Advanced Centrifugal Compression and Pumping for CO$_2$ Applications

Southwest Research Institute Team:
J. Jeffrey Moore, Ph.D.
Neal Evans
Andrew Lerche
Timothy Allison, Ph.D.
Brian Moreland

Dresser-Rand Team:
Jorge Pacheco, Ph.D.
Jason Kerth
William Egan

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DOE PM: Timothy Fout
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 Ft²
- 3200 Employees
- 1200 Engineers
- 170 Buildings
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)**
  - 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 2\(^{nd}\) Qtr 2011)**
  - Pilot scale compression plant
  - 55,000 lbm/hr
• Only CO\textsubscript{2} stream considered
Proposed Solution for Optimal Efficiency

Optimal solution combines inter-stage cooling and a liquefaction approach.
## Updated Thermodynamic Calculations

### Results for 4 input streams, IGCC power plant application

<table>
<thead>
<tr>
<th>Option</th>
<th>Compression Technology</th>
<th>Power Requirements</th>
<th>% Diff from Option A</th>
<th>Cooling Technology</th>
<th>Cooling Requirements</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>A</strong></td>
<td>Conventional Dresser-Rand Centrifugal 16-stage Compression</td>
<td>23,251 BHP</td>
<td>0.0%</td>
<td>Air-cool streams between separate stages</td>
<td>Air Mass Flow = 2.03e6 lbm/hr</td>
</tr>
<tr>
<td><strong>C.7</strong></td>
<td>Semi-isothermal compression at 100 degF, Pressure Ratio ~ 1.55</td>
<td>17,979 BHP (Required Cooling Power TBD)</td>
<td>-22.7%</td>
<td>Tc = 100degF in between each stage.</td>
<td>To be determined</td>
</tr>
<tr>
<td><strong>E.1 Old</strong></td>
<td>Centrifugal compression to 250 psia, Liquid cryo-pump from 250-2215 psia</td>
<td>17,055 BHP (Includes 7,814 BHP for Refrigeration)</td>
<td>-26.6%</td>
<td>Air cool up to 250 psia, Refrigeration to reduce CO2 to -25degF to liquefy</td>
<td>Refrigeration requires 7814 HP for 3428 tons, Air Mass Flow = 6.3e5 lbm/hr</td>
</tr>
<tr>
<td><strong>E.2 Old</strong></td>
<td>Centrifugal compression to 250 psia with semi-isothermal cooling at 100 degF, Liquid cryo-pump from 250-2215 psia</td>
<td>16,001 BHP (Includes 7,814 BHP for Refrigeration)</td>
<td>-31.2%</td>
<td>Air cool up to 250 psia between centrifugal stages, Refrigeration to reduce CO2 to -25degF to liquefy</td>
<td>Refrigeration requires 7814 HP for 3428 tons, Air Mass Flow = 5.1e5 lbm/hr</td>
</tr>
<tr>
<td><strong>E.1 Updated</strong></td>
<td>Centrifugal compression to 250 psia, Liquid cryo-pump from 250-2215 psia, Measured pump eff. No N2 Cooling, Updated cooling cost to 1.582 kW/ton</td>
<td>22,721 BHP (Includes 13,480 BHP for Refrigeration)</td>
<td>-2.3%</td>
<td>Air cool up to 250 psia, Refrigeration to reduce CO2 to -25degF to liquefy</td>
<td>Refrigeration requires 13480 HP for 6354 tons.</td>
</tr>
<tr>
<td><strong>E.2 Updated</strong></td>
<td>Centrifugal compression to 250 psia with semi-isothermal cooling at 100 degF, Liquid cryo-pump from 250-2215 psia, Measured pump eff. No N2 Cooling, Updated cooling cost to 1.582 kW/ton</td>
<td>21,667 BHP (Includes 13,470 BHP for Refrigeration)</td>
<td>-6.8%</td>
<td>Air cool up to 250 psia between centrifugal stages, Refrigeration to reduce CO2 to -25degF to liquefy</td>
<td>Refrigeration requires 13480 HP for 6354 tons.</td>
</tr>
</tbody>
</table>
Challenges: High Reliability

**Integrally Geared Isothermal Compressor**
- 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**
- 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

* Courtesy of MAN
* Courtesy of Dresser-Rand

Southwest Research Institute
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 CO2 pump
Internally Cooled Compressor Concept

- Investigate an internally-cooled compressor concept
  - Red - CO₂ flow path through compressor stage
  - Blue - Liquid cooling in the diaphragm
  - Grey - Solid

Courtesy of Dresser-Rand
Conjugate Heat Transfer CFD Model

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

Adiabatic Results (No cooling)

<table>
<thead>
<tr>
<th></th>
<th>OEM Data</th>
<th>Model</th>
<th>(%) Difference</th>
</tr>
</thead>
<tbody>
<tr>
<td>Total Pressure Ratio</td>
<td>1.550</td>
<td>1.648</td>
<td>6.3</td>
</tr>
<tr>
<td>Total Temperature Ratio</td>
<td>1.136</td>
<td>1.139</td>
<td>0.3</td>
</tr>
<tr>
<td>Gas Power [HP]</td>
<td>102.0</td>
<td>104.3</td>
<td>2.3</td>
</tr>
</tbody>
</table>

Flow Boundary Conditions for Cooling Fluid

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
### CFD Results of Adiabatic and Conjugate Heat Transfer Models

<table>
<thead>
<tr>
<th>Model</th>
<th>Quantity</th>
<th>Impeller Ratio</th>
<th>Stage Ratio</th>
</tr>
</thead>
<tbody>
<tr>
<td>Adiabatic</td>
<td>Total Pressure</td>
<td>1.773</td>
<td>1.670</td>
</tr>
<tr>
<td></td>
<td>Total Temperature</td>
<td>1.142</td>
<td>1.142</td>
</tr>
<tr>
<td>Diabatic with Heat Transfer Coefficients</td>
<td>Total Pressure</td>
<td>1.764</td>
<td>1.671</td>
</tr>
<tr>
<td></td>
<td>Total Temperature</td>
<td>1.141</td>
<td>1.116</td>
</tr>
<tr>
<td>Diabatic with Full Conjugate Heat Transfer</td>
<td>Total Pressure</td>
<td>1.767</td>
<td>1.678</td>
</tr>
<tr>
<td></td>
<td>Total Temperature</td>
<td>1.141</td>
<td>1.117</td>
</tr>
</tbody>
</table>

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

Dimpled Walls

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

Grooved Airfoil Surface

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

Ribs on Walls
Analysis of Design Configurations

- **Adiabatic** – No heat transfer from CO₂, serves as the baseline for other cases.
- **Smooth wall (SW) heat transfer** – Smooth walls on both the water and CO₂ 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 CO₂ side walls** – A convection coefficient augmentation case.
- **Ribbed water side walls, dimpled CO₂ side walls, and grooved airfoils** – The second convection coefficient augmentation case.
Updated Estimates – Low P

• Comparison with previous straight-through estimates
  – Same stage P1 & P2, first stage T1, efficiencies, and HX effectiveness values
  – Gas properties from REFPROP

<table>
<thead>
<tr>
<th>Geometry</th>
<th>RPM</th>
<th>Radius Ratio</th>
<th># Stages</th>
<th>HX Effectiveness</th>
<th>Gas Power Savings</th>
</tr>
</thead>
<tbody>
<tr>
<td>Adiabatic Reference</td>
<td>12850</td>
<td>1.5</td>
<td>5</td>
<td>NA</td>
<td>0%</td>
</tr>
<tr>
<td>Smooth Wall</td>
<td>12850</td>
<td>1.5</td>
<td>5</td>
<td>0.15</td>
<td>7.0%</td>
</tr>
<tr>
<td>Smooth Wall</td>
<td>12850</td>
<td>1.8</td>
<td>5</td>
<td>0.197</td>
<td>8.6%</td>
</tr>
<tr>
<td>Ribs and Dimples</td>
<td>12850</td>
<td>1.5</td>
<td>5</td>
<td>0.25</td>
<td>1.2%</td>
</tr>
<tr>
<td>Ribs, Dimples, and Grooves</td>
<td>12850</td>
<td>1.5</td>
<td>5</td>
<td>0.31</td>
<td>-0.93%</td>
</tr>
<tr>
<td>Adiabatic Reference</td>
<td>9155</td>
<td>1.5</td>
<td>9</td>
<td>NA</td>
<td>0%</td>
</tr>
<tr>
<td>Smooth Wall</td>
<td>9155</td>
<td>1.5</td>
<td>9</td>
<td>0.15</td>
<td>13.3%</td>
</tr>
<tr>
<td>Smooth Wall</td>
<td>9155</td>
<td>1.8</td>
<td>9</td>
<td>0.197</td>
<td>15.3%</td>
</tr>
</tbody>
</table>
Test Rig Construction

- Diffuser side of bulb
- Main structural section (diffuser side)
- Removable lid
- Main structural section (return channel side)
- Return channel side of bulb
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

- 28 Temperature Probes
- 30 Pressure Measurements
- Flow Rate (CO₂ and Cooling)
- Speed
- Shaft Torque
- Axial Thrust
- Gas Samples Taken
Main Screen of Data Acquisition Code
Some Definitions

• Heat Exchanger Effectiveness

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

where

\[ \dot{Q} = c_{H_2O}(T_{H_2O,\text{out}} - T_{H_2O,\text{in}}) = c_{CO_2}(T_{CO_2,\text{in}} - T_{CO_2,\text{out}}) \]

\[ \dot{Q}_{\text{max}} = c_{\text{min}}(T_{CO_2,\text{in}} - T_{H_2O,\text{in}}) \]
Measured Polytropic Head vs. Flow
30-90 psia (2-6 bar) Suction Pressure

Normalized Head vs. Normalized Flow

Actual 10280 rpm 30 psia Adiabatic
Actual 10280 rpm 30 psia Diabatic 65 deg F
Actual 10280 rpm, 30 psia Diabatic 50 deg F
Actual 11565 rpm, 30 psia Adiabatic
Actual 11565 rpm, 30 psia Diabatic 65 deg F
Actual 11565 rpm, 30 psia Diabatic 50 deg F
Actual 11565 rpm, 60 psia Adiabatic
Actual 11565 rpm, 60 psia Diabatic 65 deg F
Actual 11565 rpm, 90 psia Diabatic 76 deg F
Actual 12850 rpm, 30 psia Adiabatic
Actual 12850 rpm, 30 psia Adiabatic 2nd Try
Actual 12850 rpm, 30 psia Diabatic 73 deg F
Actual 12850 rpm, 30 psia Diabatic 63 deg F
Actual 12850 rpm, 60 psia Adiabatic
Actual 12850 rpm, 60 psia Diabatic 65 deg F
Actual 12850 rpm, 90 psia Diabatic 70 deg F
Actual 12850 rpm, 90 psia Adiabatic
Actual 12850 rpm, 90 psia Diabatic 77 deg F
Actual 12850 rpm, 90 psia Diabatic 77 deg F 20 gpm
Actual 12850 rpm, 60 psia Diabatic 77 deg F 20 gpm
Measured Total Temperature Profiles

Normalized Temperature Throughout Stage

Suction
Bridgeover

Impeller
Exit

Diffuser
Vane Exit

Return
Channel Bend

Discharge
Bridgeover

(Total Temperature) / (Impeller Discharge Total Temperature)

10280 rpm, 30 psia Adiabatic
10280 rpm, 30 psia Diabatic 65 deg F
10280 rpm, 30 psia Diabatic 50 deg F

11565 rpm, 30 psia Adiabatic
11565 rpm, 30 psia Diabatic 65 deg F
11565 rpm, 30 psia Diabatic 50 deg F

12850 rpm, 30 psia Adiabatic
12850 rpm, 30 psia Diabatic 73 deg F
12850 rpm, 30 psia Diabatic 70 deg F
12850 rpm, 30 psia Diabatic 77 deg F 20 gpm
12850 rpm, 90 psia Diabatic 77 deg F 20 gpm
Measured Heat Exchanger Effectiveness vs. Flow at 30 psia Suction Pressure

- Actual 10280 rpm, 30 psia Diabatic 65 deg F
- Actual 11565 rpm, 30 psia Diabatic 65 deg F
- Actual 12850 rpm, 30 psia Diabatic 65 deg F
- Actual 10280 rpm, 30 psia Diabatic 50 deg F
- Actual 11565 rpm, 30 psia Diabatic 50 deg F
- Actual 12850 rpm, 30 psia Diabatic 63 deg F
Measured Heat Exchanger Effectiveness vs. Flow
30-90 psia (2-6 bar) Suction Pressure

Normalized Flow
Heat Exchanger Effectiveness vs. Cooling Flow Rate

- 12850 rpm, 60 psia Diabatic 77 deg
- 12850 rpm, 90 psia Diabatic 78 deg
Heat Exchanger Effectiveness vs. Suction Pressure

- CO2 is Cmin
- H2O is Cmin

Suction Pressure (psia)

Heat Exchanger Effectiveness

- Q/Qdesign = 1.12
- Q/Qdesign = 1.0
- Q/Qdesign = 0.87
- Q/Qdesign = 0.74
Fraction of Heat Removal in the Stage vs. Flow

\[ TRR = \frac{\Delta T_{\text{Cooled Temp Reduction}}}{\Delta T_{\text{Adiabatic Stage Temp Rise}}} \]

- Actual 10280 rpm, 30 psia Diabatic 65 deg F
- Actual 11565 rpm, 30 psia Diabatic 65 deg F
- Actual 12850 rpm, 30 psia Diabatic 73 deg F
- Actual 10280 rpm, 30 psia Diabatic 50 deg F
- Actual 11565 rpm, 30 psia Diabatic 50 deg F
- Actual 11565 rpm, 60 psia Diabatic 65 deg F
- Actual 12850 rpm, 60 psia Diabatic 70 deg F
- Actual 12850 rpm, 90 psia Diabatic 77 deg F
- Actual 12850 rpm, 30 psia Diabatic 63 deg F
- 12850 rpm, 60 psia Diabatic 77 deg F 20 gpm
Fraction of Heat Removal in the Stage vs. Impeller Exit Temperature
Comparison to CFD Predictions

Normalized Temperature Throughout Stage

- **12850 rpm, 30 psia Adiabatic 2nd try**
- **12850 rpm, 30 psia Diabatic 73 deg F 12 gpm**
- **CFD 12850 rpm, 30 psia Adiabatic**
- **CFD 12850 rpm, 30 psia Diabatic 70 deg F 20 gpm**
Comparison to Predictions

Heat Exchanger Effectiveness vs. Normalized Flow

- Actual 10280 rpm, 30 psia Diabatic 65 deg F
- Actual 11565 rpm, 30 psia Diabatic 65 deg F
- Actual 12850 rpm, 30 psia Diabatic 73 deg F
- CFD 12850 rpm, 30 psia Diabatic 70 deg F 20 gpm
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
- Development of a pilot scale compression facility currently under design
Phase 3 Deliverables

• Deliverables:
  – The cooled diaphragm concept will be extended to a multi-stage design. Many design challenges remain to mature the design for commercialization. Since the cooled diaphragm concept works by reducing the power required in the downstream stages, actual power reduction will be measured.
  – An overall power balance will be measured, including all coolers and chillers.

• Technology will be considered field ready following this demonstration program
Phase 3 Work Breakdown

Year 1 – Cooled Diaphragm and Loop Design
- Finalize compressor selection
- Perform conjugate heat transfer CFD analysis
- Support D-R with FEA analysis of multi-stage diaphragm and cooling circuit
- Develop functional requirement of flow loop including process diagram and P&ID
- Design liquefaction system
- Select major pieces of equipment
- Develop solid model of flow loop
- Perform piping and pressure vessel analysis
- Simulate flow loop using pipeline simulation software
- Generate complete BOM and cost summary
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
- Compressor loop with be mated to liquefaction plant and liquid CO₂ pump
3 MW Compression Facility

- Suction Scrubber
- HX
- Compressor
Compressor Specifications

- Dresser-Rand DATUM D12R6B
- Approximate operating conditions are:
  - Suction Pressure: 15-25 psi
  - Discharge Pressure: 230-260 psi
  - Mass Flow = 55,000-75,000 lbm/hr
  - Power: 3,000 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.
Phase 3 Work Breakdown

Year 2 – Hardware Procurement and Site Preparation

• Compressor Procurement
• Procure all Major Equipment
  – Piping, Valves, Coolers, Liquefaction System, and Vaporizer
• Procure Instrumentation and Develop Data Acquisition and Control Program
• Prepare Site
  – Pour Concrete Pad
  – Install Electrical Supply and Transformer
• Construct Control Room and Laboratory
Year 3 – Test Loop Assembly, Commissioning, and Testing

- Test Loop Assembly
  - Install major pieces of equipment including coolers, heat exchangers, cooling tower and compressor
  - Relocate pump loop to new facility
- Install compressor package including cooling water and lube oil to the coolers.
- Install electrical connections to all equipment
- Install instrumentation on both compressor and pump skids
- Commission compressor loop
- Commission pump loop
- Commission liquefaction plant
- Test fully integrated compression/liquefaction/pumping system
2011 – Design of Multi-Stage Diaphragm and Test Loop
2012 – Hardware Procurement and Site Preparation
2013 – Test Loop Assembly, Commissioning, and Testing

• Work is proceeding on Schedule

• Total Project Budget: $9.86 million
Questions???

www.swri.org

Dr. J. Jeffrey Moore
Southwest Research Institute
(210) 522-5812
Jeff.Moore@swri.org