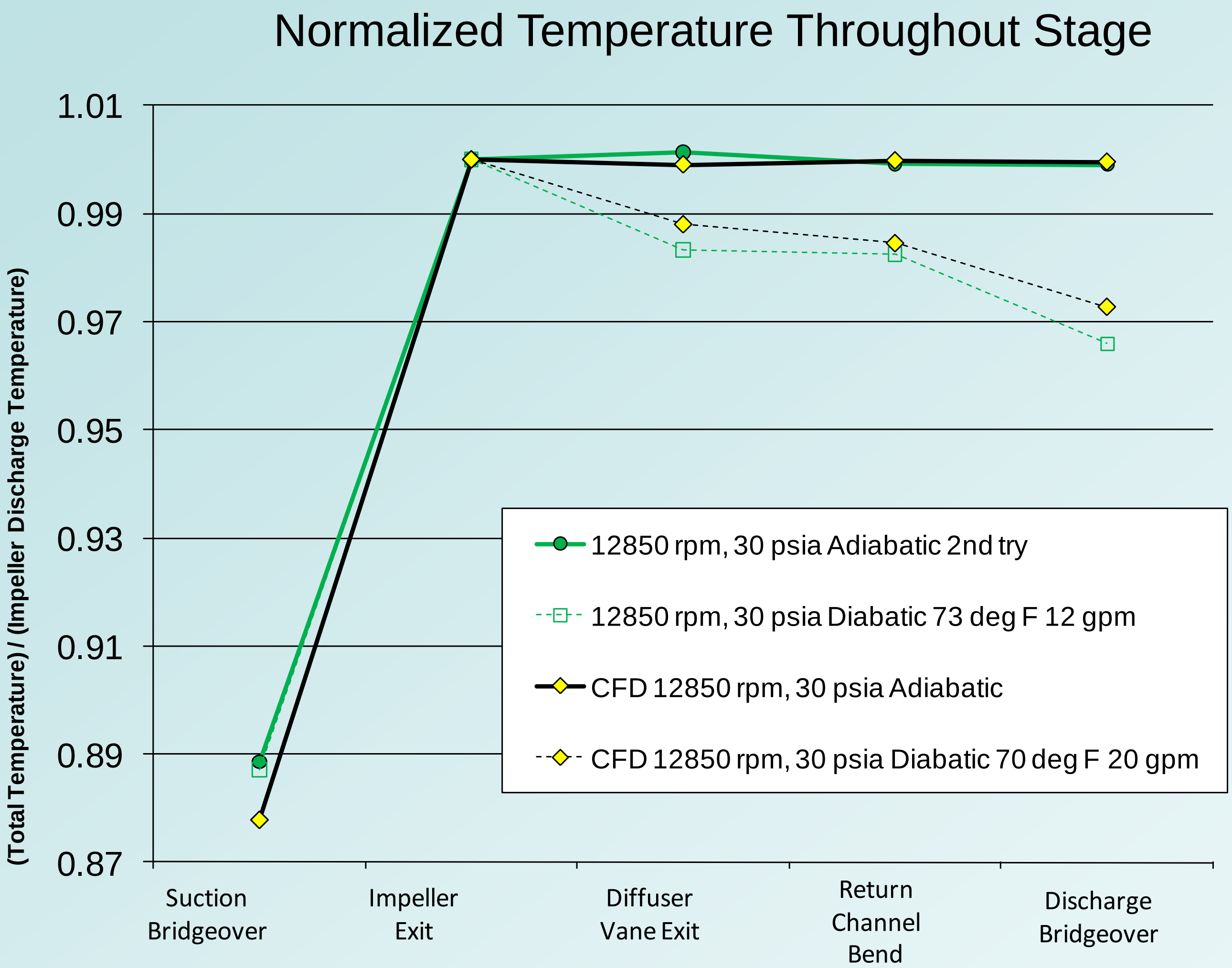
Heat Exchanger Effectiveness vs. Cooling Flow Rate



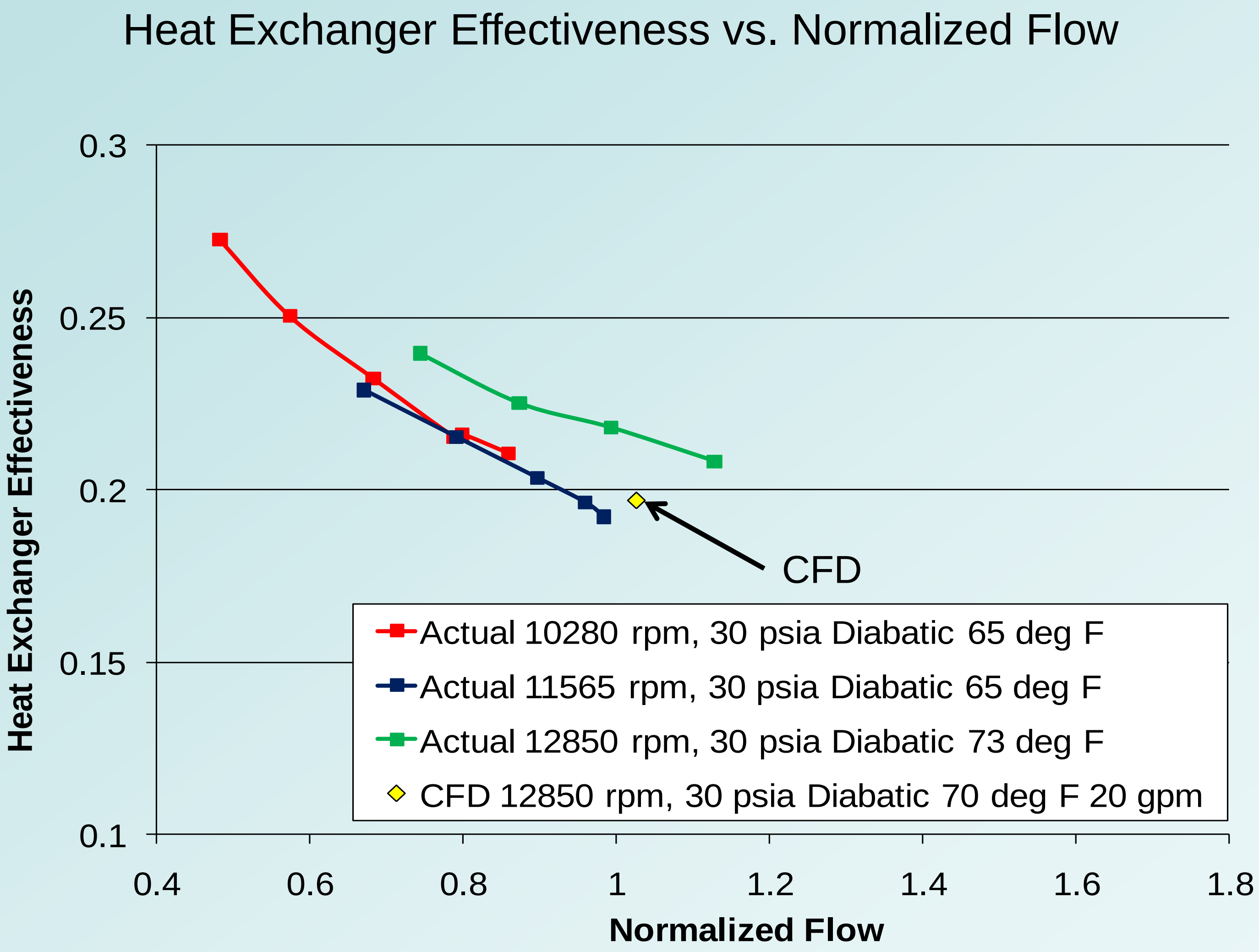
Fraction of Heat Removal in the Stage vs. Impeller Exit Temperature



Comparison to CFD Predictions



Comparison to Predictions



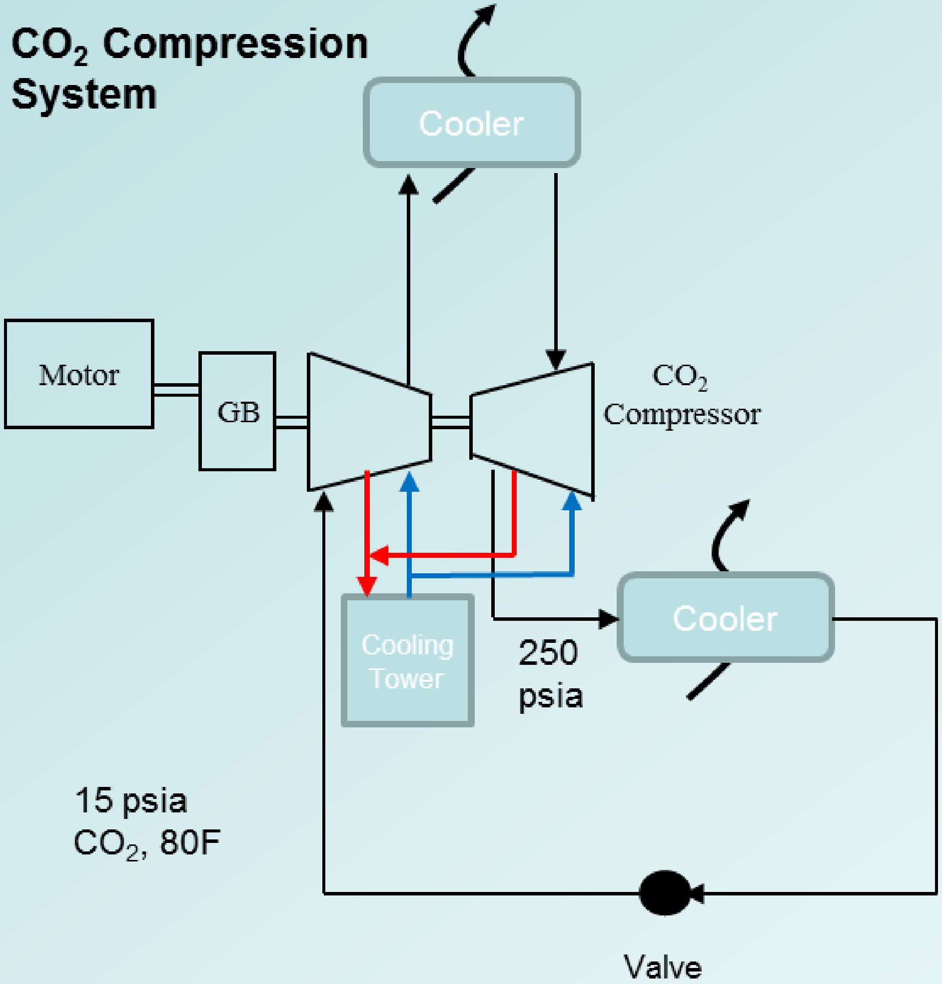
Phase 2 Summary

- Compressor Testing
  - Testing performed for a range of speeds, flows, suction pressure, suction temperature, cooling water flow and temperature
  - Testing performed both adiabatic and diabatic (with cooling)
  - Results show cooled diaphragm can remove up to 55% of the heat of compression in each stage
  - Heat exchanger effectiveness decreases slightly with increasing pressure
  - Heat removal improves in latter stages of a multi-stage compressor
  - Optimum cooling flow rate a function of the gas conditions.
  - Over 15% reduction in power is possible for a multi-stage application
- Technology is applicable to other compression applications with high pressure ratio
- Based on successful testing, a pilot scale compression facility is being developed.

Phase 3 Goals

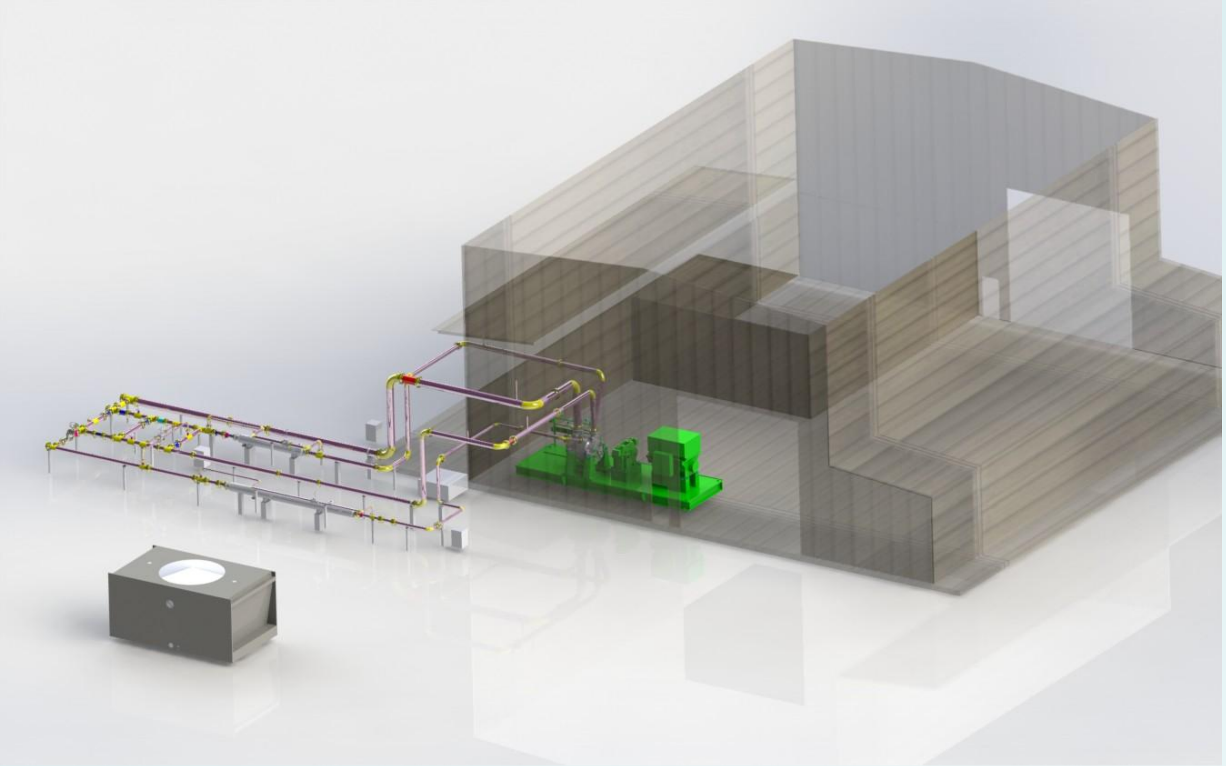
- The cooled diaphragm concept will be extended to a multi-stage design.
- A pilot scale test loop will be build based on a 3 MW Dresser-Rand 6 stage back-to-back compressor
- An overall power balance will be measured, including all coolers and cooling water pumps
- Technology will be considered field ready following this demonstration program

Phase 3 Pilot Test Facility

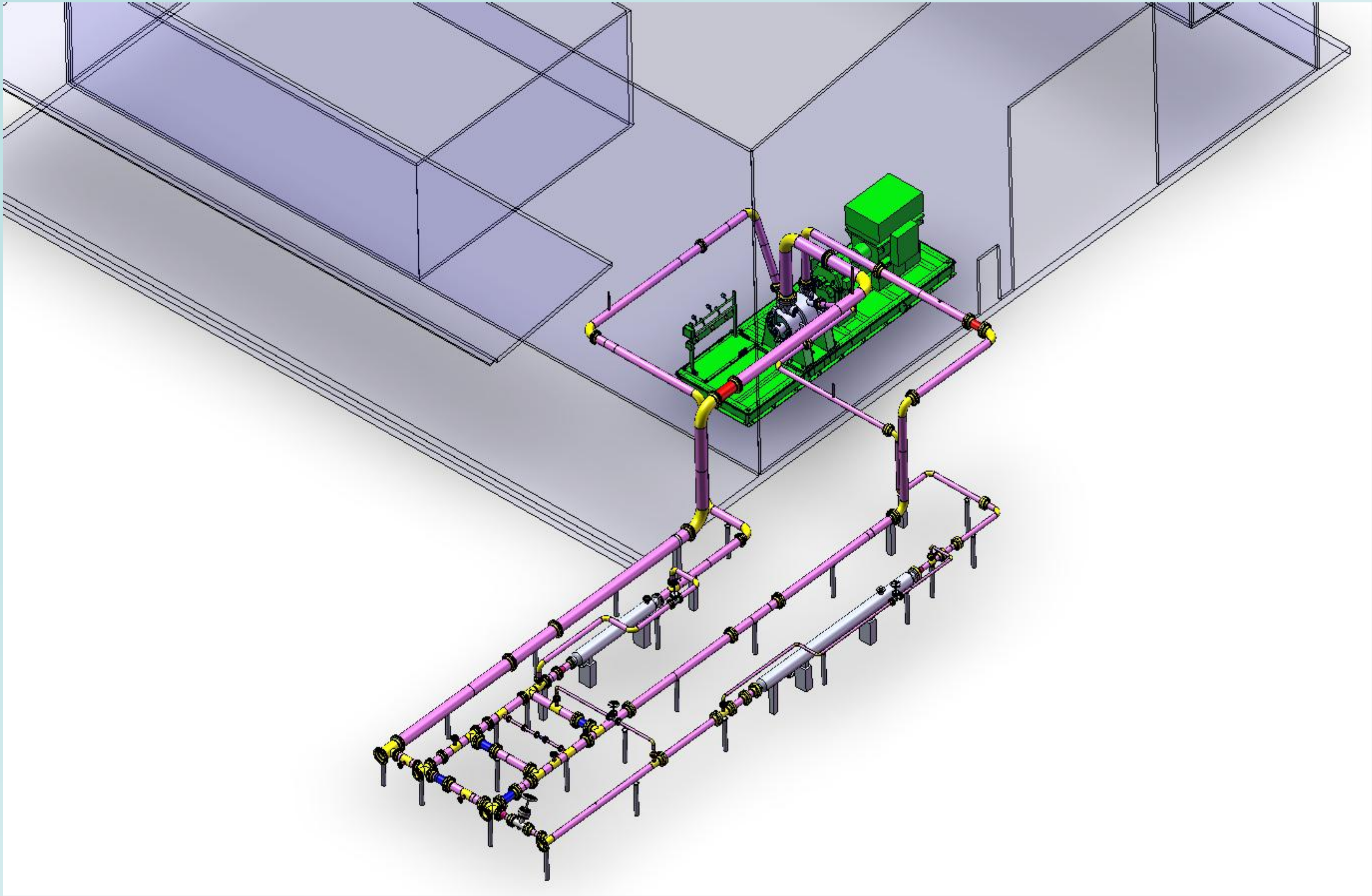


New Building and Compressor Facility

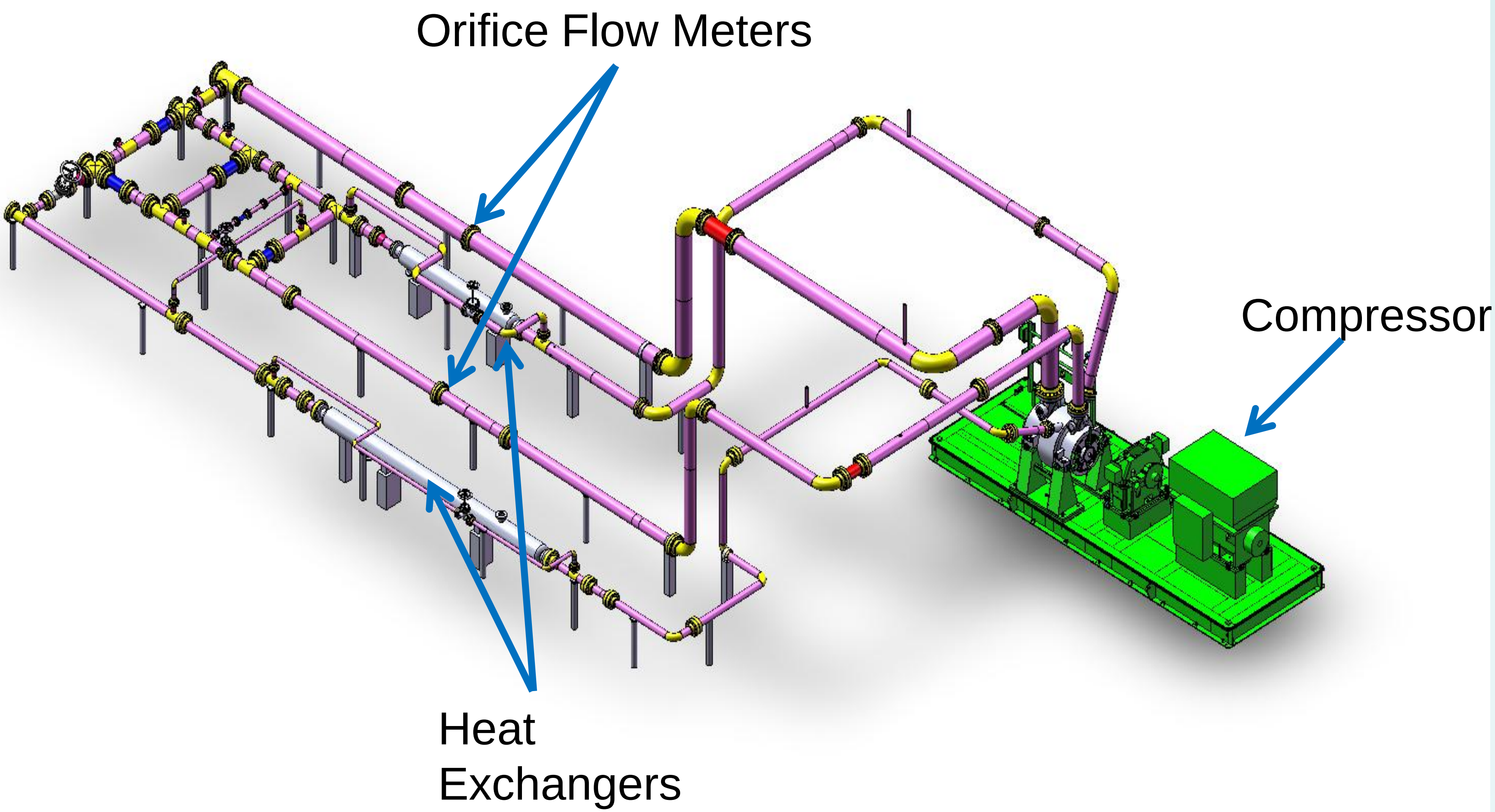
- New facility with high-bay to house compressor
- Piping system permits series or parallel operation of back-to-back compressor



3 MW Compression Facility

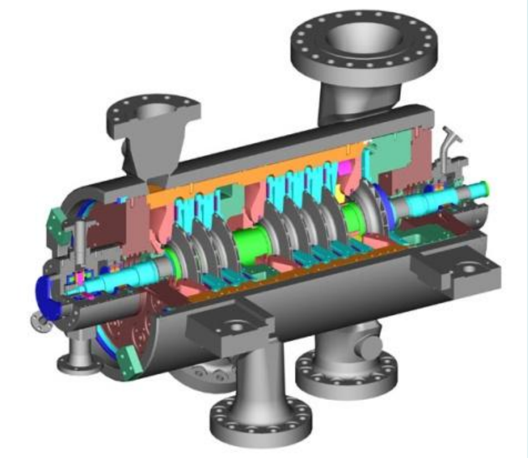


3 MW Compression Facility



Compressor Specifications

- Dresser-Rand DATUM D12R6B
- Approximate operating conditions are:
  - Suction pressure: 15 - 25 psi (rated to 300 psi)
  - Discharge pressure: 230 - 260 psi
  - Compressor casing rated for 1,200 psi (loop rated to 3200 psi)
  - Mass flow rate = 55,000 - 75,000 lb_m/hr (6000 to 9500 ACFM)
  - Power: 3,000 HP (can be upgraded to 10,000 HP)



- Design: Multistage centrifugal compressor with back-to-back sections with internally cooled diaphragm technology
- Intercooling and aftercooling will be supplied to run compressor in adiabatic mode
- The compressor will be mounted with a variable speed electric motor and gearbox on a single skid.
- Dry gas seal system and the variable frequency drive will also be supplied.
- Equipped with torque meter to directly measure power savings

Turbomachinery Research Facility

- New 10,000 ft^2 lab space is scheduled to be completed this month
- 4,000hp centrifugal compressor
- Dedicated rotordynamics and gas testing labs
- 40-ton bridge crane
- 14’x14’x10’ spin pit

Multi-Stage DATUM D12R6B

Future Work

- Compressor package will be delivered July 2013.
- Commissioning late 2013.
- Testing Complete 1st Quarter 2014.

Questions???

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