Advanced Centrifugal Compression and Pumping for CO₂ Applications

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Project Funded by DOE NETL
DOE PM: Mr. Timothy Fout

Co-Funded by Dresser-Rand and BP
Project Motivation

- \( \text{CO}_2 \) 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 \(~600,000 \text{ to } 700,000 \text{ lbm/hr.}\)
- Project goal: Double-digit reduction of compression power for \( \text{CO}_2 \) 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 (Complete in 2010)
  – Pilot testing of two concepts:
    • Isothermal compression (complete)
    • Liquid CO₂ pumping (complete)

• Phase III – Kick off February 2011
  – Pilot plant combining compression, liquefaction, and pumping technology
Only CO$_2$ stream considered
Proposed Solution for Optimal Efficiency

Compression Technology Options for IGCC Waste Carbon Dioxide Streams

Optimal solution combines inter-stage cooling and a liquefaction approach.
## Summary of Thermodynamic Analysis for IGCC Plant

<table>
<thead>
<tr>
<th>Option</th>
<th>Compression Technology</th>
<th>Power Requirements</th>
<th>% Diff from Option A</th>
<th>Cooling Technology</th>
</tr>
</thead>
<tbody>
<tr>
<td>A</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>
</tr>
<tr>
<td>B</td>
<td>Conventional Dresser-Rand Centrifugal 16-stage Compression with additional cooling</td>
<td>21,522 BHP</td>
<td>-7.4%</td>
<td>Air-cool streams between separate stages using ASU cool N2 stream</td>
</tr>
<tr>
<td>C.1</td>
<td>Isothermal compression at 70 degF and 80% efficiency</td>
<td>14,840 BHP</td>
<td>-36.2%</td>
<td>Tc = 70 degF inlet temp throughout</td>
</tr>
<tr>
<td>C.4</td>
<td>Semi-isothermal compression at 70 degF, Pressure Ratio ~ 1.55 (Required Cooling Power TBD)</td>
<td>17,025 BHP</td>
<td>-26.8%</td>
<td>Tc = 70degF in between each stage.</td>
</tr>
<tr>
<td>C.7</td>
<td>Semi-isothermal compression at 100 degF, Pressure Ratio ~ 1.55 (Required Cooling Power TBD)</td>
<td>17,979 BHP</td>
<td>-22.7%</td>
<td>Tc = 100degF in between each stage.</td>
</tr>
</tbody>
</table>
### Summary of Thermodynamic Analysis for IGCC Plant Cont.

<table>
<thead>
<tr>
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</thead>
<tbody>
<tr>
<td>D.3</td>
<td>High ratio compression at 90% efficiency - no inter-stage cooling</td>
<td>34,192 BHP</td>
<td>47.06%</td>
<td>Air cool at 2215 psia only</td>
</tr>
<tr>
<td>D.4</td>
<td>High ratio compression at 90% efficiency - intercooling on final compression stage</td>
<td>24,730 BHP</td>
<td>6.36%</td>
<td>Air cool at 220 and 2215 psia</td>
</tr>
<tr>
<td>E.1</td>
<td>Centrifugal compression to 250 psia, Liquid cryo-pump from 250-2215 psia</td>
<td>16,198 BHP (Includes 7,814 BHP for Refrigeration)</td>
<td>-30.33%</td>
<td>Air cool up to 250 psia, Refrigeration to reduce CO2 to -25degF to liquify</td>
</tr>
<tr>
<td>E.2</td>
<td>Centrifugal compression to 250 psia with semi-isothermal cooling at 100 degF, Liquid cryo-pump from 250-2215 psia</td>
<td>15,145 BHP (Includes 7,814 BHP for Refrigeration)</td>
<td>-34.86%</td>
<td>Air cool up to 250 psia between centrifugal stages, Refrigeration to reduce CO2 to -25degF to liquify</td>
</tr>
</tbody>
</table>

Note: Heat recovery not accounted for.
Liquefaction process

- Utilize a refrigeration system to condense CO\textsubscript{2} at 250 psia and -12\textdegree F.
- Liquid then pumped from 250 to 2,200 psia.
- Requires significantly less power to pump liquid than to compress a gas.
- The cost of the refrigeration system must be accounted for.
Compression Power for PC Plant

Liquefaction/Pumping Compression

Graph showing the total power (HP) against cooling COP for different pressure levels (81.6 PSI, 262.0 PSI, 628.3 PSI). The graph indicates the reference case with a dashed line.
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 or high speed motor

Courtesy of MAN

Courtesy of Dresser-Rand

Southwest Research Institute
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
Phase 2 Project Plan

- Experimentally validate thermodynamic predictions.
- Two test programs envisaged:
  - Liquid CO\textsubscript{2} pumping loop
  - Closed-loop CO\textsubscript{2} compressor test with internal cooling
- Power savings will be quantified in both tests.
• 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.
Case 4- Conjugate heat transfer model with enhanced heat transfer coefficients to simulate ribbed surfaces for the cooling liquid
Final Design

- Conjugate heat transfer model with enhanced heat transfer coefficients to simulate ribbed surfaces for the cooling liquid
- Two radius ratios shown
Summary of CFD Results

Temperature Rise and Pressure Drop of CFD Models

- Inlet to Stage Exit ΔTotal
- Impeller Exit to Stage Exit |ΔPtotal|

| Model Type                          | ΔT [°F] | |ΔP| [psi] |
|-------------------------------------|---------|---|------|
| Adiabatic                           | 90      |   |      |
| Full Conjugate model                | 70      |   |      |
| Smooth liquid-smooth gas            | 50      |   |      |
| Ribbed liquid-dimpled gas, smooth blades | 40    |   |      |
| Ribbed liquid-dimpled gas, grooved blades | 30   |   |      |
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 (CO$_2$ and Cooling)
- Speed
- Shaft Torque
- Axial Thrust
- Gas Samples Taken
Compressor Test Results

Normalized Head vs. Normalized Flow

- Predicted 12850 rpm
- Predicted 11565 rpm
- Predicted 10280 rpm
- 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 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
Compressor Test Results

Normalized Efficiency vs. Normalized Flow

Normalized Polytropic Efficiency

Normalized Flow

Predicted 12850 rpm
Predicted 11565 rpm
Predicted 10280 rpm
Actual 10280 rpm, 30 psia Adiabatic
Actual 11565 rpm, 30 psia Adiabatic
Actual 12850 rpm, 30 psia Adiabatic 2nd Try
Compressor Test Results

Normalized Temperature Throughout Stage

- 10280 rpm, 30 psia Adiabatic
- 10280 rpm, 30 psia Diabatic 65 deg F
- 11565 rpm, 30 psia Adiabatic
- 11565 rpm, 30 psia Diabatic 50 deg F
- 11565 rpm, 60 psia Adiabatic
- 11565 rpm, 60 psia Diabatic 65 deg F
- 12850 rpm, 30 psia Adiabatic
- 12850 rpm, 30 psia Diabatic 2nd try
- 12850 rpm, 30 psia Diabatic 73 deg F
- 12850 rpm, 60 psia Adiabatic
- 12850 rpm, 60 psia Diabatic 70 deg F
- 12850 rpm, 90 psia Diabatic 77 deg F
- 12850 rpm, 60 psia Diabatic 77 deg F 20 gpm
- 12850 rpm, 90 psia Diabatic 78 deg F 20 gpm

Suction Bridgeover
Impeller Exit
Diffuser Vane Exit
Return Channel Bend
Discharge Bridgeover
Fraction of Heat Removal in the Stage

- 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

![Graph showing the relationship between temperature reduction ratio and impeller exit temperature.](#)
Comparison to 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**

Normalized Temperature / (Impeller Discharge Temperature)
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
## Multi-Stage Compressor Example

<table>
<thead>
<tr>
<th>Geometry</th>
<th>RPM</th>
<th>Radius Ratio</th>
<th>Power Savings* (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Adiabatic reference</td>
<td>12850</td>
<td>1.5</td>
<td>0.0</td>
</tr>
<tr>
<td>Smooth wall</td>
<td>12850</td>
<td>1.5</td>
<td>9.5</td>
</tr>
<tr>
<td>Smooth wall</td>
<td>9155</td>
<td>1.5</td>
<td>16.6</td>
</tr>
<tr>
<td>Smooth wall</td>
<td>12850</td>
<td>1.8</td>
<td>12.3</td>
</tr>
</tbody>
</table>

- 5-Stage straight-through compressor
- Suction Pressure = 30 psia, Discharge Pressure = 250 psia
- Uses heat exchanger effectiveness of 0.22 for 1.8 radius ratio
- Savings for the 1.8 radius ratio at the reduced speed of 9155 rpm is expected to approach 20% due to an increased number of stages required.
Cryogenic Turbopump Validation Testing
Liquid CO2 Pumping Pilot Scale Testing

- Testing will measure pump efficiency
- Validate pump design
- Measure NPSH requirements looking for signs of cavitation
- An industrial pump manufacturer supplied the pump
  - 250 KW, 100 gpm, 53,000 lbm/hr
• Vessel layout showing elevated reservoir and knock-out drum

• Pump is mounted at ground level.

• Orifice run located between pump and control valve (in supercritical regime)
Pump Loop Construction
Pump Loop Completed
Data Acquisition Code

DOE Pump Test Rig Software

Performance Results Cluster
- 249.174 Psix [psia]
- -11.8381 Tsix [deg F]
- 2099.88 Pxx [psia]
- 17.6525 Txx [deg F]
- 4151.49 Head [ft]
- 56628.9 Flow [gpm]
- 241.287 Pump Brake hp [hp]
- 6331.25 Pump Hydraulic hp [hp]
- 26.2395 Pump Eff [%]
- 3506.79 Speed [rpm]

Current
- 19.2888 V, 0.0000 A, 0.0000 Hz
- 241.287 RPM

Control Line
- 3500 [rpm]

Snapsots
- 3500 [rpm]
- 2700 [rpm]
- 2500 [rpm]
- 1570 [rpm]

Speed [rpm]
- 3507 [rpm]

Record Snapshot
- 0
- 2

Import Map

Pump Conditions
- Current Value
- Alarm Value
- Trip Value
- 70.7067
- 70.7067
- N/A
- 157.111
- 157.111
- N/A
- 149.746
- 149.746
- 239
- 76.0094
- 76.0094
- 32
- 2.92192
- 2.92192
- 2
- 0.00326
- 0.00326
- N/A
- 249.174
- 249.174
- N/A

Pause Playback
- PAUSE

Replay Loop Time (ms)
- 5000

Replay Point
- 6005

Time
- 17:20:23.32
Test Results

Flow, GPM

Head, ft

Flow, GPM

Test 1 1578 RPM
Test 1 2500 RPM
Test 1 3510 RPM
Test 2 1520 RPM
Test 2 2500 RPM
Test 2 3510 RPM

1578 RPM Predicted
2500 RPM Predicted
3510 RPM Predicted
Dynamic Data – Design Point

Suction Dynamic Pressure

Casing Vibration - X

Casing Vibration - Y
Dynamic Data – Minimum Flow Point

Suction Dynamic Pressure

Subsynchronous Component

Casing Vibration - X

Casing Vibration - Y
Dynamic Suction Pressure Waterfall while Throttling Decreasing Flow
Phase 2 Testing 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 removal improves in latter stages of a multi-stage compressor
  – Over 20% reduction in power is possible for a multi-stage application

• Pump Testing
  – Pump performed match the measured performance during factory testing on LN₂
  – Met discharge pressure goals
  – LCO₂ introduced no mechanical issues for the pump
  – Vibration levels were acceptable
  – A subsynchronous vibration occurred at minimum flow point but only at very low flow rates

• Both Technologies are Ready for Pilot Plant Demonstration
Phase 3 Pilot Test Facility

3 MW Motor

CO₂ Compression System

Cooler

CO₂ Compressor

Cooling Tower

250 psia

Motor

GB

250 psia

2200 psia, 80F

Tank

Motor

CO₂ Pump

250 psia

-10 F

15 psia, 80F

Evaporator

Evaporator

Valve

Valve

Refrigeration Unit
Phase 3 Proposal

• 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.
  – The refrigeration system, including an economizer, will be designed and tested. The actual power required for the refrigeration system will be quantified. The effect of entrained gases found in actual carbon capture and sequestration applications will be tested by injecting nitrogen upstream of the liquefaction process and separating this gas.
  – The system dynamics and interaction between the compressor and the pump will be measured, including required recycle lines.
  – An overall power balance will be measured, including all coolers and chillers.
Questions???

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