Ramgen Power Systems

Workshop on Future Large CO2 Compression Systems

DOE Office of Clean Energy Systems, EPRI, and NIST
National Institute of Standards and Technology (NIST) Headquarters,
Gaithersburg, MD 20899
March 30-31, 2009
Forward Looking Statement

Some of the information contained in this document contains “forward-looking statements”. In many cases you can identify forward-looking statements by terminology such as “may,” “will,” “should,” “expects,” “plans,” “anticipates,” “estimates,” “predicts,” “potential,” or “continue,” or the negative of such terms and other comparable terminology. Forward-looking statements are only predictions and as such inherently include risks and uncertainties. Actual events or results may differ materially as a result of risks facing Ramgen Power Systems, LLC (“Ramgen”) or actual results differing from the assumptions underlying such statements. These forward-looking statements are made only as of the date of this presentation, and Ramgen undertakes no obligation to update or revise the forward-looking statements, whether as a result of new information, future events or otherwise. Your decision to remain and receive the information about to be presented to you shall constitute your unconditional acceptance to the foregoing.
Company

• Privately-held R&D company founded in 1992
• Focused on unique applications of proven supersonic aircraft technology
• Primary technology innovations
  – Supersonic stationary air & gas compressors
  – High velocity combustor
  – Supersonic expander
• Product embodiments
  – Two-stage 100:1 Pr CO2 Compressor
  – 30:1; 42% LHV ASCE Engine
  – Airborne APU
  – H₂ fuel combustor

US Army Corps of Engineers

NETL

Department of Defense

Department of Energy
Ramgen Compressor Technology
Shock Waves to Supersonic Inlets

M₀ = 1.7 Inflow

Oblique Shock Causes Instantaneous Compression

Schlieren Photo of Projectile with Shocks

2-D Mixed Compression Inlet Model

- Initial External Shock System Followed by Internal Shock System
- Throat Bleed Slot For Inlet Starting
- Side Window For Schlieren Photography

Schlieren Photo of Inlet Center-body and Cowl with Shocks
F-15 2-D Planar Supersonic Inlet

\[ M_{rel} = \sim 2 \]

Cowl deflection

Third ramp deflection

Inlet Cross Section

Engine Face
Rampressor Rotor Development

Supersonic F-15 Inlet

M_{rel} = \sim 2

M = \sim 0.3 - 0.5

Rampressor Rotor

M_{rel} = \sim 2

M < 1

M = \sim 0.5

Stationary Engine Case

Normal Shock

Planar Shocks

Rotor Rim

Compressed Fluid
Typical Rotating Supersonic Flow Path

• Rotor Flow Path:
  – 3 Supersonic Compression Inlet
  Flow Paths On Disk Rim
  – High Efficiency, Compact
  Compression
  – Minimal Number of Leading Edges
  – Flow Path Geometry Similar For
  Different Pressure Ratios

• Combination of Supersonic Flight
Inlet & Conventional Axial Flow
Compressor Aerodynamics:
  – Rotor Rim Radius Change Produces
  Compression
  – 3 “Blades” (Strakes) Do Minimal
  Flow Work
  – Axial Inflow/Outflow
Compression Applications vs. Pr/Tip Speed

- Technology Development
  - Rampressor II
  - DoD FC/GT Hybrid
  - Rampressor 1

- Gas Turbines
- Air Compression
- H2O Vapor Compression
- Large HVAC Chiller R-134
  - Single Stage Compressor
- Microturbines (34%)
- Turbochargers
- Rampressor Equipped Microturbines (40%)

Tip Speed (ft/sec) vs. Pressure Ratio

- Air
- CO2
- H2O

- Mechanical Limit
- Start of Aero Limits

- 1-Stage 7:1
- 2-Stage 100 bar
- 2-Stage 200 bar
- 3-Stage 200 bar

Product Development
Enter Dresser-Rand
Dresser-Rand Invests in Ramgen

- Dresser-Rand invests in Ramgen’s “game-changing technology”
  - Support on-going CO2 compressor development
  - Satisfy DOE matching funds requirement
  - Consistent with strategy to be technology leader in our industry
  - Extend served market into Electric Utility industry
  - Invest up to $49 million
    - Fund development & demonstration
    - Obtain an option to purchase assets

Dresser-Rand is consistently ranked among top three manufacturers in its served markets
- Turbomachinery
- Reciprocating compressors
- Steam turbines

• #1 in North America
• Leading supplier of CO2 compressors
• Global sales & service presence
• Strong products & brands
• Established customer base
Dresser-Rand Historical Overview

- Turbodyne formed
- 1985 Dresser Industries acquires.
- 1985 KDP
- 1994 GE’s Navy Business
- Acquires Whiton
- Acquires Khunert
- Tuthill Corp.
- Tuthill Energy Systems
- Gimpel

D-R Services, LLC acquires all stock of Arrow Ind., Inc.
First Reserve Corp purchases D-R (2004), IPO completed (2005)
D-R affiliate acquires assets of Enginuity LLC
D-R acquires Gimpel Valves
D-R acquires Coppus, Murray and Nadowski assets.
D-R acquires certain Peter Brotherhood assets.
Dresser-Rand Heritage

Serving the energy markets since 1840
Dresser-Rand’s Global Presence
Dresser-Rand Key Clients

Note: Partial list as of December 2007.
Products for All Served Markets
World Class Test Facilities
Ramgen CO₂ Compressor Product

- **100:1 CO₂ compressor ⇒ 2-casings/2-stages/Intercooled**
  - No aero Mach# limit
  - 10+:1 pressure ratio; 400°F temperature rise
  - 1400 fps tip speeds; Shrouded rotor design

- **Single-stage, discrete-drive**
  - Single stage per drive optimizes specific speed match
  - Simple single-step external gearbox or high speed direct drive
  - Lower mechanical losses

- **Variable speed option**
  - Match MW and temperature changes with speed changes

- **Configuration adapts easily to match process requirements**
  - Mismatched thru-flow
  - Side stream additions

- **Active IGV Flow control on each stage**
  - Match CO₂ capture system constant pressure requirement

- **Heat exchangers**
  - Inter/aftercooler can be the CCS or power plant
  - “Compressor” heat exchanger cost can be eliminated
  - Eliminate or substantially reduce cooling tower requirement
  - Eliminate or substantially reduce cooling tower make-up water
  - 3x LMTD ⇒ heat exchangers with 1/3 the surface area

- **1/10th the physical size – facilitate space constrained retrofits**
- **1/2 the installation cost**
can be handled with sufficient accuracy for most purposes when the unit is a typical single-stage air compressor. A little more discretion must be used on multistage compressors handling heavy gases, however, because fan-law deviation can become quite significant for speed changes as small as 10 per cent.

**Choke Effect**

The basic slope of the head curve has been discussed at some length, but the choke or stonewall effect that occurs at flows higher than design flow and which must be superimposed upon the basic slope (Fig. 11.19) has not yet been discussed.

Just as basic slope is controlled by impeller-tip vector geometry, the stonewall effect is normally controlled by impeller-inlet vector geometry. In Fig. 11.24, vector $U_i$ may be drawn to represent the tangential velocity of the leading edge of the blade, similar to that in Fig. 11.23. The radial distance of the impeller relative to the center is drawn at $V$. At design flow, the magnitude of $V$ is shown.

**Mach Number Considerations**

The magnitude of $V_i$, compared to the speed of sound at the inlet pressure and temperature is called the relative inlet Mach number. It is the magnitude of this ratio that indicates stonewall effect in a conventional stage. While true stonewall effect should theoretically not be reached until the relative inlet Mach number is unity, it is conventional practice to limit the Mach number to 0.85 or 0.90 at design flow. It is evident from Fig. 11.24 that, for a given rpm, the magnitude of $V_i$ will diminish with decreasing flow, since $V$ is proportional to flow. If $V_i$ decreases, then relative inlet Mach number decreases, so the stonewall effect is normally not a factor at flows below design flow. It is also evident that at low flows the direction of $V_i$ is such that the gas impinges on the leading edge of the blade, resulting in positive incidence, a factor of positive incidence.

Let us now discuss $V_i$ and relative Mach number. High values of $V_i$ and Mach number so also have the trailing edge removed that high degrees of negative incidence tend to contribute to the stonewall problem as Mach number 1.00 is approached, presumably because of boundary layer separation and reduction of effective flow area in the blade pack.

**Significance of Gas Weight**

Since values of $U_i$ are typically in the 500-fps (152.4-m/second) range and values of $V$ in the 250-fps (76.2-m/second) range, it is obvious that, since the speed of sound for air at 80 deg. F (26.7 deg. C) is 1140 fps (348 m/second), lighter gases suffer no true impeller stonewall problems as described, even at high loads. Some head loss below the basic slope will be observed, however, in even the lightest gases, due in part to increased frictional losses throughout the entire stage and in part to the extreme negative incidence at high loads.

The lightest common gas handled by conventional centrifugal compressors for which stonewall effect can be a definite factor is propylene with a sonic speed of 740 fps (225.7 m/second) at $-40$ deg. F $(-40$ deg. C). In order of increasing severity are propane at 718 fps (219 m/second) at $-40$ deg. F $(-40$ deg. C), butane at 630 fps (192.1 m/second) at $-20$ deg. F $(-29$ deg. C), chlorine, and the various Freons. The traditional method of handling such gases is to use an impeller of larger than normal flow area to reduce $V_i$ and run it at lower than normal rpm to reduce $U_i$, thus keeping the value of $V_i$ abnormally low. This procedure requires the use of more than the usual number of stages for a given head requirement and sometimes even requires the use of an abnormally large frame for the flow handled.

**Inducer Impeller Increases Head Output**

Much development work has been done in recent years toward the goal of running impellers at normal speeds on heavy gases in order to reduce hardware costs to those incurred in the compression of light gases. One approach has been to use inducer impellers (Fig. 11.25). The blades on this impeller extend down around the hub radius so that the gas first encounters the blade pack while flowing axially. Figure 11.25 shows the vector analysis at the inducer outer radius. Assuming that the inducer radius is the same as the leading edge radius of the conventional radial inlet impeller, the vector geometries of the two are identical.

The advantage of the inducer lies in the fact that, as we move radially inward along the blade leading edge, the value of $U_i$, and therefore of $V_i$ and Mach number, decreases. As we move along the leading edge of the conventional impeller, the vector geometry remains essentially constant. It can be seen, then, that while maximum Mach number for the two styles is the same, the average Mach number for the inducer
Technology Development Needs & Direction
Fossil Fuel Power Plant – CC&S

- All fossil fuel power plants produce some level of CO2
  - CO2 compressor power
    - Advanced pulverize coal – 8-12%
      - 600MW ⇒ 70MW ⇒ 93,000 hp
    - IGCC - 5%
      - 600MW ⇒ 30MW ⇒ 40,000 hp
    - CCGT – 8%
      - 400MW ⇒ 32MW ⇒ 43,000 hp
- 100 new power plants annually
  - $1.5 billion annual compressor market
- Retrofit opportunity
  - $0.7 billion annual compressor market

Over $2 Billion annual market opportunity
CCS Technologies

• Amine systems
  – Suction pressures – 15; 22; 25; 30 psia
  – Regeneration heat required
    ▪ Conventional amines – 1550 Btu/lbm-CO2
    ▪ Advanced amines – 1200 Btu/lbm-CO2
    ▪ Really advanced amines – 800 Btu/lbm-CO2
  – 8% parasitic power
  – Post combustion - New & Retrofit
• Ammonia-based systems
  – Suction pressures – ~30-300 psia
  – Regeneration heat required
    ▪ Aqueous ammonia – 493 Btu/lbm-CO2
    ▪ Chilled ammonia – TBD
  – 4% parasitic power
  – Post combustion - New & Retrofit
• Chemical Looping
  – Suction pressure atmospheric
• Selexol/Rectisol
  – Suction pressures 50, 150 & 300 psia with sidestreams
  – Regeneration heat required for the Claus Plant
  – 5% parasitic power
  – IGCC (new) only
• Oxy-fuel systems
  – Raw gas feed – 15 to 500 psia
  – Twin purified suction streams – ~150 & 300 psia
  – 12-13% parasitic power
  – New plants only
• Membrane Separation & Enzyme Processes
  – Suction pressures from <3.0-14.7 psia
• Discharge pressures – 1200; 1600; 2000; 2215; 2500; 2700; 2900 psia
Baseline Case for Comparison

Data Provided
- Case 3 ASME TurboExpo Berlin - June 2008
- Case 12 in the Baseline Cost & Performance Study – May 2007
- Compressor 6-stage integrally geared design
- 84% isentropic efficiency all stages
- Inlet conditions 23.52 psia; 69°F inlet temperature; 92.4% RH
- Discharge conditions 2215 psia
- Cooling water 60°F
- Stage pressures
- 1,259,600 lbm/hr
- 2 units

Assumptions
- Intercooler approach temperature 9°F
- Interstage pressure drop DP = (P2^0.7)/10 ; but not greater than 5 psi
- Mechanical loss 1.5%
- Drying between stages 3 & 4
- Partial cooling between stages 5 & 6
- 46,900kW Published (2 unit total)
- 46,898kW Calculated with these assumptions

<table>
<thead>
<tr>
<th>Stage</th>
<th>1</th>
<th>2</th>
<th>3</th>
<th>4</th>
<th>5</th>
<th>6</th>
</tr>
</thead>
<tbody>
<tr>
<td>P1 - psia</td>
<td>23.52</td>
<td>52.00</td>
<td>113.01</td>
<td>248.00</td>
<td>545.00</td>
<td>1200.00</td>
</tr>
<tr>
<td>T1 - °F</td>
<td>69</td>
<td>69</td>
<td>69</td>
<td>69</td>
<td>69</td>
<td>100</td>
</tr>
<tr>
<td>P2 - psia</td>
<td>53.65</td>
<td>115.80</td>
<td>253.00</td>
<td>550.00</td>
<td>1205.00</td>
<td>2219.99</td>
</tr>
<tr>
<td>Pr</td>
<td>2.23</td>
<td>2.28</td>
<td>2.24</td>
<td>2.22</td>
<td>2.21</td>
<td>1.85</td>
</tr>
</tbody>
</table>

Baseline case needs realistic assumptions
It’s No Fun Being Overlooked!

Carbon Dioxide Capture, Compression and Geologic Storage
A CORE ELEMENT OF A GLOBAL ENERGY TECHNOLOGY STRATEGY TO ADDRESS CLIMATE CHANGE

A TECHNOLOGY REPORT FROM THE SECOND PHASE OF THE GLOBAL ENERGY TECHNOLOGY STRATEGY PROGRAM

GTSP
Immaculate Compression

Sequestration
Compressor Power & Things That Affect It

- The basic inputs
  - Gas composition, including moisture content
  - Mass flow
  - Inlet pressure
  - Inlet temperature
  - Discharge pressure

- Often forgotten
  - Cooling media & temperature
    - Air
    - Water-cooled
    - Process cooled
  - Interstage assumptions
    - Pressure drop
    - Design practice
    - Fluor estimate $\Delta P = P_2^{0.7}/10$; not to exceed 5 psi
    - Intercooler/heat exchanger approach temperature or Cold Temperature Difference – CTD
    - 15°F CTD normal approach temperature
  - Mechanical losses
    - Compressor
    - Gearbox
  - Sparing philosophy (i.e., $2 \times 50\% + 1$)

Only the first stage is affected by the inlet conditions…..all the other stages are affected by interstage assumptions.

- CCS Application Specific Issues
  - Capture system flash levels & control requirements
    - Pressure
    - Mass flow additions
  - Water knockout
    - Process location (i.e., pressure)
    - Method – Glycol/Molecular sieve/PSA
  - CO2 compressor inlet pressure
  - Heat integration
  - Materials of construction
    - Heat exchangers
    - Piping
  - Discharge pressure
Heat Exchangers are a Big Deal!
Retrofit Capture Cost Assumptions

• "Carbon Dioxide Capture from Existing Coal-Fired Power Plants"
  – DOE/NETL 401/110907 – Revised November 2007
  – AEP/Alstom Conesville Unit #5
  – Base line & Case 1

• Process Conditions
  – P1 19 psia
  – T1 115 F
  – P2 2015 psia
  – Illinois #6 @ 1.80/mmBtu
  – 90% capture
  – 85% capacity factor

• Financial Assumptions
  – Make-up power 6.4 cents/kWh
  – Burden rate 2.28

• Baseline Compressor Horsepower
  – CO2 compressor 31,262
  – Propane refrigeration 23,321
  – CO2 product pump 2,932
  – Total 57,515 hp
  – Compressor only equivalent 56,800
  – Analysis 56,800 hp
Conventional CO₂ Compression

• CO₂ compressor power
  – Advanced pulverize coal – 9.1%
  – 463MW ⇒ 42MW ⇒ 56,800 hp

• Capital Cost for 56,800 hp
  – 2 x 50% operating units @ $1000/hp 57
  – 1 x 50% spare 28
  – Burdened Installation cost 109
  – Total Cost $194M
  – $194M/303MW = $640/kW

• Cost of Electricity (COE)
  – Baseline w/o CCS 6.07
  – Capture system 4.74
  – Compressor 2.70
  – Total cents/kWh 13.51
  – Increase in COE for CCS 122%

• Cost per tonne
  – Capture system 41
  – Compressor 23
  – Total $64

Compression Costs are 36% of Total Cost/Mt of CO₂

• Heat recovery – Btu/lbm-CO₂
  – Regeneration Heat 1548
  – Heat recovery 0
    Net Btu/lbm-CO₂ 1548

• Plant output
  – Original rating 463
  – De-rating @ 1548 Btu/lbm 160
  – Net 303 MW
  – Value @ 6.4 cents/kWh $62M/year

Figure ES-3: Plant Performance Impact of Retrofitting a Pulverized Coal-Fired Plant at Various Levels of Carbon Capture
Ramgen CO₂ Compression w/Advanced CCS

• CO2 compressor power
  – Advanced pulverize coal – 4.2%
  – 463MW ⇒ 20MW ⇒ 26,000 hp

• Capital Cost for 26,000 hp
  – 2 x 50% operating units @ $400/hp  11
  – 1 x 50% spare  5
  – Installation cost  20
  – Total Cost  $36M
  – $36M/388MW = $93/kW

• Cost of Electricity (COE)
  – Baseline w/o CCS  6.07
  – Capture system  2.02
  – Compressor  0.47
  – Total cents/kWh  8.56
  – Increase in COE for CCS  41%

• Cost per tonne
  – Capture system  22
  – Compressor  5
  – Total  $28

• Heat recovery – Btu/lbm-CO2
  – Regeneration Heat  450
  – Heat recovery @ 230F  93
    Net Btu/lbm-CO2  357
  – HR potential @ 100F  87

• Plant output
  – Original rating  463
  – De-rating @ 450 Btu/lbm  75
    Net  388 MW
  – Value @ 6.4 cents/kWh  $22M/year

CC&S cost can be reduced by 56% from $64 to $28/tonne CO₂
PT Diagram & Supercritical Phase

- Compression process transitions from superheated to supercritical phases
- Avoids liquid (sub-cooled) phase

Separate Phases
Visible-Meniscus Clearly Observed

Increase in Temperature-Diminished Meniscus

Further Increase in Temperature-Gas & Liquid Densities more Similar

At Critical P & T-Distinct Gas & Liquid Phases no Longer Visible “Supercritical Fluid” with Properties of Both Liquids & Gases

![PT Diagram & Supercritical Phase](image)
Ramgen Heat Recovery

<table>
<thead>
<tr>
<th></th>
<th>Low Pressure Stage</th>
<th>High Pressure Stage</th>
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<tbody>
<tr>
<td></td>
<td>22 - 220 psia</td>
<td>220 - 2200 psia</td>
</tr>
<tr>
<td>Compressor Shaft Input Work</td>
<td>90.6 Btu/lbm</td>
<td>87.0 Btu/lbm</td>
</tr>
<tr>
<td>Discharge Temperature</td>
<td>489 °F</td>
<td>509 °F</td>
</tr>
<tr>
<td>Lower Recovery Temperature</td>
<td>100 °F</td>
<td>100 °F</td>
</tr>
<tr>
<td>Recovered Heat</td>
<td>92.4 Btu/lbm</td>
<td>178.8 Btu/lbm</td>
</tr>
<tr>
<td>Recovered Heat/Compression Work</td>
<td>102%</td>
<td>205%</td>
</tr>
</tbody>
</table>

- Heat available in the HP hot discharge CO2 is more than double the compressor shaft work
- 153% of the combined LP + HP shaft work is available as heat in the discharge CO2
Optimizing Compressor Selection

Adiabatic Head vs. icfm

Flow Cut to Match Capacity

Change Speed to Match Design Head

Frame Sizes

icfm (000)

Adiabatic Head $h$ ft-lbf/lbm

Constant Specific Speed

$\eta = \Delta h \times \eta_{ad} \times 778 \text{ ft-lbf / Btu}$

$Head_{ad} = ZRT \times \frac{k}{k-1} \times \left( \frac{P_2}{P_1} \right)^{\frac{k-1}{k}} - 1$

$P_2 = 2215$

$P_2 = 1615$

$P_2 = 1215$
## “The Convenient Half-Truth”

<table>
<thead>
<tr>
<th></th>
<th>PC</th>
<th>SCPC</th>
<th>IGCC</th>
<th>NGCC</th>
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<tbody>
<tr>
<td></td>
<td>w/out</td>
<td>with</td>
<td>w/out</td>
<td>with</td>
</tr>
<tr>
<td><strong>Gross Power</strong></td>
<td>583,315</td>
<td>679,923</td>
<td>580,260</td>
<td>663,445</td>
</tr>
<tr>
<td><strong>Net Power</strong></td>
<td>550,445</td>
<td>549,613</td>
<td>550,150</td>
<td>545,995</td>
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<tr>
<td><strong>Coal Flowrate - lbm/hr</strong></td>
<td>437,699</td>
<td>646,589</td>
<td>411,282</td>
<td>586,627</td>
</tr>
<tr>
<td><strong>Natural Gas Flowrate - lbm/hr</strong></td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td><strong>Net Plant Heat Rate - Btu/kW-hr</strong></td>
<td>9276</td>
<td>13724</td>
<td>8721</td>
<td>12534</td>
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<tr>
<td><strong>Net Plant Efficiency - HHV%</strong></td>
<td>36.8%</td>
<td>24.9%</td>
<td>39.1%</td>
<td>27.2%</td>
</tr>
<tr>
<td><strong>Carbon Factor - lbm-CO2/mmBtu</strong></td>
<td>203.3</td>
<td>203.3</td>
<td>203.3</td>
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<tr>
<td><strong>Capacity Factor</strong></td>
<td>85.0%</td>
<td>85.0%</td>
<td>85.0%</td>
<td>85.0%</td>
</tr>
<tr>
<td><strong>Capture %</strong></td>
<td>0.0%</td>
<td>90.0%</td>
<td>0.0%</td>
<td>90.0%</td>
</tr>
<tr>
<td><strong>Capital Cost - $/kW</strong></td>
<td>$1,549</td>
<td>$2,895</td>
<td>$1,575</td>
<td>$2,870</td>
</tr>
<tr>
<td><strong>LCOE - $/kW-hr</strong></td>
<td>$0.0640</td>
<td>$0.1188</td>
<td>$0.0633</td>
<td>$0.1148</td>
</tr>
<tr>
<td><strong>CO2 lbm/MW-hr Net Output</strong></td>
<td>1886</td>
<td>278</td>
<td>1773</td>
<td>254</td>
</tr>
</tbody>
</table>

**Capture % to Achieve 797 or 278 lbm/MW-hr**

75.7%  71.4%  55.0%  68.7%  54.6%  61.4%  0.0%  70.0%

797  797  797  797  797  797  797  278

**Note:** Baseline Report Cases 1 & 2

\[
\text{tons/ year} = \left(\text{power}_{\text{net}} \times 8760 \times \text{capacity factor} \times \text{heat rate}_{\text{net}} \times \text{carbon factor}\right)/10^6
\]

\[
\text{CO}_2 \text{ lbm/MWh}_{\text{net}} = \text{heat rate}_{\text{net}} \times \text{carbon factor} \times \left(1 - \text{capture\%}\right)/10^3
\]

NETL Cost & Performance Baseline

NETL May 2007
Technology Development Needs

Compressor System
- Compressor
- Drives
  - High power 2-pole motor
  - High power VFD’s
  - Steam turbine drives & control
- Gearboxes
  - Industry capacity
  - Auxiliary drive
- Coolers - conventional service
  - Air-cooled
  - Water-cooled
- Heat Recovery Coolers
  - Boiler feedwater
  - Solvent regeneration
  - Coal drying
  - Air pre-heater
  - Flue gas re-heating

Capture System
- Improved solvents
  - Higher loading
  - Reduced regeneration heat
  - Improved thermal stability
  - Lower regeneration temperatures
  - Lower cost
  - Faster reaction kinetics
  - High pressure CO₂

Design & Analysis Tools
- NIST REFPROP CO₂ Mixtures with:
  - Water
  - CO
  - Argon
  - Oxygen
  - Ammonia
  - Hydrogen
  - Heat exchangers for supercritical fluids
  - Impurities & phase change models
  - Generic capture system modeling capabilities – (Excel & ASPEN)
  - Installed first cost & operating cost models
  - Materials selection guidance
Questions?

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