Off-design Performance of the Recompression sCO2 Cycle for CSP Applications

5th International Supercritical CO2 Power Cycles Symposium

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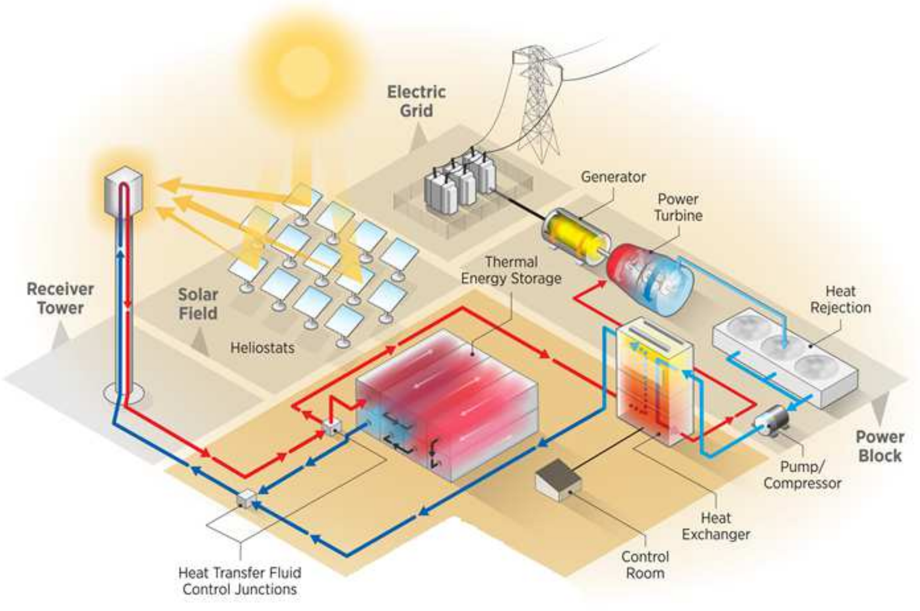
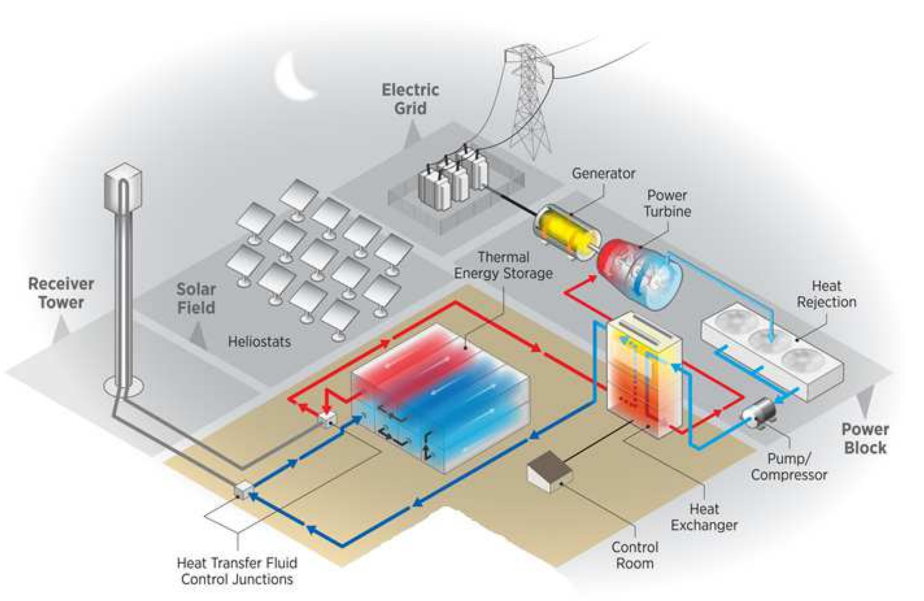
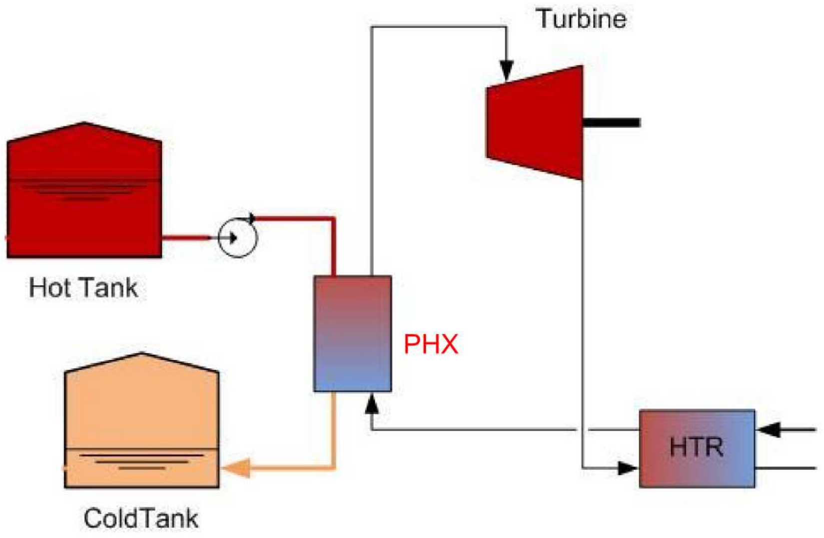
Concentrating Solar Power Overview





- Concentrated sunlight heats a fluid (e.g. molten salt) that delivers thermal power to cycle
- Hot fluid is stored and dispatched to cycle to maximize revenue
- Storage decouples electricity generation from sunlight; CSP acts as dispatchable generator



sCO2 Cycle Attributes and Areas of Focus

Attributes

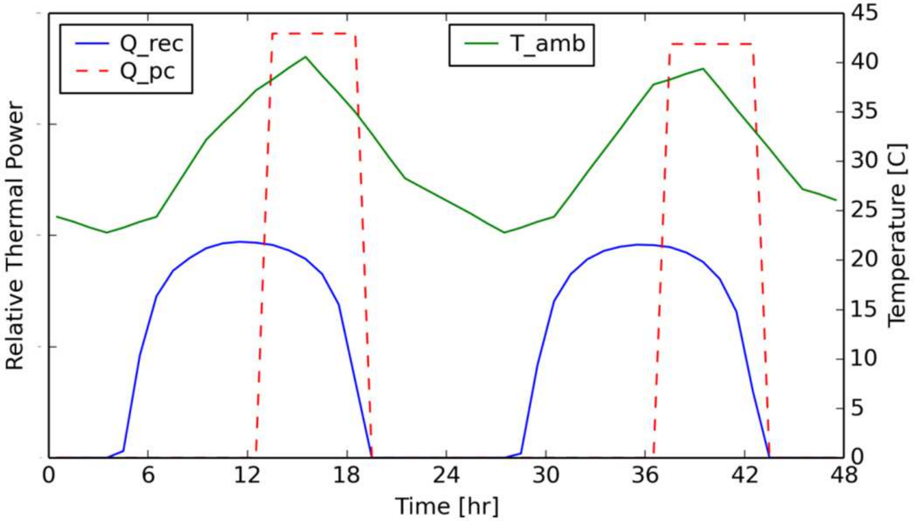
- **Efficiency at CSP temps (550 – 750 C)**
- **Performance at smaller capacity (10-50 MWe)**

  Enables smaller, cheaper, more efficient CSP systems

  - **Potentially allows for more favorable financing options**

CSP Areas of Focus

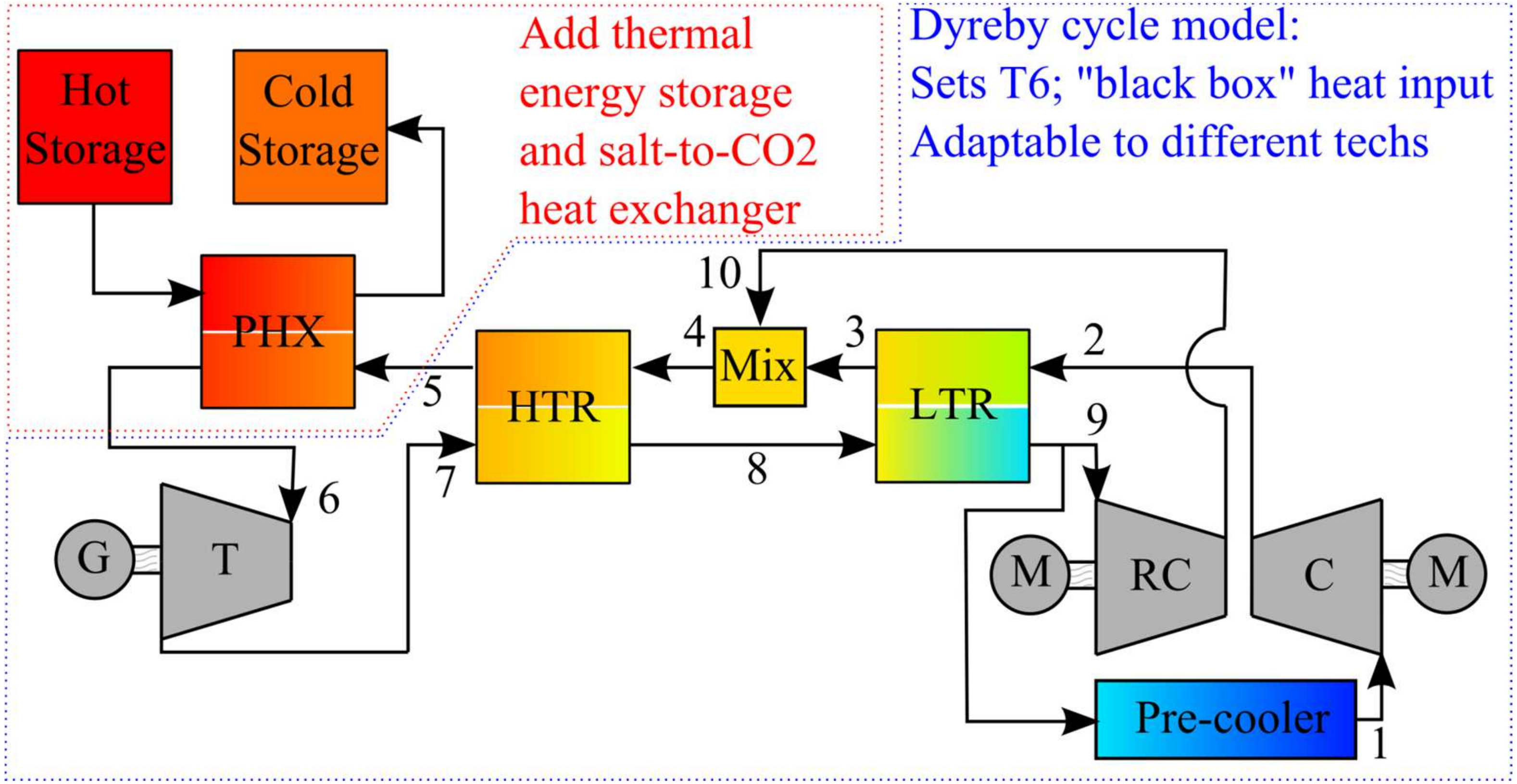
- **Performance & control of dry-cooled cycles in desert climates**
  - Cycle operation over varying (hot) compressor inlet temps
- **ΔT over primary heat input & thermal storage: storage cost**
  - Off-design & part-load operation impact on storage capacity
- **Dispatchable generation**
  - Part-load operation, ramping, and standby



Modeling Overview

Approach: Extend and analyze Dyreby's cycle models for CSP applications

Dyreby, J. J., 2014, "Modeling the Supercritical Carbon Dioxide Brayton Cycle with Recompression," THE UNIVERSITY OF WISCONSIN-MADISON


Add thermal energy storage and salt-to-CO2 heat exchanger

Dyreby cycle model:
Sets T6; "black box" heat input
Adaptable to different techs

Component Models:

- Counterflow HX
- Compressor
- Turbine

Simplifications:

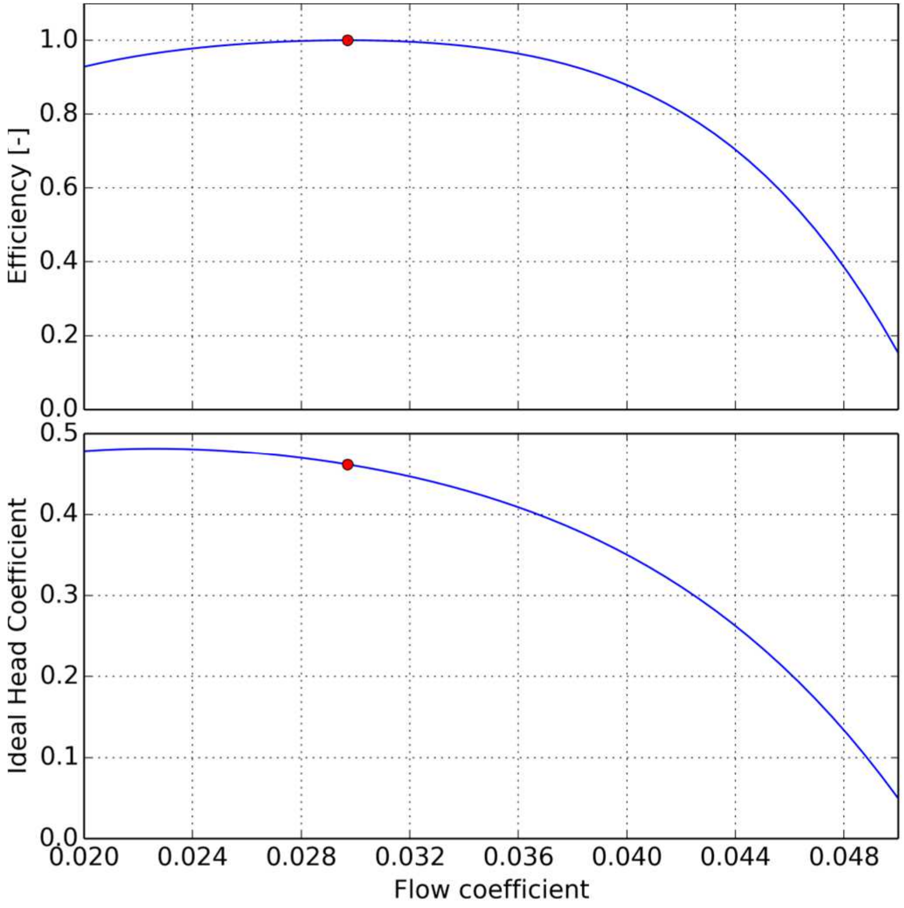
- Pre-cooler can achieve constant 15°C difference between comp inlet and ambient temperatures
  - Fan parasitics ignored
- Generator/motor efficiencies included in turbomachinery isentropic efficiencies

Compressor Design

Characterize performance using dimensionless parameters and performance curves fitted from SNL radial compressor data:

Ideal Head Coefficient: $\psi_i = \frac{\Delta h_i}{U_c^2}$

Flow Coefficient: $\phi = \frac{\dot{m}}{\rho U_c D_c^2}$



Δh_i	ideal (isentropic) enthalpy change
U_c	rotor tip speed
$\dot{m}$	mass flow rate
ρ	density
D_c	rotor diameter

Design

- Know: $T_{in}, P_{in}, T_{out}, P_{out}, \dot{m}, \eta_{isen}$
- Design: $\phi_{opt}, \psi_{i,opt}$
- Calculate: D_c, U_c, N_c

Compressor Off-Design Performance

Characterize performance using dimensionless parameters and performance curves fitted from SNL radial compressor data:

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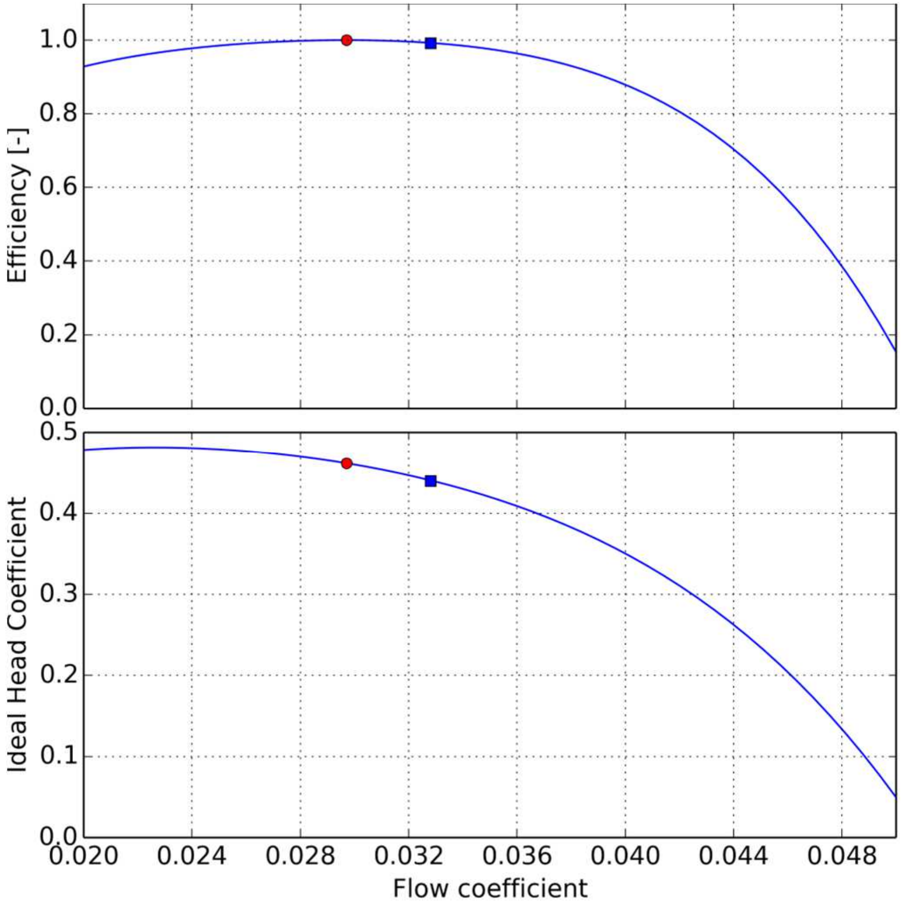


Off-Design Performance

- Design: D_c, η_{isen}
- Set: $P_{in}, T_{in}, N_c, \dot{m}$
- Calculate: $\phi, \psi, P_{out}, \eta_{adj}$

Turbine Modeling

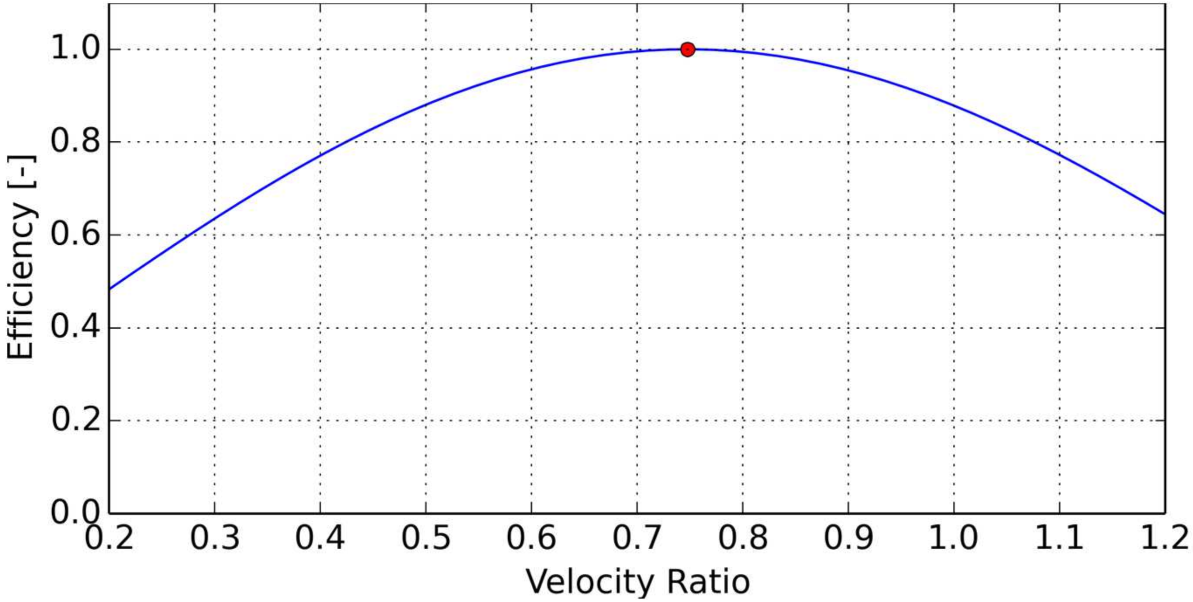
Characterize performance using dimensionless parameters and modeled efficiency curves (Wright et al. 2011):

$$\dot{m} = C_s A_{nozzle} \rho$$

$$C_s = \sqrt{2\Delta h_i}$$

$$\nu = \frac{U}{C_s}$$

A_{nozzle}	Effective nozzle area
C_s	Spouting velocity
ν	Velocity ratio



Design

- ○ Know: $T_{in}, P_{in}, T_{out}, P_{out}, \dot{m}, \eta_{isen}$,N
- ○ Design: ν_{opt}
- ○ Calculate: A_{nozzle}, D, U

Off-Design Performance

- ○ Design: $A_{nozzle}, D, \eta_{isen}$
- ○ Set: $P_{in}, T_{in}, N, P_{out}$
- ○ Calculate: $\nu, \dot{m}, \eta_{adj}$

Counterflow Heat Exchanger Modeling

Design

- Know design performance:
  - inlet and outlet states
  - mass flow rates
- Solve for required UA
- Example: PHX, CR = 1

$$\dot{q}_{max} = (\dot{m}cp)_{CO_2,des} \cdot \left(T_{HTF,hot} - T_{CO_2,PHX,in}\right)$$

$$\epsilon = \frac{\dot{q}_{actual}}{\dot{q}_{max}}$$

$$UA_{des} = \frac{\varepsilon}{1-\varepsilon}(\dot{m}cp)_{CO_2,des}$$

Off-Design

- Know hot and cold inlet conditions
- Scale conductance for off-design mass flow rate
- Solve for outlet conditions

$$U = U_{des}\left(\frac{1}{2}\left(\frac{\dot{m}_{CO_2}}{\dot{m}_{CO_2,des}} + \frac{\dot{m}_{HTF}}{\dot{m}_{HTF,des}}\right)\right)^{0.8}$$

$$C_R = \frac{\dot{C}_{min}}{\dot{C}_{max}} = \frac{\min\left(\left[\left(\dot{m}c_p\right)_{CO_2}\right],\left[\left(\dot{m}c_p\right)_{HTF}\right]\right)}{\max\left(\left[\left(\dot{m}c_p\right)_{CO_2}\right],\left[\left(\dot{m}c_p\right)_{HTF}\right]\right)}$$

$$NTU = \frac{UA_{des}}{\dot{C}_{min}}$$

Design Model

1) Find smallest total recuperator conductance resulting in design point efficiency

- Optimize compressor inlet pressure, pressure ratio, and recompression fraction

Design Point Parameters		
Net Power Output	10	MW
Thermal Efficiency (no cooling)	0.48	-
Turb. Inlet Temp	690	°C
Turb. Isentropic Efficiency	0.93	-
Turb. Shaft Speed (fixed)	3600	rpm
Main Comp. Inlet Temp	45	°C
Comp. Isentropic Efficiency	0.89	-
Maximum Pressure	25	MPa
Neglecting Pressure Drops		

Optimized Design Point Parameters		
Comp. Inlet Pressure	9.00	MPa
Comp. Pressure Ratio	2.74	-
Recompression Fraction	0.18	-

Design Point Solution Results		
Recuperator Conductance	1375	kW/K
Main Comp. Shaft Speed	33294	rpm
Turb. Rotor Diameter	2.32	m
CO2 Mass Flow Rate	82.0	kg/s
PHX CO2 Inlet Temperature	505	°C

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PHX CO2 Inlet Temperature	505	°C

2) Use design cycle solution and PHX parameters to find PHX design

Required Design Point Solution Results	Design Point Parameters			Design Point Solution Results		
PHX CO2 Inlet Temperature	HTF Inlet Temp	700	°C	PHX Conductance	2.9×10^5	kW/K
Turb. Inlet Temp	Capacitance ratio	1		PHX NTU	9.39	
CO2 Mass Flow Rate				HTF Mass Flow Rate	72.4	kg/s

Off-Design Model

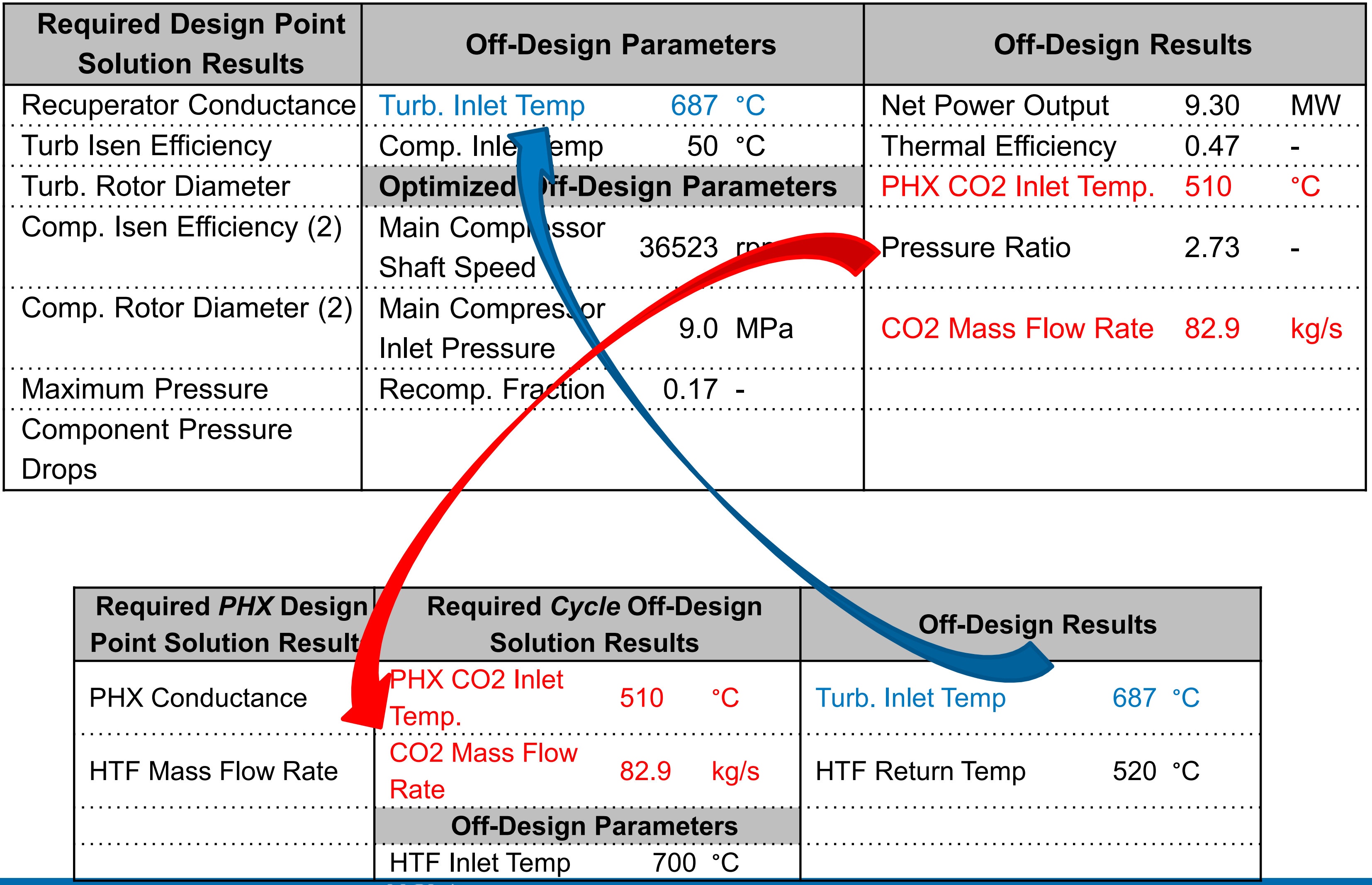
Off-Design Conditions: 1) Ambient Temperature, 2) HTF mass flow rate, 3) HTF Temp

1) Given a turbine inlet temperature, optimize compressor shaft speed, compressor inlet temperature, and recompression fraction to maximize efficiency

Required Design Point Solution Results	**Off-Design Parameters**			**Off-Design Results**		
Recuperator Conductance	Turb. Inlet Temp	687	°C	Net Power Output	9.30	MW
Turb Isen Efficiency	Comp. Inlet Temp	50	°C	Thermal Efficiency	0.47	-
Turb. Rotor Diameter	**Optimized Off-Design Parameters**			PHX CO2 Inlet Temp.	510	°C
Comp. Isen Efficiency (2)	Main Compressor Shaft Speed	36523	rpm	Pressure Ratio	2.73	-
Comp. Rotor Diameter (2)	Main Compressor Inlet Pressure	9.0	MPa	CO2 Mass Flow Rate	82.9	kg/s
Maximum Pressure	Recomp. Fraction	0.17	-			
Component Pressure Drops						

Off-Design Model

Iterate to find turbine inlet temperature that converged cycle and PHX models

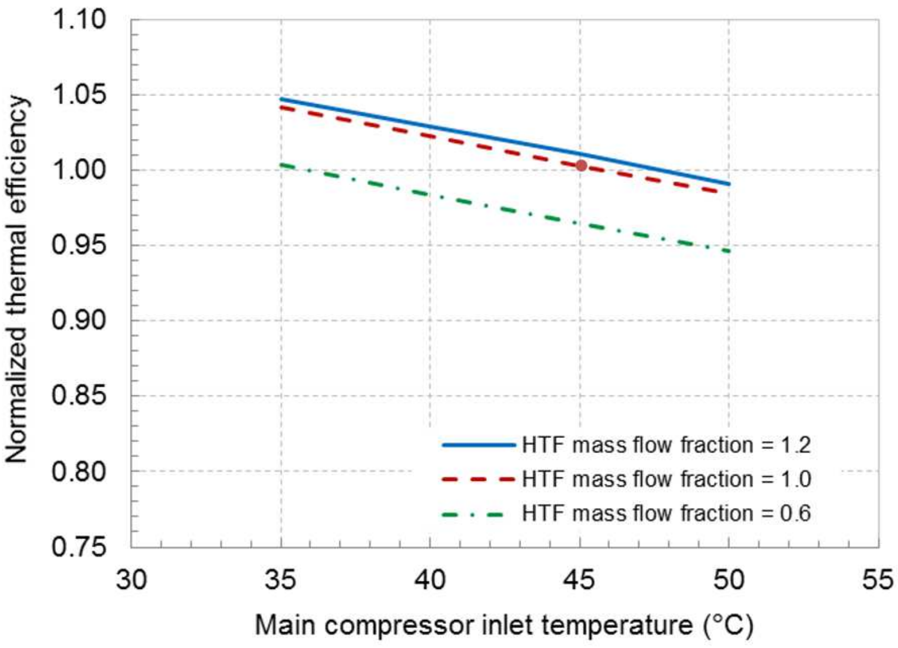


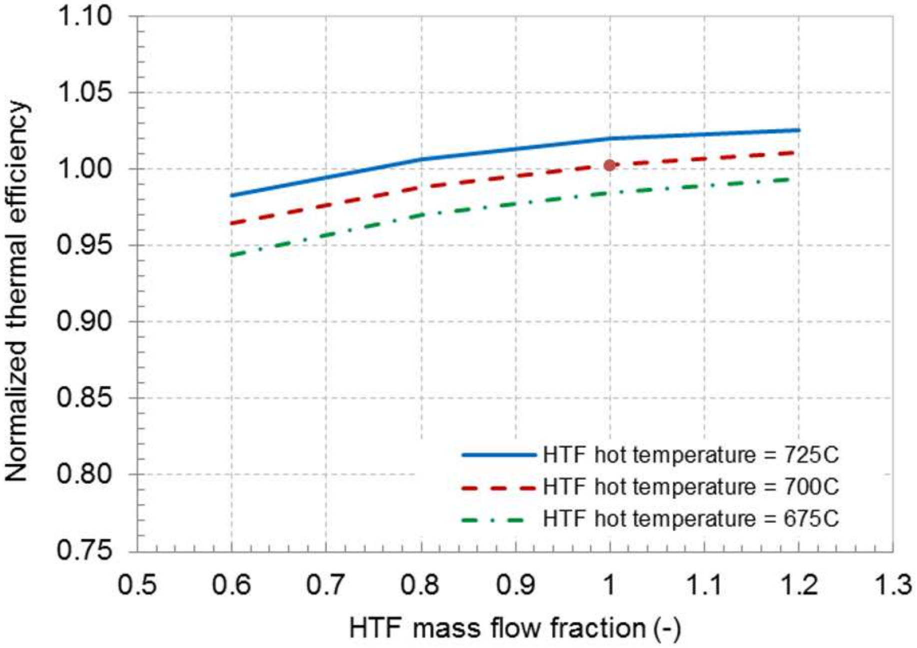
Required Design Point Solution Results	Off-Design Parameters			Off-Design Results		
Recuperator Conductance	Turb. Inlet Temp	687	°C	Net Power Output	9.30	MW
Turb Isen Efficiency	Comp. Inlet Temp	50	°C	Thermal Efficiency	0.47	-
Turb. Rotor Diameter	**Optimized Off-Design Parameters**			PHX CO2 Inlet Temp.	510	°C
Comp. Isen Efficiency (2)	Main Compressor Shaft Speed	36523	rpm	Pressure Ratio	2.73	-
Comp. Rotor Diameter (2)	Main Compressor Inlet Pressure	9.0	MPa	CO2 Mass Flow Rate	82.9	kg/s
Maximum Pressure	Recomp. Fraction	0.17	-			
Component Pressure Drops						

Required *PHX* Design Point Solution Results	Required *Cycle* Off-Design Solution Results			Off-Design Results		
PHX Conductance	PHX CO2 Inlet Temp.	510	°C	Turb. Inlet Temp	687	°C
HTF Mass Flow Rate	CO2 Mass Flow Rate	82.9	kg/s	HTF Return Temp	520	°C
	Off-Design Parameters					
	HTF Inlet Temp	700	°C			

Off-Design Cycle Efficiency

- **Trends compare well against Dyreby's "black box" heat input model**
- **HTF mass flow > 1 results in turbine temps > design**
  - Will add constraint in future analyses
- **High efficiencies at part load**
  - Inventory control & improved regeneration



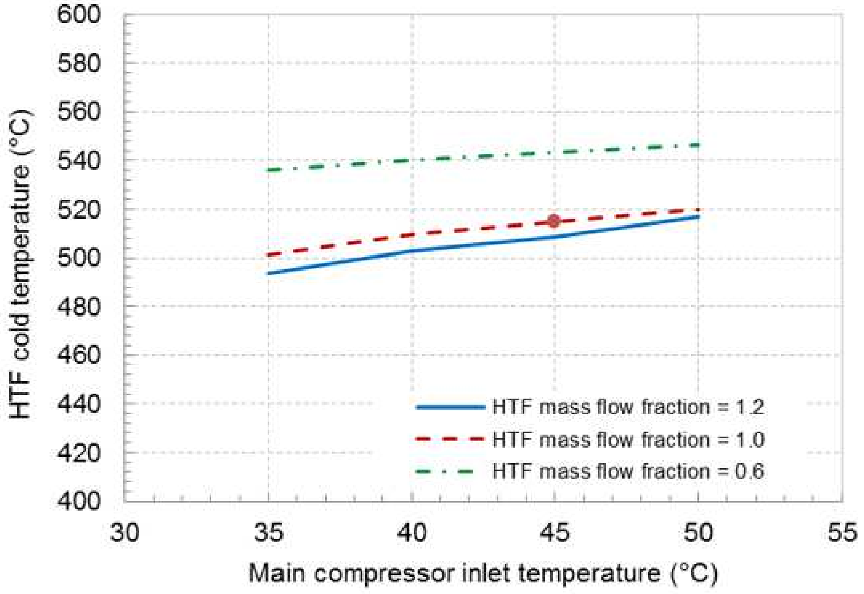


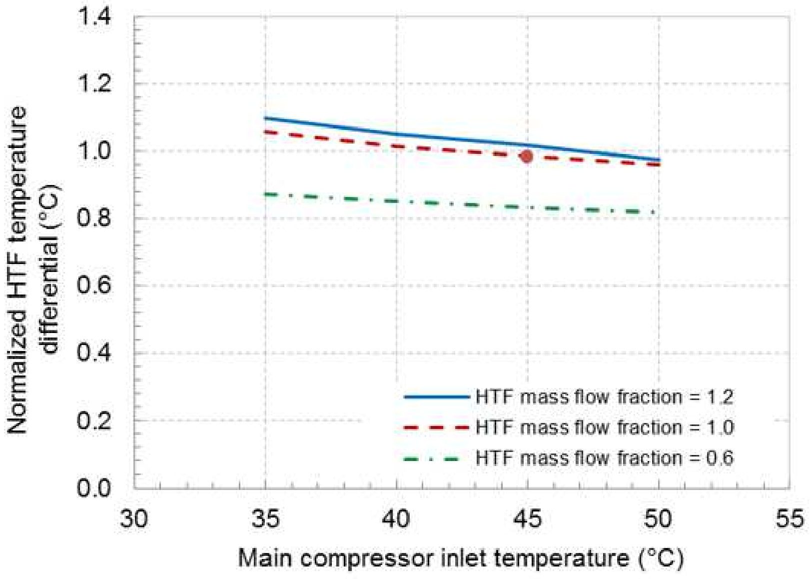
Off-Design TES Temperature Difference

$$Energy_{TES} = mass * specific\ heat\ * (T_{hot} - T_{cold})$$

For sensible heat storage:

- The required mass of HTF is proportional to the hot and cold tank temperatures.
- All else equal, a cycle with a larger temperature difference is preferred to a cycle with a smaller temperature difference



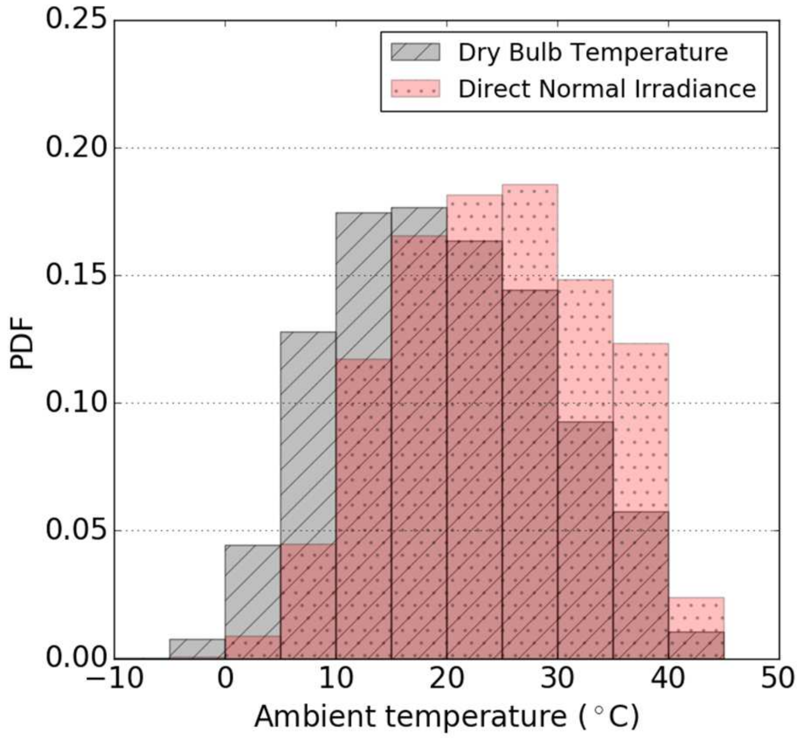


Annual Operating Ambient Temperatures

How might the design compressor inlet temperature affect annual performance

Scenarios

1. **Cycle operates every hour of year (baseload)**
2. **Cycle generates power proportionally to available DNI (CSP w/ small TES)**

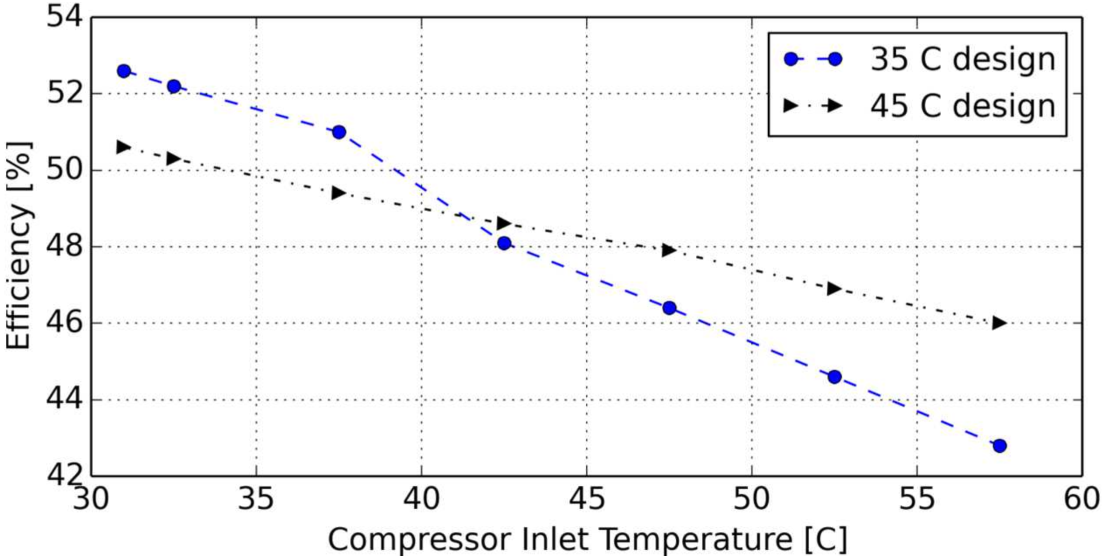


$$\text{Annual Average Ambient Temperature} = 19.5°\text{C}$$

$$\text{Annual DNI} - \text{weighted Ambient Temp} = \sum_{i=1}^{8760} T_{amb,i} * \frac{DNI_i}{\sum_{j=1}^{8760} DNI_j} = 24.3°\text{C}$$

Annual Efficiency Estimates

- Design cycles for three compressor inlet temperatures
- Calculate off-design efficiencies at each PDF bin (assuming design point HTF mass flow rate & temperature)
- Don’t allow compressor inlet temperatures colder than critical temperature



	Compressor inlet temperature 35°C (51.4%)	Compressor inlet temperature 45°C (48.0%)
Ambient temperature-weighted annual efficiency (-)	50.5%	49.5%
Direct normal irradiance-weighted annual efficiency (-)	49.3%	48.9%

Conclusions

- **Off-design cycle performance can affect the performance of the heat source**
  - Reduced thermal energy storage capacity
- **Cycle design should consider likely cycle *operating* conditions, especially ambient temperature *at generation***
  - Electricity peak pricing would drive “average” annual temperature even higher than DNI-weighted
- **Next steps:**
  - Air-cooler design and operation
  - Cost models as function of design
  - Economic analysis
- **Off-design optimization can be difficult**
  - Lot’s of infeasible points, especially at large HTF mass flow rates & hot ambient temps
  - Develop proxy model to screen solution space and generate guess values?

Thank You!

Questions?