Dynamic Modeling and Control of Supercritical CO₂ Power Cycle using Waste Heat from Industrial Process

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Outline

- Introduction
- Contribution of this study
- Aim and objectives
- Description of s-CO$_2$ power cycle
- Dynamic model development
- Results and discussions
- Conclusions
Background and Motivation

• The left-over heat after combustion process, or any chemical or thermal process is known as waste heat as it is usually exhausted to the environment.

• Large amount of waste heat is available for recovery in industrial processes worldwide.

• Significant proportion (up to 50%) of the fuel energy is released directly to the environment (Imran et al., 2016).

• Application of waste heat to power (WHP) technologies can increase the energy efficiency and cut CO₂ emissions from these facilities.
Steam Rankine cycle (SRC) and organic Rankine cycle (ORC) are the commonly deployed WHP system
  - High irreversibility in the heat recovery heat exchanger
  - Restricted heat source temperature range
  - ORC only applicable to heat source temperature below 300 °C
  - SRC has poor efficiency at temperature below 400 °C

Supercritical CO₂ (S-CO₂) power cycle is considered an attractive option for waste heat recovery
  - Low irreversibility – better matching of waste heat temperature
  - Compact design - about 10 times smaller than SRC (Santini et al., 2016)
  - Improved performance
  - Applicable to a wide range of waste heat source
Contribution of this Study

- Most previous studies focused on the steady state design point performance and analysis of s-CO$_2$ power cycle waste heat recovery (s-CO$_2$/WHR) system (Kim et al., 2012; Manjunath et al., 2018)
- In real scenarios, industrial processes undergo transient operation leading to highly dynamic mass flow rates and temperature of the exhaust gases (Xu et al., 2017)
- This is likely to significantly affect the dynamic performance, operation and control of the s-CO$_2$/WHR system due to strong coupling with the industrial process through the heat recovery heat exchanger
- Dynamic model is developed for operating conditions and power out predictions and for control system design of the s-CO$_2$/WHR system
Aim and Objectives

Aim

• To assess the dynamic performance and control of s-CO$_2$ power cycle for waste heat recovery from industrial processes

Objectives

• Steady state simulation to determine design point values
• Dynamic model development for s-CO$_2$ power cycle using industrial waste heat
• Open loop simulation to show inherent dynamic response to step change in the flow rate and temperature of the waste heat flue gas
• Dynamic modelling, simulation and control of the system under varying flue gas flow rate between 100% and 80% of the design value
• Dynamic modelling, simulation and control of the system under varying flue gas temperature between 380 °C and 360 °C
Description of S-CO$_2$ Power Cycles

(a) Simple recuperated closed Brayton cycle

- The baseline cycle is the simple recuperated cycle
- Recompression cycle is used to minimise the effect of heat capacity mismatch
  - Improved efficiency but more complex
  - Not suitable for low temperature heat source
  - Off design operating range is limited for low temperature application (Olumayegun, 2017)
- Single recuperator recompression cycle also resolves Cp mismatch and simpler than recompression cycle
  - Suitable for low temperature heat source

(b) Recompression closed Brayton cycle

(c) New concept-Single recuperator recompression closed Brayton cycle (Olumayegun, 2017)
Description of S-CO$_2$ Power Cycles

Operation, control and simulation challenges

- Rapid fluid property changes and non-ideal gas behavior
  - Makes solving the equation of state challenging
  - The fluid property calculation requirements were satisfied by implementing a digitized form of NIST REFPROP property data as look-up tables in the Simulink models
- Parallel operation of compressors
Dynamic Model Development

Heat exchanger model (HRHX, recuperator and precooler)

- Counter-flow heat exchanger divided into three regions

- Cold/hot stream control volume conservation of mass and energy equations:
  \[ V \frac{d\rho}{dt} = \dot{m}_i - \dot{m}_o \]
  \[ V \frac{d(\rho h)}{dt} = \dot{m}_i h_i - \dot{m}_o h_o + Q_{fw} \]

- Metal wall energy conservation equation:
  \[ M_w C_w \frac{dT_w}{dt} = Q_{fw1} - Q_{fw2} \]

- Convective heat transferred between the fluid stream and the metal wall:
  \[ Q_{fw} = K_{fw} \dot{m}_f^\alpha (T_f - T_w) \]

- Heat exchanger pressure loss estimated by:
  \[ P_i - P_o = f_s \frac{\dot{m}_i^2}{\rho_i} \]
Dynamic Model Development

Turbomachinery model (turbine and compressors)

- Mass and energy balance equations, and performance curves
- Normalised flow coefficient

\[
\dot{m}_{\text{nor}} = \frac{\dot{m}_i}{\dot{m}_{\text{ref}}} \sqrt{\frac{T_i}{T_{\text{ref}}}} \frac{P_{\text{ref}}}{P_i}
\]

- Normalised shaft speed

\[
N_{\text{nor}} = \frac{N}{N_{\text{ref}}} \sqrt{\frac{T_i}{T_{\text{ref}}}}
\]

- Pressure ratio

\[
\pi = f_{\text{map}}(\dot{m}_{\text{nor}}, N_{\text{nor}}, P_{\text{ref}})
\]

- Isentropic efficiency

\[
\eta = f_{\text{map}}(\dot{m}_{\text{nor}}, N_{\text{nor}}, \eta_{\text{ref}})
\]

- Power added to the shaft

\[
P_{\text{shaft}} = \dot{m}_i (h_i - h_o)
\]

Turbine performance curve scaled from "Control strategies for supercritical carbon dioxide power conversion systems", Carstens, N.A. (2007), MIT
Dynamic Model Development

• Rotating shaft model:

\[
(I_t + I_{mc} + I_{rc} + I_{gen}) \frac{dN}{dt} = P_t - P_{mc} - P_{rc} - P_{\text{gen}} - P_{\text{loss}}
\]

• Control valve model:

\[
\dot{m} = C_v y \sqrt{P_i \rho_i (1 - P_o / P_i)}
\]

• Actuator model:

\[
\tau \frac{dx}{dt} = x_d - x
\]

• PID controller model:

\[
u(t) = K_p e(t) + K_i \int_0^t e(t) \, d\tau + K_d \frac{de(t)}{dt}
\]
Dynamic Model Development

Model integration in Matlab®/Simulink®
Results and Discussions

Steady state simulation conditions (design point values)

- Exhaust flue gas from a cement kiln with a mass flow rate of 100 kg/s and temperature of 380 °C was considered
- Turbine inlet temp: 360 °C
- Precooler cooling stream is water at 22 °C
- MC inlet temp: 32 °C
- Turbine efficiency: 90%
- MC efficiency: 89%
- RC efficiency: 88%

- HRHX terminal temperature difference: 20 °C
- Recuperator TTD: 10 °C
- Precooler TTD – 30 °C
Results and Discussions

Heat exchangers design parameters (PCHE)

- The transient behavior of the s-CO\(_2\)/WHR system is determined by:
  - Volume of CO\(_2\) in the heat exchanger
  - Heat exchanger material properties
  - Thermal mass between the working fluid and exhaust gas

<table>
<thead>
<tr>
<th>Description</th>
<th>HRHX</th>
<th>Recuperator</th>
<th>Precooler</th>
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</thead>
<tbody>
<tr>
<td>Heat transfer duty (MW)</td>
<td>15.07</td>
<td>16.42</td>
<td>10.01</td>
</tr>
<tr>
<td>Fluid, hot side/cold side</td>
<td>Flue gas/CO(_2)</td>
<td>CO(_2)/CO(_2)</td>
<td>CO(_2)/Water</td>
</tr>
<tr>
<td>Number of modules</td>
<td>14</td>
<td>8</td>
<td>2</td>
</tr>
<tr>
<td>Surface area density (m(^2)/m(^3))</td>
<td>714.11</td>
<td>714.11</td>
<td>714.11</td>
</tr>
<tr>
<td>Thermal density (MW/m(^3))</td>
<td>1.2</td>
<td>1.4</td>
<td>4.95</td>
</tr>
<tr>
<td>Hot side pressure loss (kPa)</td>
<td>10</td>
<td>80</td>
<td>67</td>
</tr>
<tr>
<td>Cold side pressure loss (kPa)</td>
<td>0.4</td>
<td>13</td>
<td>6</td>
</tr>
<tr>
<td>Total core volume (m(^3))</td>
<td>12.60</td>
<td>11.62</td>
<td>2.02</td>
</tr>
<tr>
<td>Total core mass (kg)</td>
<td>56829</td>
<td>52418</td>
<td>9253</td>
</tr>
</tbody>
</table>
Results and Discussions

Temperature profile along the heat exchanger

- Large increase in $C_p$ in the precooler
- Precooler heat transfer is nearly a two-phase condensation process (constant temperature in some section of the precooler)
- Small change in precooler temperature will require large amount of heat transfer in the precooler
Results and Discussions

Open loop responses to step change in heat source temperature

- Compressor outlet pressure exceeds acceptable level
- System becomes unstable due to “positive feedback” of the high pressure if not controlled
- Step change in flue gas flow rate has similar effect
Results and Discussions

Suggested control schemes for the s-CO$_2$ power cycle

BVC – Bypass valve controller
CWC – Cooling water controller
V1 – Control valves
V2 – Throttle valves
TURB – Turbine
MC – Main compressor
RC – Recompression compressor
G – Generator
HRHX – Heat recovery heat exchanger
Results and Discussions

Dynamic response to change in flue gas flowrate with cooling water control and throttle valve regulation
Results and Discussions

Dynamic response to change in flue gas flowrate with addition of bypass valve control

- Rise in flue gas exit temperature during transient
- Large drop in net power output compared to the previous case of no bypass valve control
Results and Discussions

Dynamic response to change in flue gas temperature with cooling water control and throttle valve regulation
Conclusions

- Open loop step response test highlights the need to maintain fixed working fluid pressure and temperature at precooler outlet/MC inlet.
- Cooling water control used to keep MC inlet temperature at design value during transient.
- Throttle valve regulation to maintain a constant compressor inlet pressure.
- Results of dynamic simulation and control system implementation indicates that it is better to allow the turbine inlet temperature to vary according to the heat source temperature.
- Between the choice of constant turbine inlet temperature and constant turbine mass flow rate, constant mass flow rate is more beneficial during transient variation of flue gas condition.
References

- Olumayegun, O., 2017. Study of closed-cycle gas turbine for application to small modular reactors (SMRs) and coal-fired power generation through modelling and simulation (Doctoral dissertation, University of Sheffield).
Questions