8^{th} sCO_2 Symposium

San Antonio, TX February 26-29, 2024

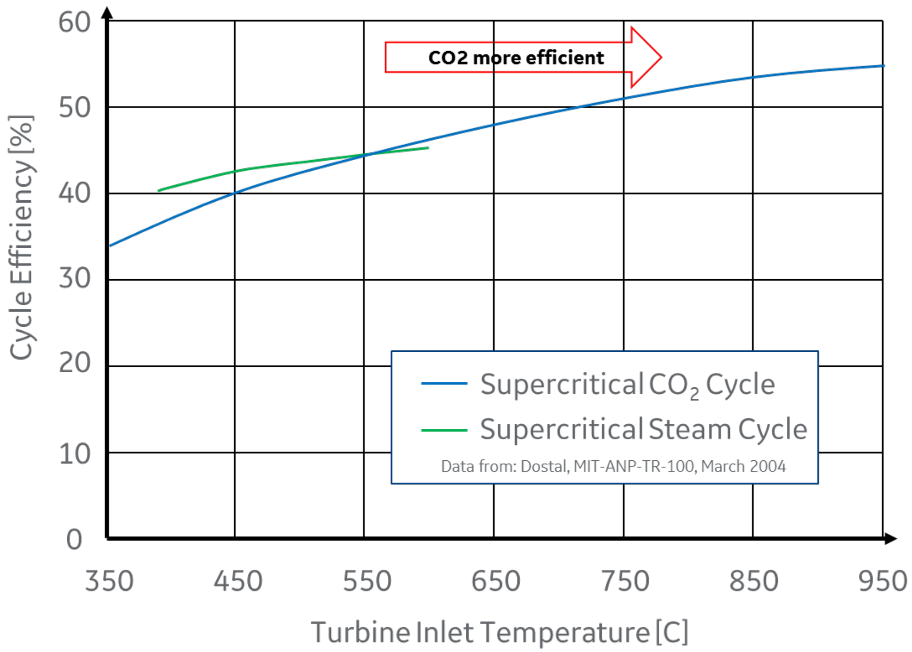
Cycles Tutorial

Doug Hofer, SwRI

Tim Held, Echogen Power Systems

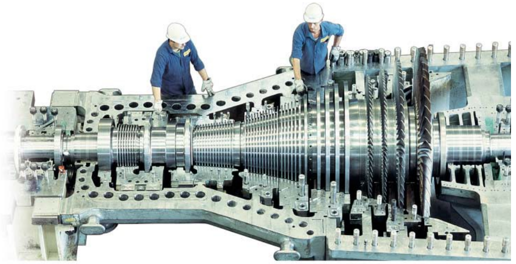
Motivation for CO_2 as a Working Fluid

High Efficiency



Smaller Turbomachinery

20MW steam turbine



14MW sCO2 turbine

27,000 RPM
150 lb rotor
7” rotor tip diameter

Renewed interest due to enabling advances in:

- Materials (USC, AUSC programs)
- Dry Gas Seals (more experience in CO_2 and LNG compressors)
- Compact Heat Exchangers

Work on sCO2 predates Dostal

Feher 1967

1. Feher, E. G., "The Supercritical Thermodynamic Power Cycle," Douglas Paper No. 4348, presented to the Intersociety Energy Conversion Engineering Conference, Miami Beach, Florida 13-17 Aug. 1967.

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AFAPL-TR-68-100

INVESTIGATION OF SUPERCRITICAL (FEHER) CYCLE

E. G. Feher et al.

Astropower Laboratory, Missile & Space Systems Division
A Division of McDonnell Douglas Corporation

TECHNICAL REPORT AFAPL-TR-68-100

October 1968

D D C
NOV 1 4 1968

Angelino 1968

Carbon Dioxide Condensation Cycles For Power Production

G. ANGELINO
Lecturer of Special Power Plants, Politecnico, Milan, Italy

The thermodynamic performance of several condensation cycles employing carbon dioxide as working medium is analyzed and discussed. A balanced distribution of thermodynamic losses between mechanical components and heat exchangers attained through a compression performed partially in the liquid and partially in the gas phase yields cycle efficiencies which are among the highest achievable in present-day energy systems. At turbine inlet temperatures higher than 650 deg C single heating CO_2 cycles exhibit a better efficiency than reheat steam cycles. This may prove of particular interest in connection with high temperature nuclear heat sources. However, the requirement of low temperature cooling water for a good cycle arrangement represents a geographical limitation to the widespread application of CO_2 condensation cycles.

Introduction

Large capacity steam power stations represent the most efficient tool for the conversion of heat into mechanical energy. After a rapid improvement of cycle configuration and equipment during the last two decades, steam plants reveal at present some thermodynamic and technological limitations which may be a hindrance to further developments. These limitations can be summarized as follows: (*a*) Cycle economy is not very sensitive to the rise of the turbine inlet temperature beyond about 600 deg C; (*b*) cycle complexity increases with the plant capacity due to the necessity of providing additional feed-water heating lines and additional low pressure turbine sections for units of growing output.

Stations employing the closed-cycle gas turbine are characterized by a simpler arrangement of the components and by the ability to take full advantage of increasing maximum temperatures. However, their efficiency, even for the highest practical temperatures, is considerably lower than that of current steam stations.

Condensing or partially condensing cycles employing carbon dioxide as a working medium allow the achievement of efficiencies similar to that of steam cycles, or even better for the highest turbine inlet temperatures. The cycle arrangement retains almost the same simplicity as the gas turbine configuration. However, due to the low critical temperature of carbon dioxide, condensing cycles are obtainable only in countries where cooling water at temperatures not higher than 12–15 deg C is available the year round.

Deep water temperature of the largest European or North American Lakes is about 4 deg C at all the times of the year. A similar temperature characterizes deep waters of the open seas (while the temperature of the Mediterranean, at the depth of 500 m, has the constant value of about 13 deg C). Along the northern coasts of the Eurasian and of the American continent, seawater at about 0 deg C is available during almost the whole year. In these regions the use of carbon dioxide as working medium is particularly attractive both from a thermodynamic point of view (full advantage can be taken of the low temperature of the cold source) and for technological reasons (there is no danger of solidification of the working fluid during shutdown periods).

The requirement of abundant, low temperature water for cooling purposes entails a geographical limitation to the application of carbon dioxide condensing cycles. Hydraulic works larger than required by steam stations could be necessary; however, their influence on the overall station economy should not be prominent.

Cycle Configurations

For carbon dioxide cycles, as for steam, a variety of cycle

Contributed by the Gas Turbine Division and presented at the Gas Turbine Conference, Washington, D. C., March 17–21, 1968, of The American Society of Mechanical Engineers. Manuscript received at ASME Headquarters, December 28, 1967. Paper No. 68—GT-23.

Nomenclature

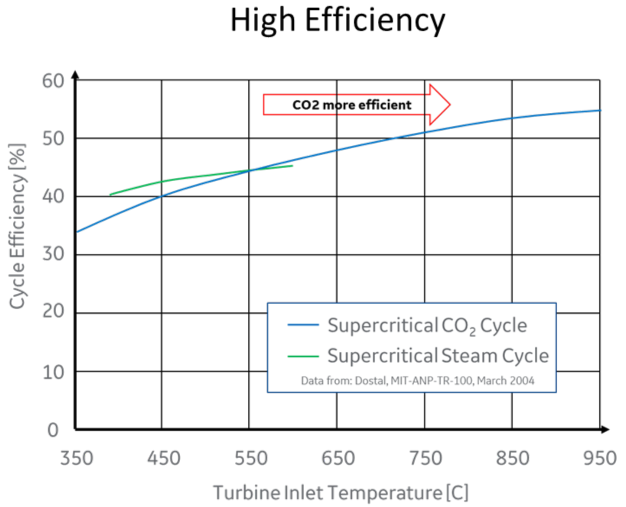
A, B, C, D = conventional denomination of cycles, Fig. 1
a = velocity of sound, m/sec
c_p = specific heat at constant pressure, kcal/kg-deg C
c_v = specific heat at constant volume, kcal/kg-deg C
h = enthalpy, kcal/kg
L = specific work, kcal/kg
L_t^* = high pressure turbine work in cycle C, kcal/kg
p = pressure, atm
Q_1 = primary heat, kcal/kg
Q_2 = waste heat, kcal/kg
S = entropy, kcal/kg-deg C
t = temperature, deg C
α = condensed fraction of working medium
Δh_r = regenerated heat, kcal/kg
$\Delta h_r/L_u\Delta t_{lm}$ = heat transfer parameter, proportional to heat exchange surface, deg C^{-1}
Δt_1 = minimum temperature difference in low temperature regenerator, deg C
Δt_2 = minimum temperature difference in high temperature regenerator, deg C being also the maximum temperature difference in low temperature regenerator
$\Sigma\Delta p/p$ = fractional pressure loss of cycle
ΔS = entropy production, kcal/kg – deg C
Δt_{lm} = log mean temperature difference, deg C
$\Delta\eta$ = loss in efficiency
η = efficiency
ρ = density, kg/cu m

Subscripts

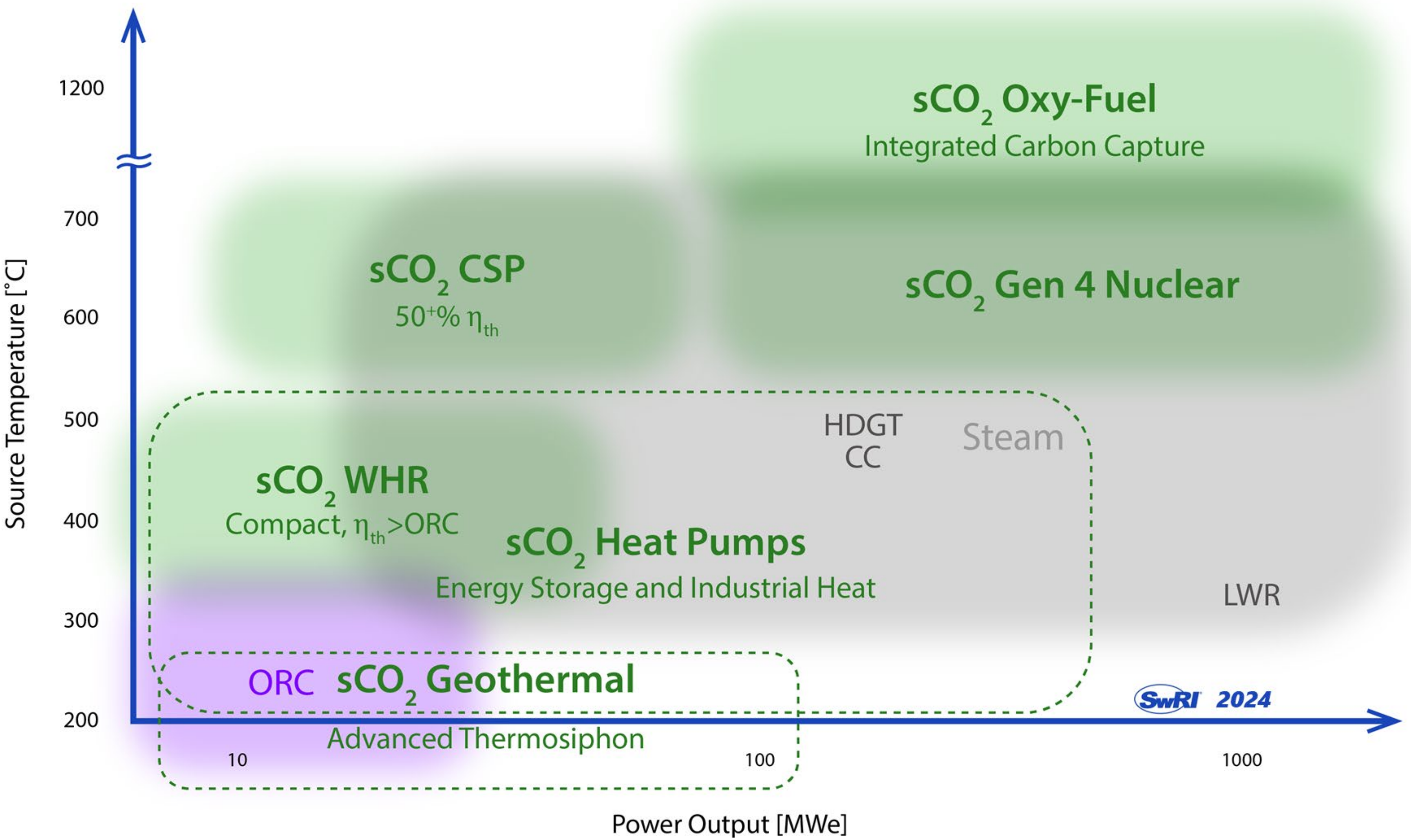
1 = low temperature regenerator
2 = high temperature regenerator
c = compressor or Carnot cycle
max = maximum
min = minimum
p = pump
s = saturation or condensation
t = turbine
u = useful or net

Journal of Engineering for Power JULY 1968 / 287

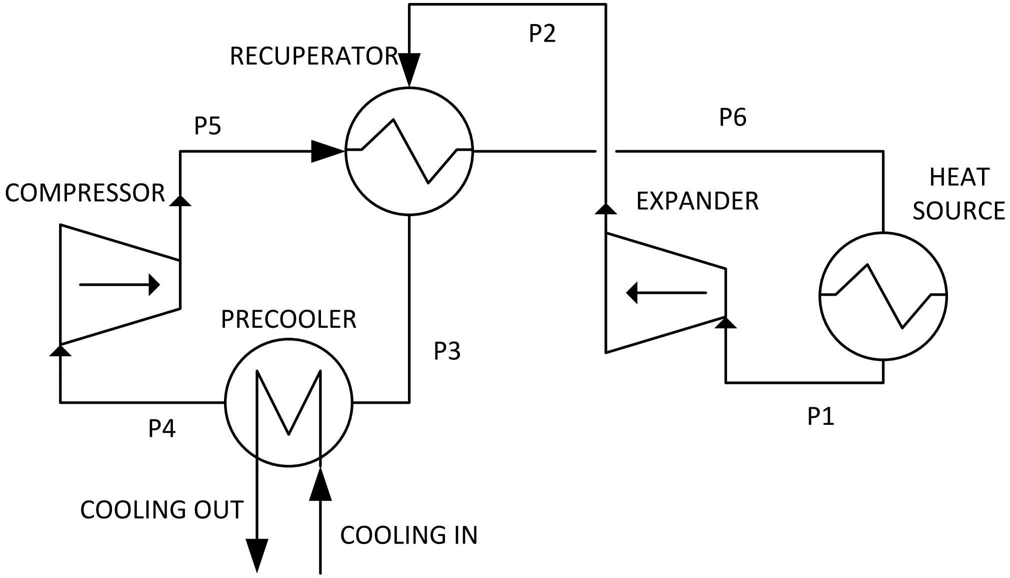
Dostal 2004

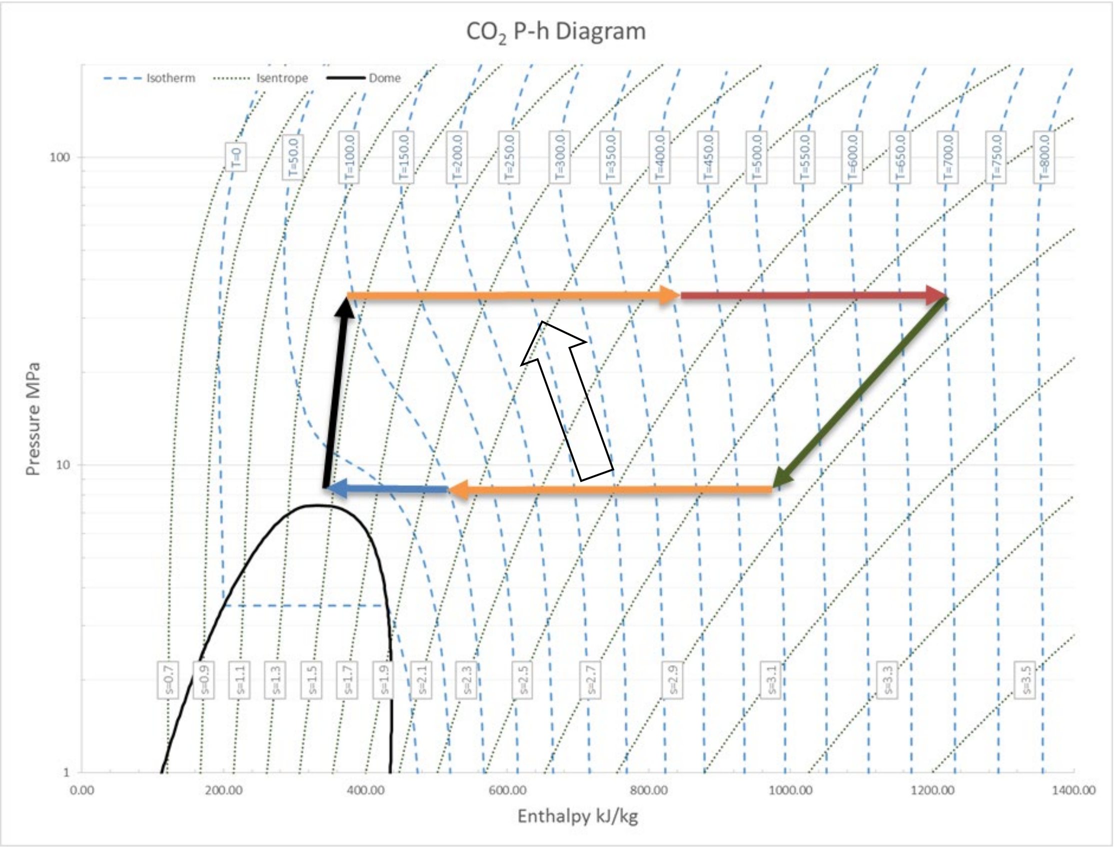


sCO2 Application Space



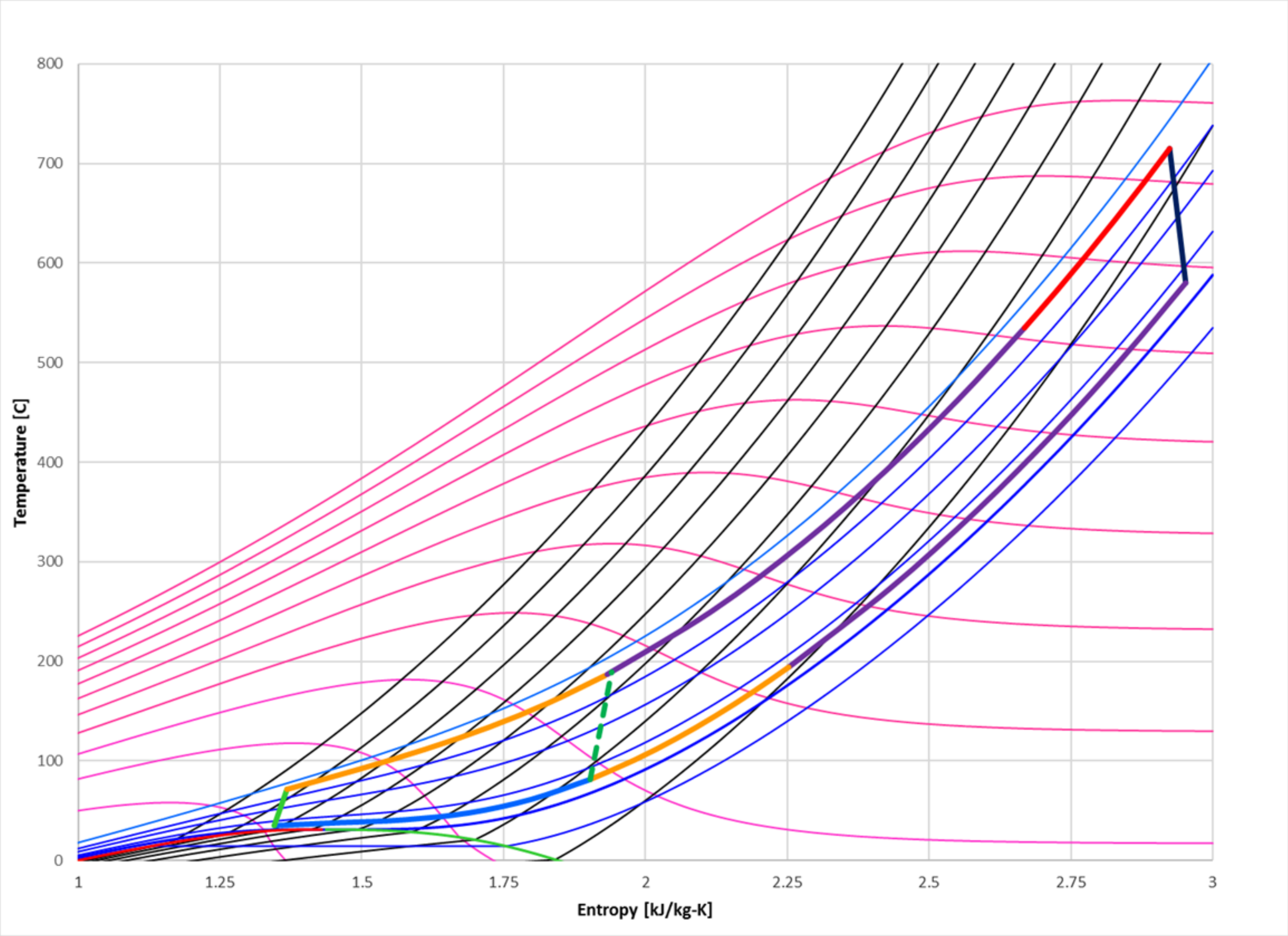
Basic Components in a Simple Recuperated Cycle





Heat In
Heat Out
Compress
Expand
Recuperate

What's Special about CO_2?



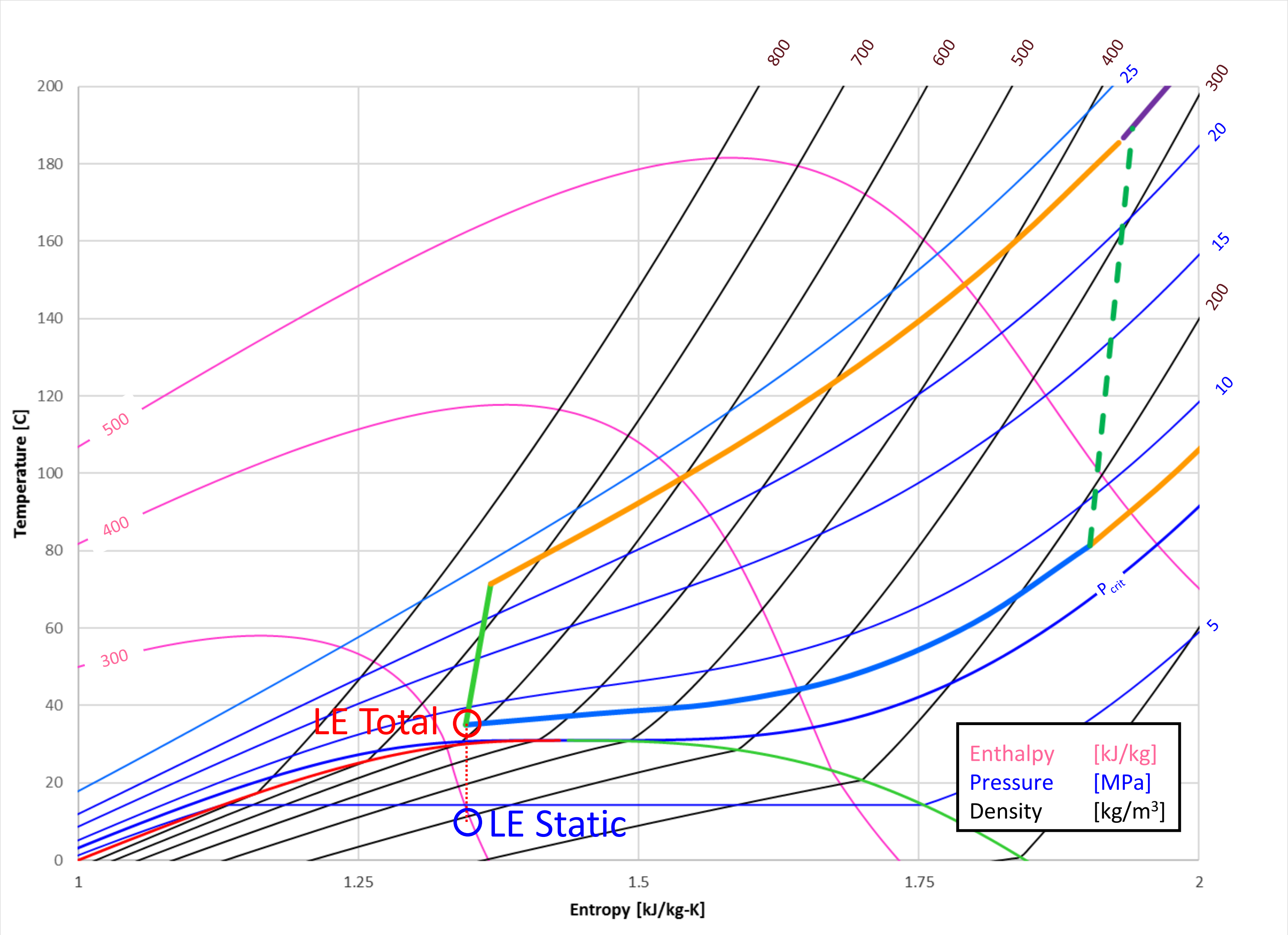
T-s Diagram for CO2 with typical high temperature recompression cycle

Note ambient temperature is near the critical point

Pink lines are constant enthalpy – horizontal indicates h=Cp*T is valid

Image Courtesy of GE
2017 IGTI Cycles Tutorial

What's Special about CO_2?



Close up of low T region

Constant enthalpy lines far from horizontal

Image Courtesy of GE
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Quiz

- ★ Statepoint 1 is at 82.4 bara and 34.2 C
- ★ Statepoint 2 is at 75.0 bara and 32.0 C

Which has a higher enthalpy?

Quiz

★ Statepoint 1 is at 82.4 bara and 34.2 C
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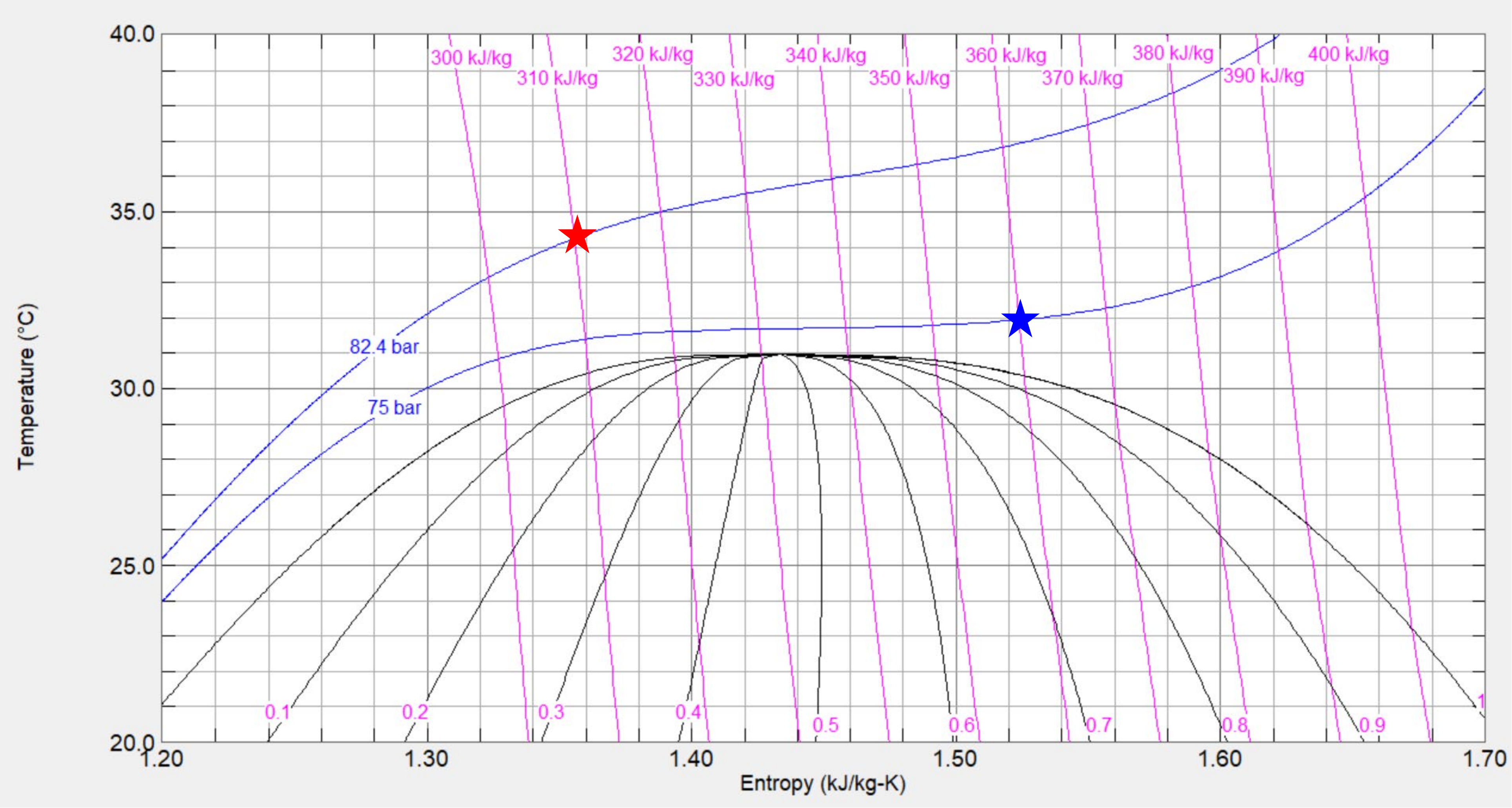
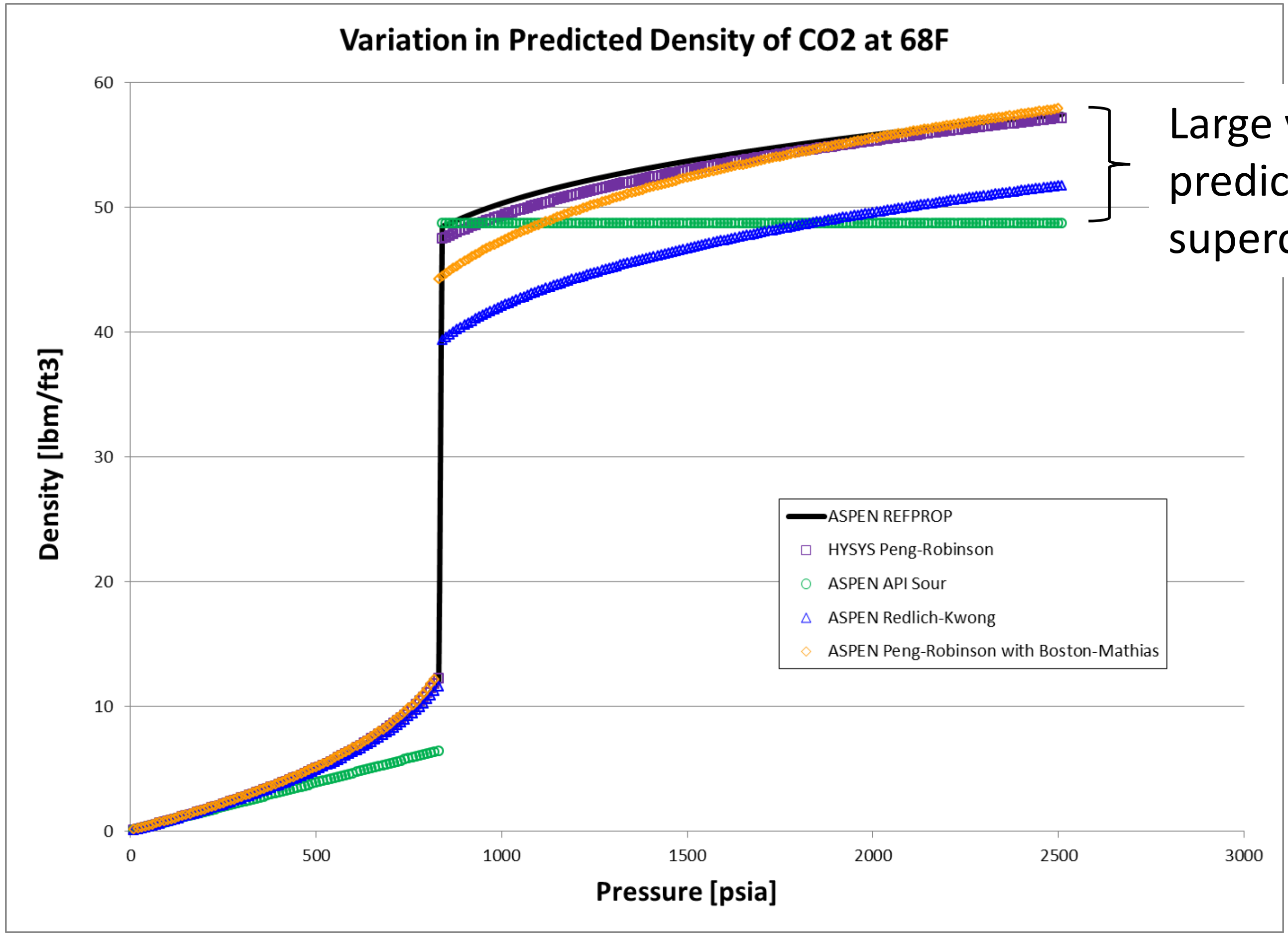
Which has a higher enthalpy?



Common CO_2 Equations of State



Large variation in predicted density for supercritical CO_2

Important to use Refprop to ensure accuracy of EOS

Image Courtesy of GE
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Carnot vs Lorenz

Carnot – Infinite Heat Reservoir
@ Constant Temperature

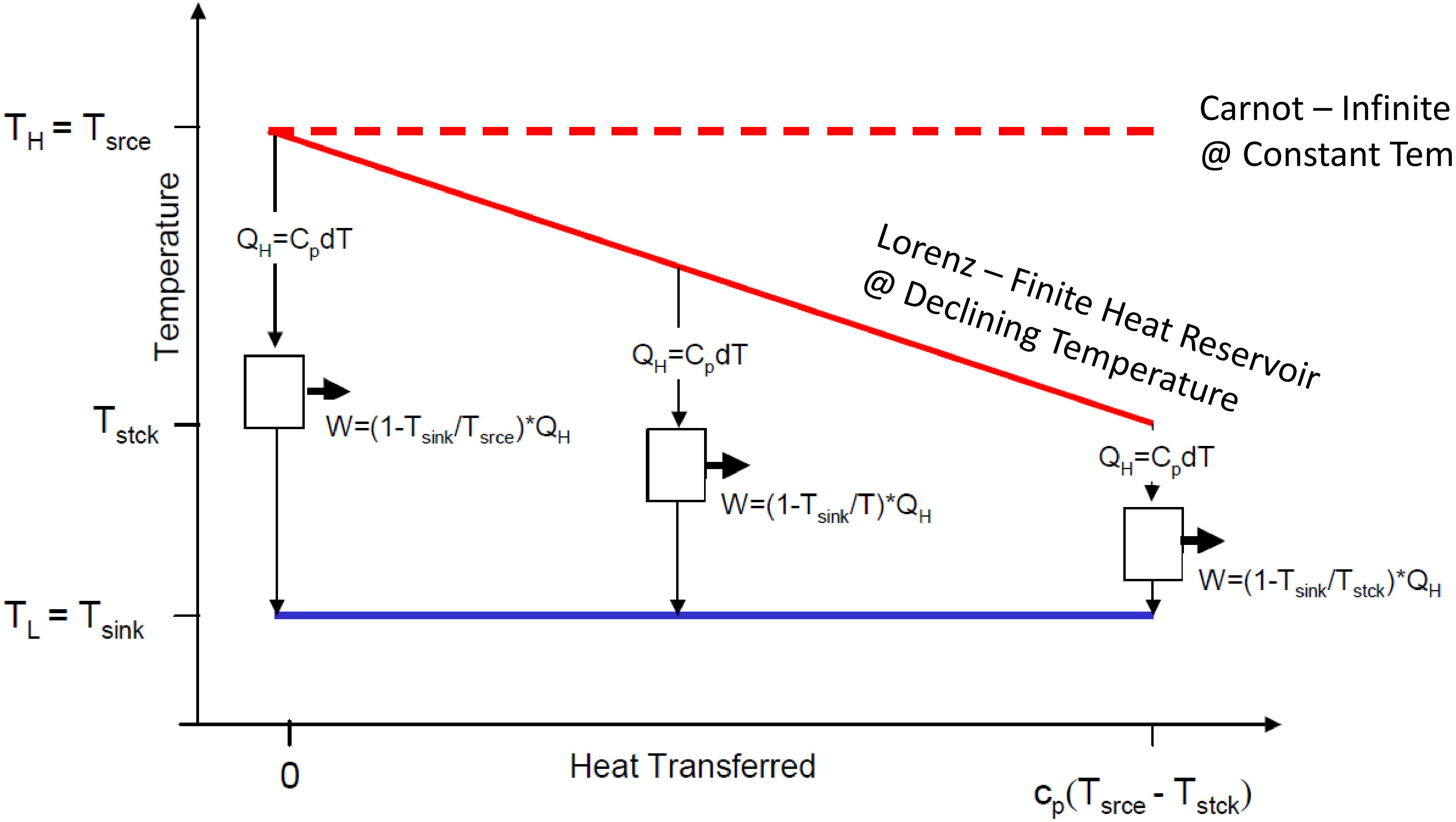
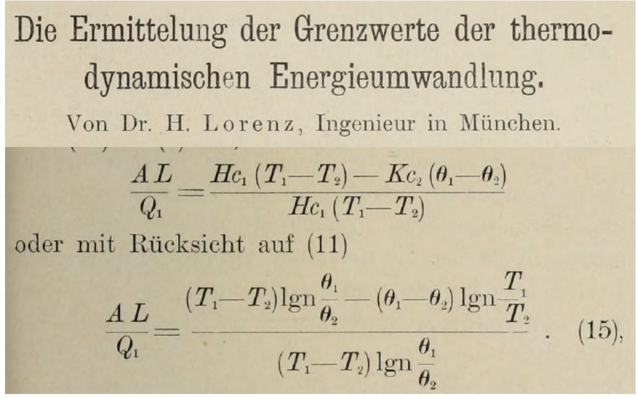


Image from Hofer & Gulen Efficiency Entitlement for Bottoming Cycles GT2006-91213

Lorenz 1895



Die Ermittelung der Grenzwerte der thermodynamischen Energieumwandlung.

Von Dr. H. Lorenz, Ingenieur in München.

$$\frac{A\,L}{Q_1} = \frac{Hc_1\,(T_1 - T_2) - Kc_2\,(\theta_1 - \theta_2)}{Hc_1\,(T_1 - T_2)}$$

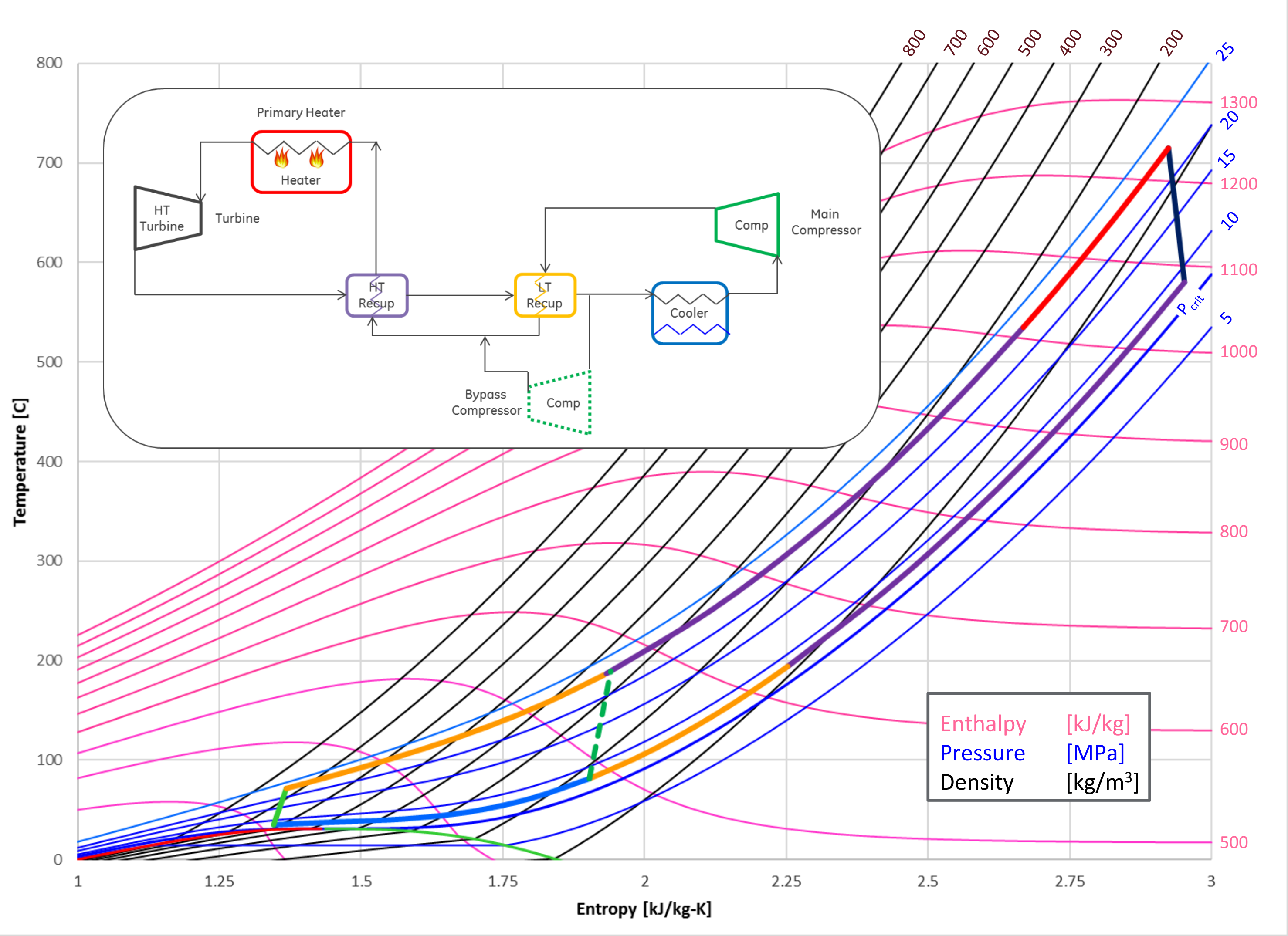
oder mit Rücksicht auf (11)

$$\frac{A\,L}{Q_1} = \frac{(T_1 - T_2)\,\mathrm{lgn}\,\frac{\theta_1}{\theta_2} - (\theta_1 - \theta_2)\,\mathrm{lgn}\,\frac{T_1}{T_2}}{(T_1 - T_2)\,\mathrm{lgn}\,\frac{\theta_1}{\theta_2}} \quad . \quad (15),$$

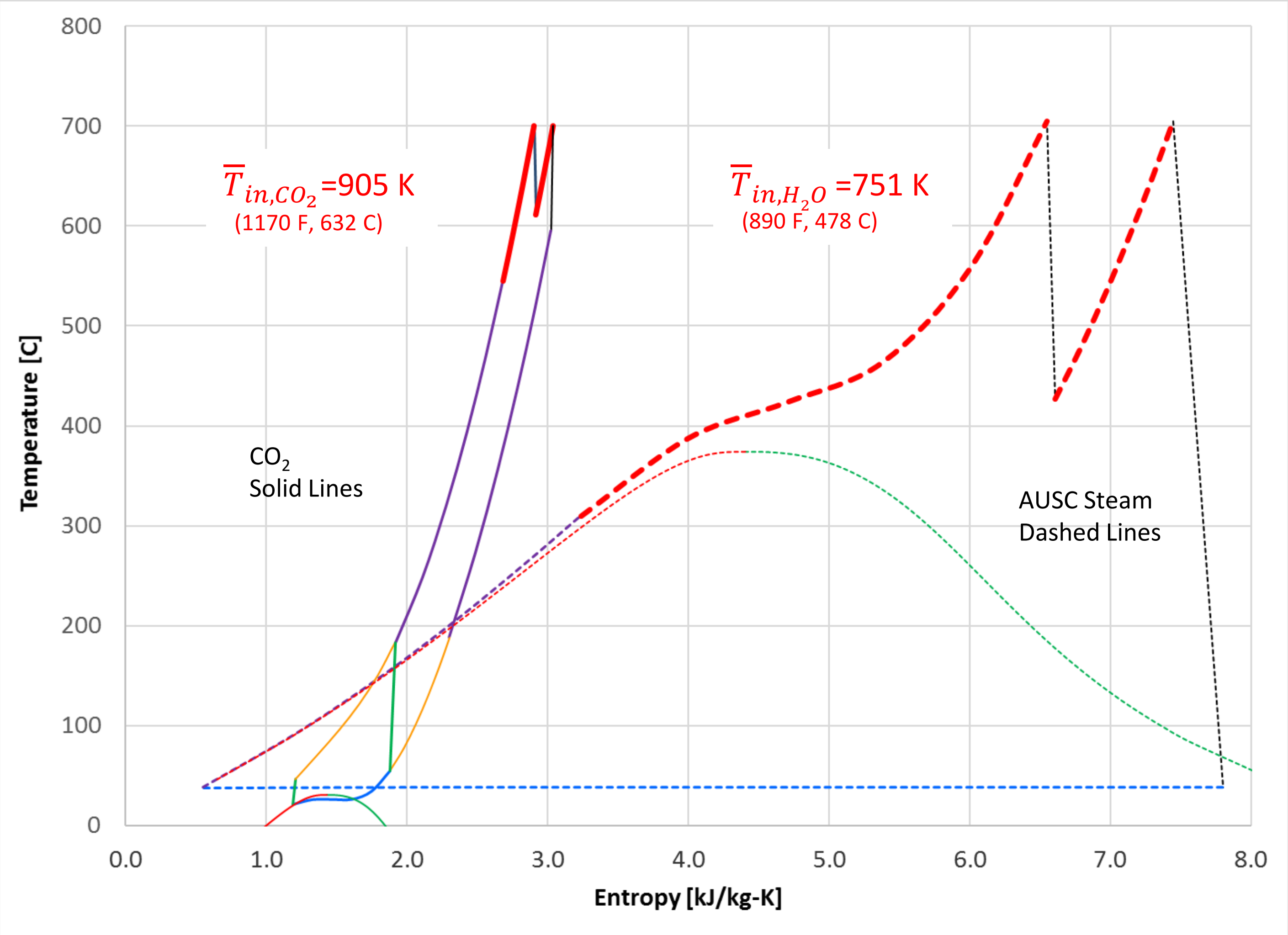
Applications and Architetures

High Temperature Cycles

sCO_2 Recompression Cycle



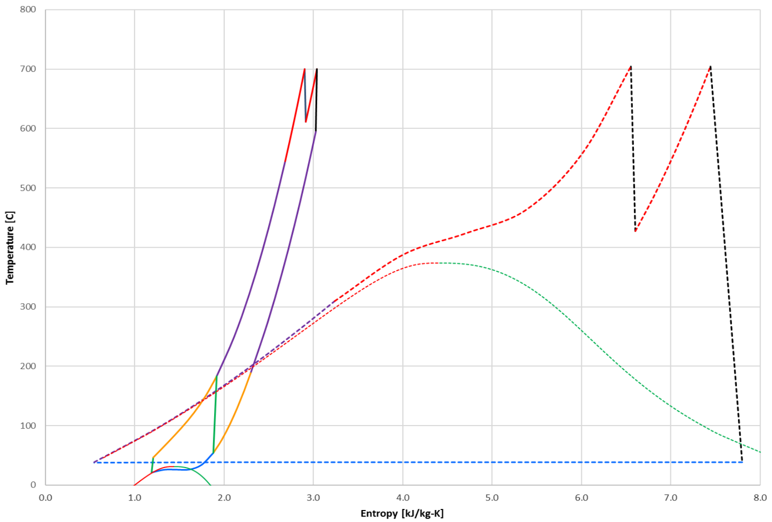
High Temperature Cycles – Thermodynamic Benefit



150 C increase in average heat addition temperature
→ 6.8 pts in Carnot efficiency

Image Courtesy of GE
2017 IGTI Cycles Tutorial

High Temperature Cycles – Flow Differences



Specific Heat Input

CO_2 = 307 kJ/kg

H_2O = 3040 kJ/kg

CO_2 mass circulation ~10X H_2O

CO_2 HP Inlet Volume Flow ~12X H_2O

CO_2 IP Inlet Volume Flow ~4.6X H_2O

$\}$ CO_2 Requires Larger Pipes for Hot Flows

CO_2 Turbine Exit Vol Flow ~0.03X H_2O → CO_2 Turbines much smaller

Image Courtesy of GE
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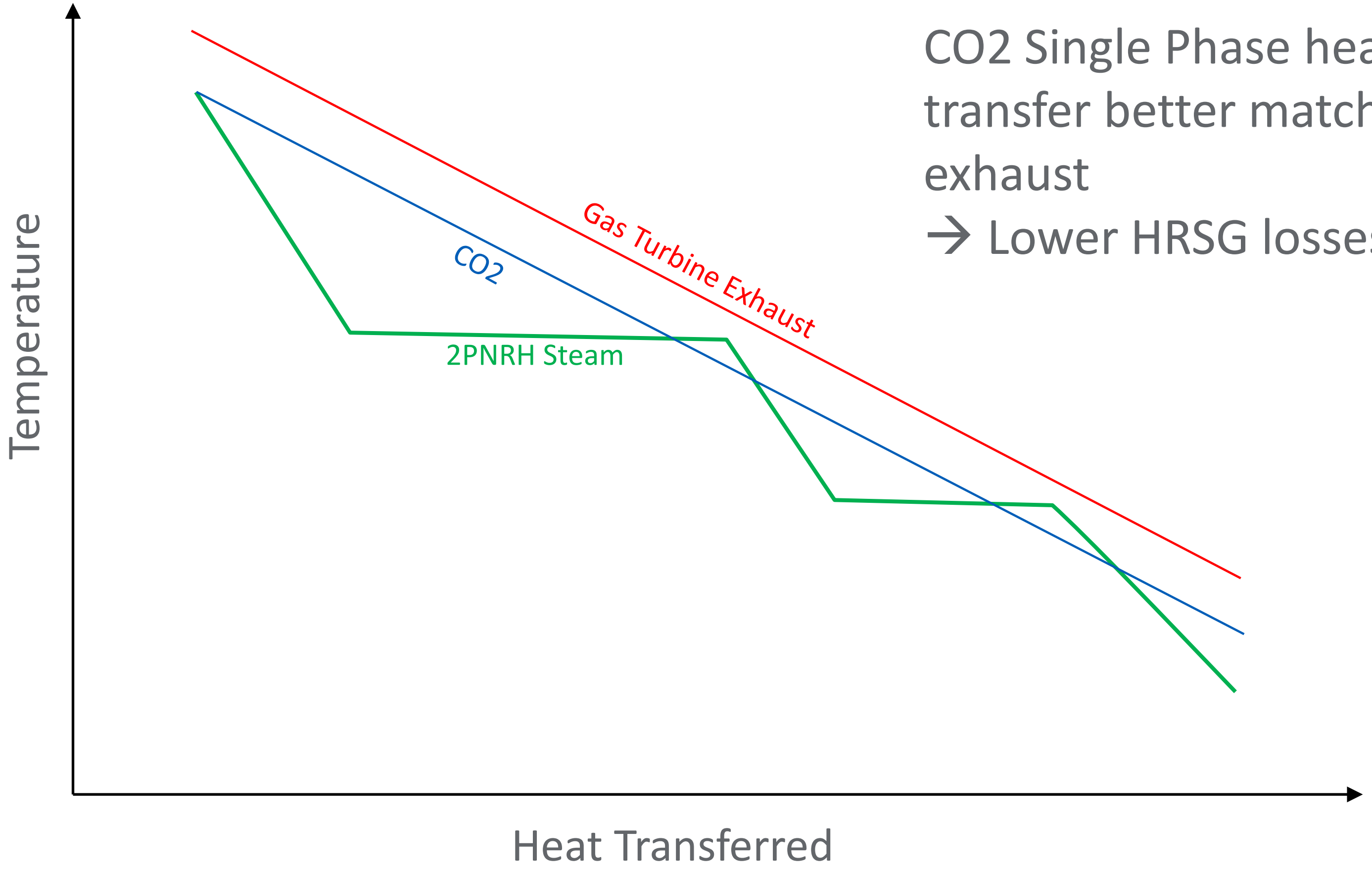
Waste Heat Recovery

Waste Heat Recovery – Thermodynamic Benefit



CO2 Single Phase heat transfer better matches GT exhaust
→ Lower HRSG losses

Fundamentally different benefit vs High Temperature cycles

Image Courtesy of GE
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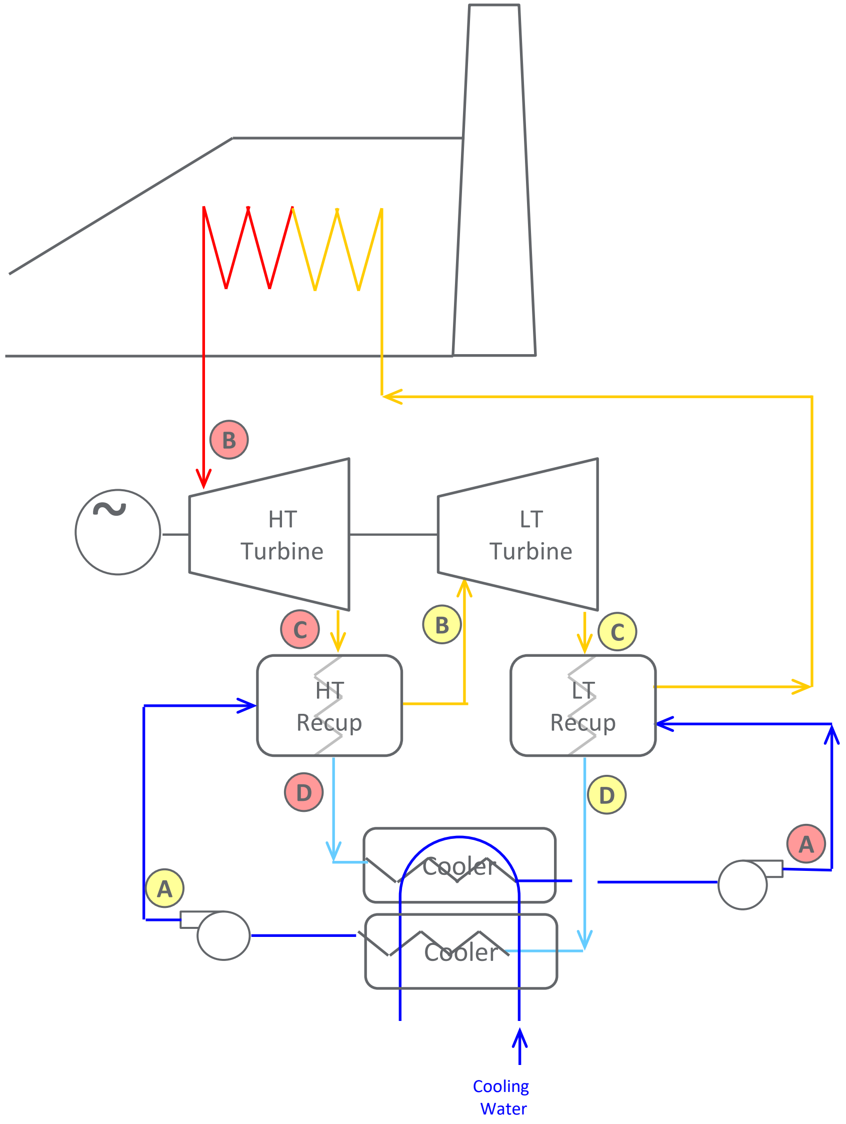
Cascaded CO_2 Bottoming Cycle



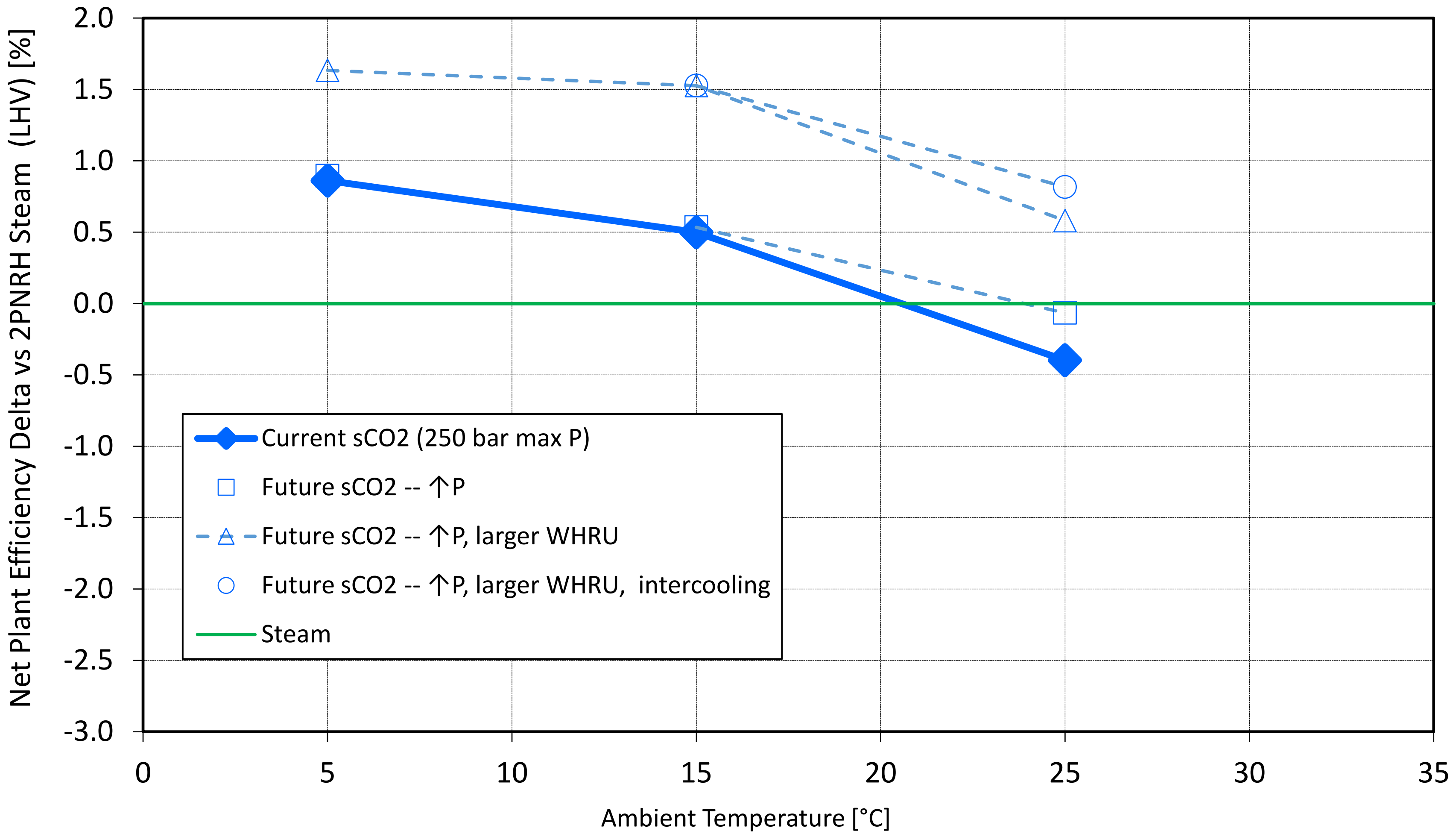
Cascaded cycle uses two sCO_2 loops to maximize performance

CO_2 circulates in two closed loop cycles – only small make-up for shaft seal leakages

Small equipment size enables BC to approach aeroderivative transient performance

Image Courtesy of GE

Impact of Ambient Temperature



GE data : P. Huck, M. Lehar, "Performance comparison of a supercritical CO2 versus steam bottoming cycles for aeroderivative gas turbines at various ambient temperature," ChemIndix, Nov 2016, Manama, Bahrain

Image Courtesy of GE
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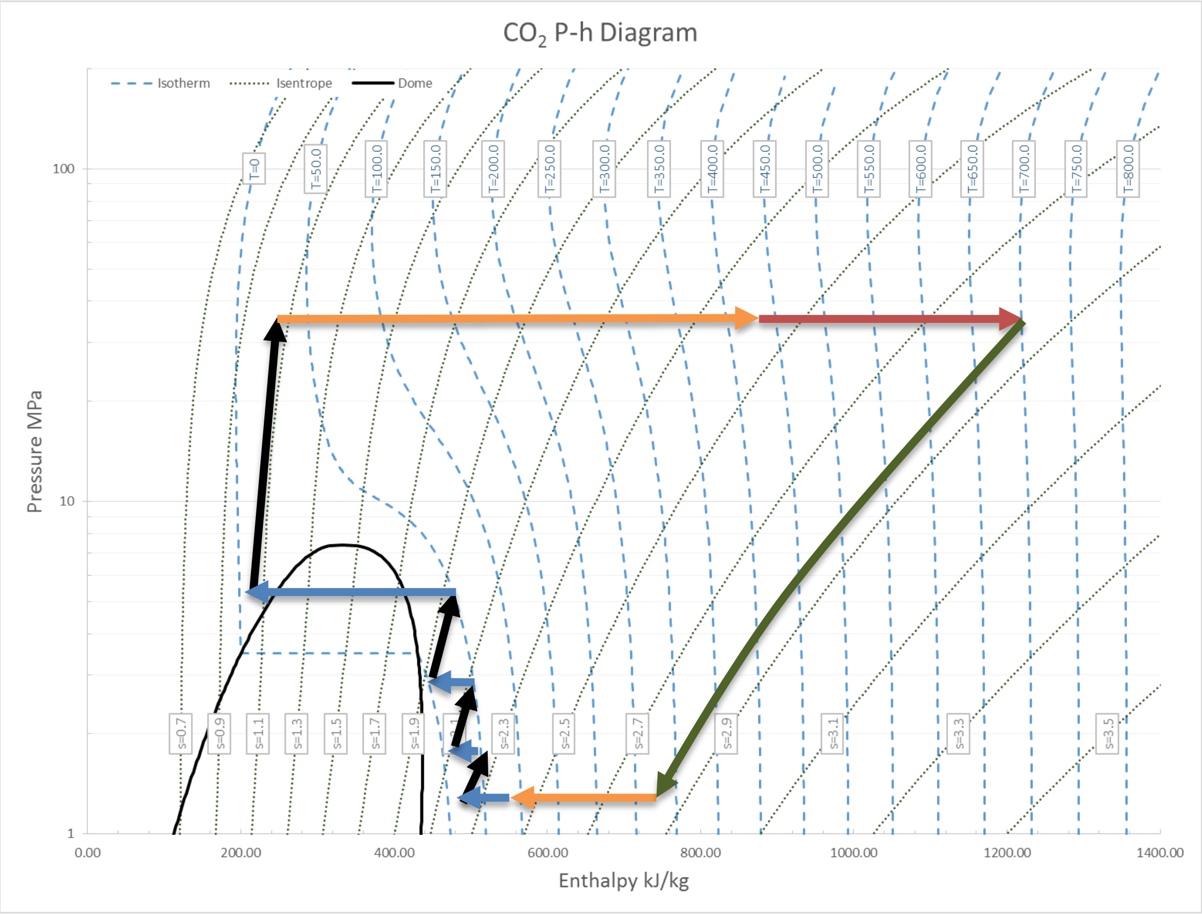
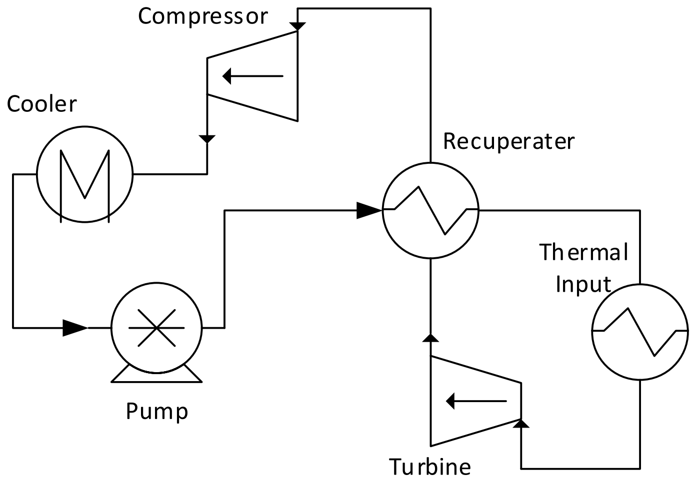
Condensing Cycles

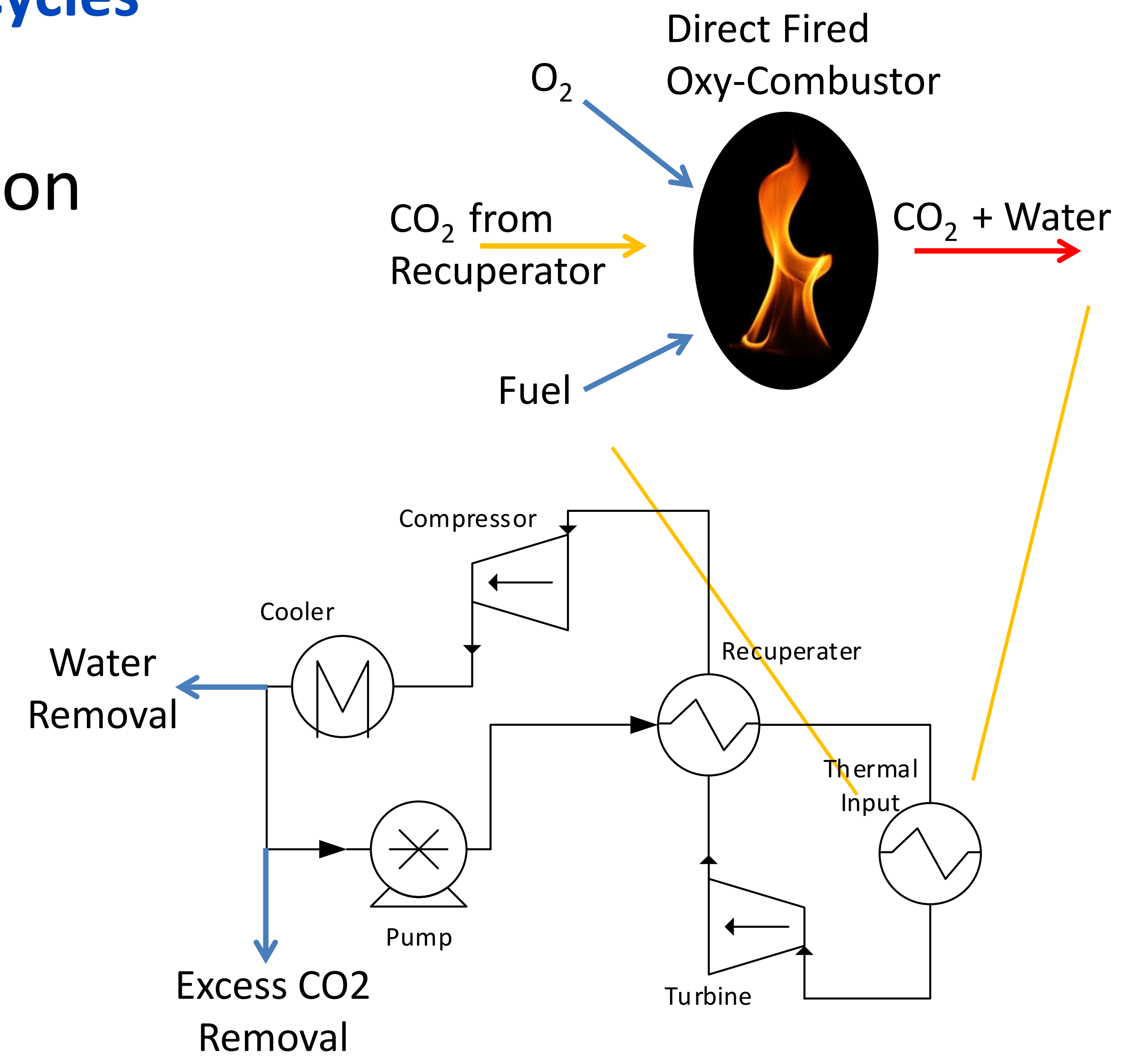


- Transcritical cycle
- Utilizes partial compression, condensation, and pumping to minimize compression work
- Must balance refrigeration / cooling requirements against compression requirements
- Can achieve efficiencies close to recompression cycle

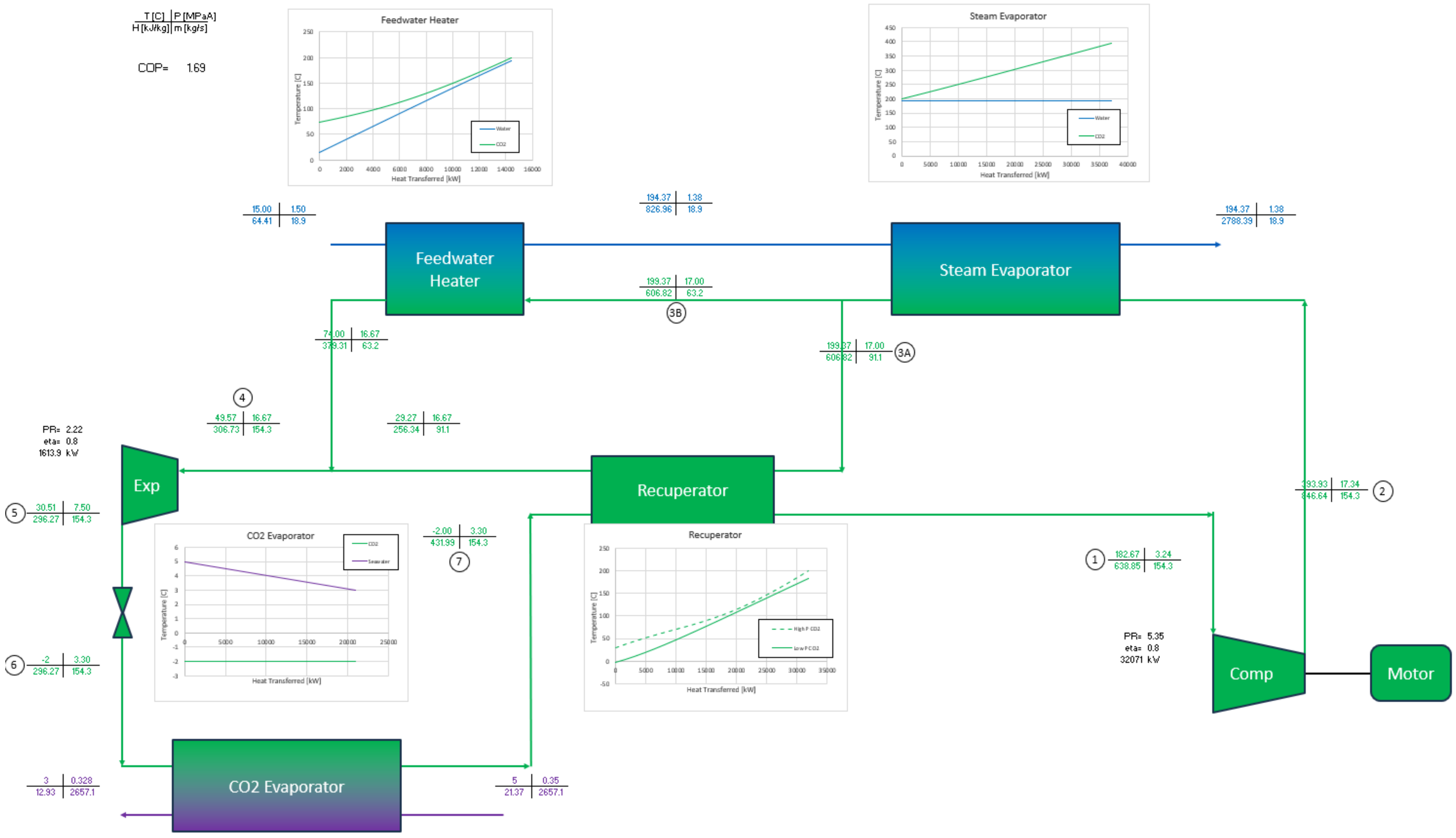


Allam (Direct Oxy-Fuel) Cycles

- Direct fire oxy-fuel combustion changes the cycle to a semi-open cycle
- Mass balance issues
- CO2 + combustion products
- Clean up and water removal



sCO_2 Heat Pumps for Industrial Heat and ETES



Cycle and component modeling

Cycle analysis

- System models that are used to:
  - Estimate and optimize overall system performance
  - Trade cost and performance
  - Provide boundary conditions for components
  - Evaluate operability
  - Design and optimize controls

Cycle model basics

- Fundamental tool to predict performance of a collection of components in a thermodynamic system
- Main principles
  - Conserve mass and energy
  - Define the components with adequate detail
  - Constrain the solution appropriately
  - If $N_{var} = N_{DOF}$, you are solving for a single unique solution
  - If $N_{var} > N_{DOF}$, you are solving a non-linear optimization problem

Optimization problems are the norm – usually performance and/or cost

Design models vs performance models

- Design model
  - Boundary conditions (e.g. heat source, heat sink) known at design point
  - Objective: Find arrangement of components that produces some form of optimal outcome
    - Performance
    - Cost
    - Other constraints (e.g. footprint, max P)
- Performance model
  - BCs cover range of values
  - Objective: With a given arrangement of fixed components, find optimal operating state of machine

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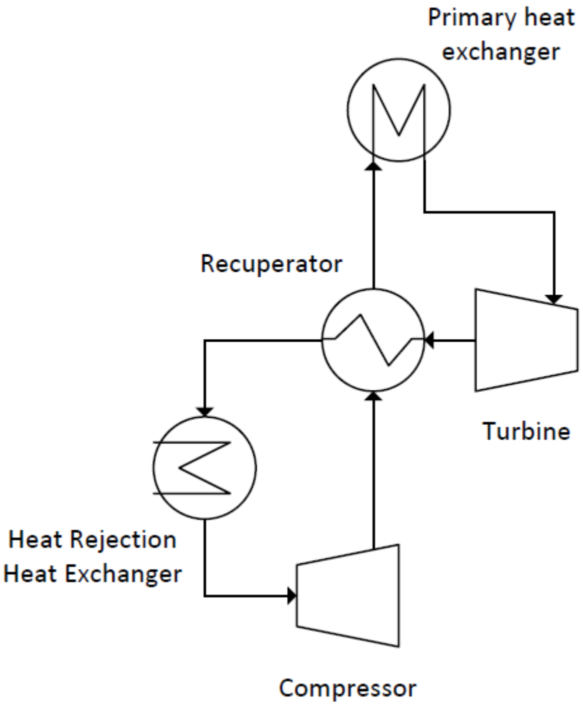
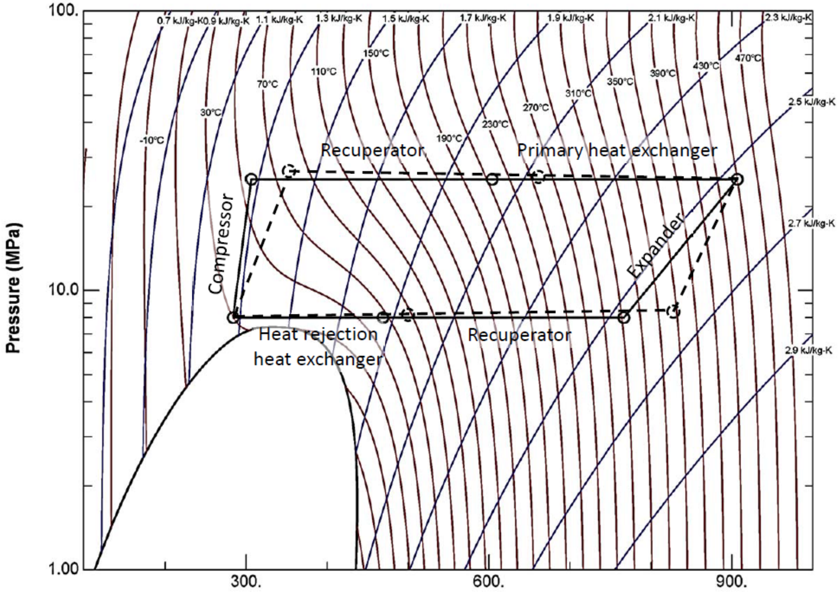
Example of a closed-loop cycle & components





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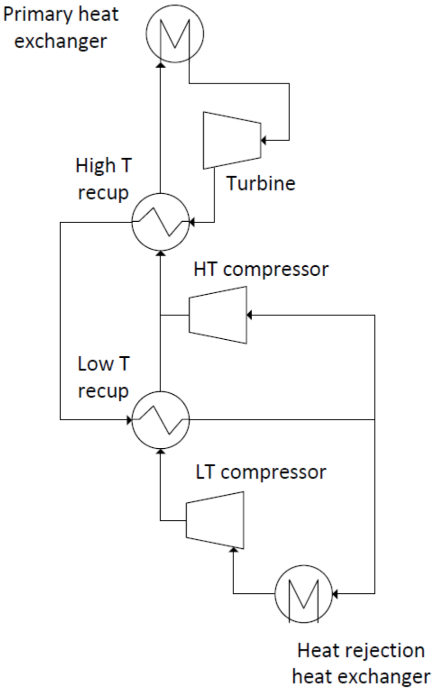
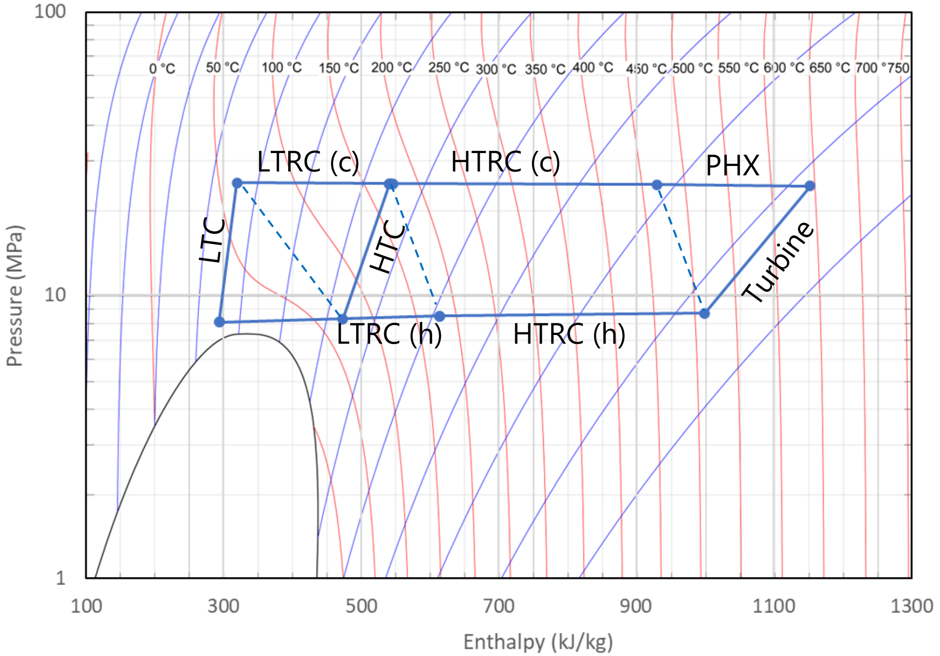
RCB cycle designed for recirculated heat applications with low ΔT





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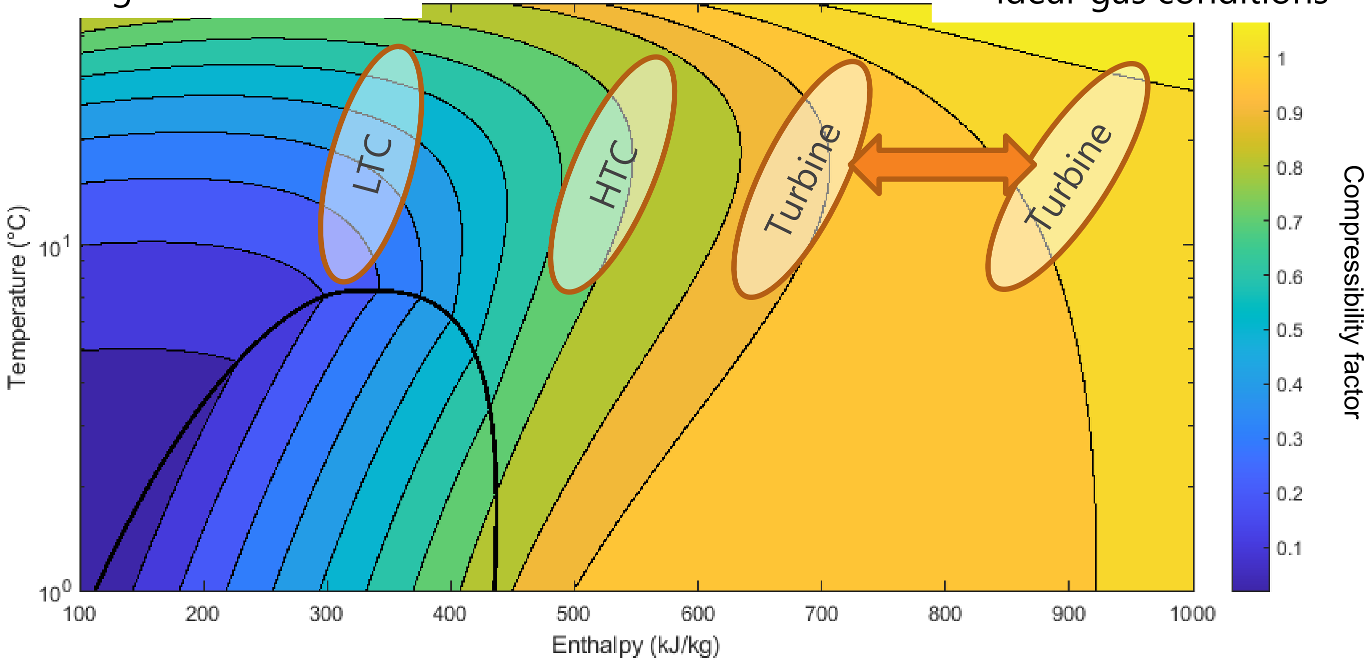
Component operating conditions affect how you model them

Compressors operate in highly non-ideal-gas conditions

Turbines operate in near ideal-gas conditions



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Compressor & turbine modeling

- Design mode
  - Inputs:
    - Pressure ratio
    - Flow rate
    - Inlet conditions
    - Isentropic (or polytropic) efficiency
  - Outputs
    - Shaft work
    - Discharge conditions

- Performance mode
  - Inputs:
    - Compressor map
    - Inlet conditions
    - Speed
  - Outputs
    - Shaft work
    - Discharge conditions

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Compressor in cycle design mode

- Know : p_1, T_1, p_2 and $\dot{m}$
  - $h_{2s} = h(p_2, s_1)^*$
  - $h_2 = h_1 + \frac{h_{2s} - h_1}{\eta_s}$
  - $T_2 = T(h_2, p_2)$
  - $W = \dot{m}(h_1 - h_2)$

- Compressor efficiency is an assumption – needs to be consistent with capability

** Simple compressible substance, with known EOS*

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Turbine in cycle design mode

- Know : p_1, T_1, p_2 and $\dot{m}$
  - $h_{2s} = h(p_2, s_1)$
  - $h_2 = h_1 - \eta_s(h_{2s} - h_1)$
  - $T_2 = T(h_2, p_2)$
  - $W = \dot{m}(h_1 - h_2)$

- Turbine efficiency is an assumption – needs to be consistent with capability

Polytropic vs isentropic efficiency

- Thermodynamics & cycle modeling wants isentropic efficiency
- Compressor designers talk in "polytropic efficiency" (efficiency for an incremental change in pressure)
- Can convert between them:
  - $\eta_s = \frac{1-PR^{\eta_p \frac{\gamma-1}{\gamma}}}{1-PR^{\frac{\gamma-1}{\gamma}}}$

  but what do you use for γ when your fluid is not an ideal gas?

  (hint, it's not $\frac{c_p}{c_v}$)

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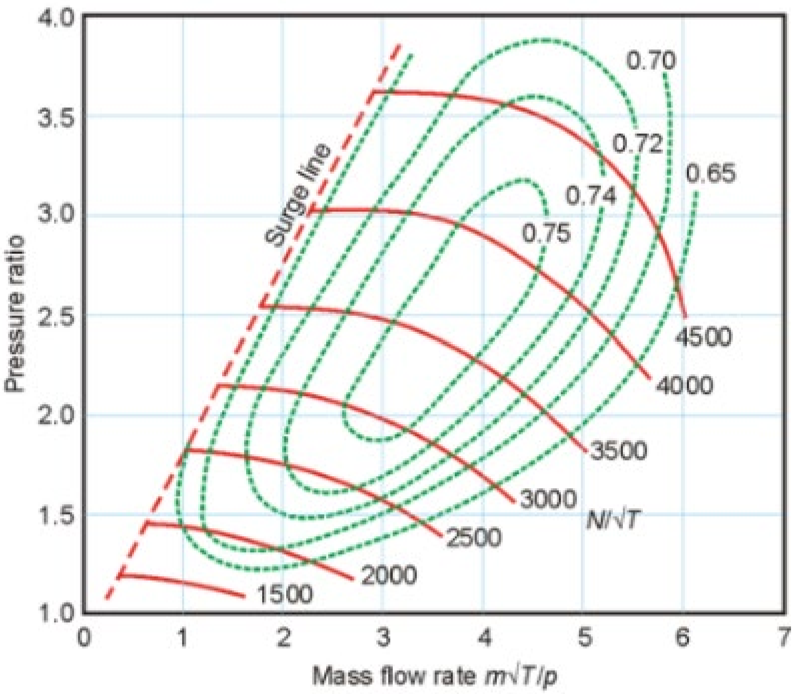
Compressor performance modeling – maps



From ConceptsNREC website

Given inlet conditions and speed, can output pressure ratio and efficiency

Standard approach bakes in ideal gas assumptions

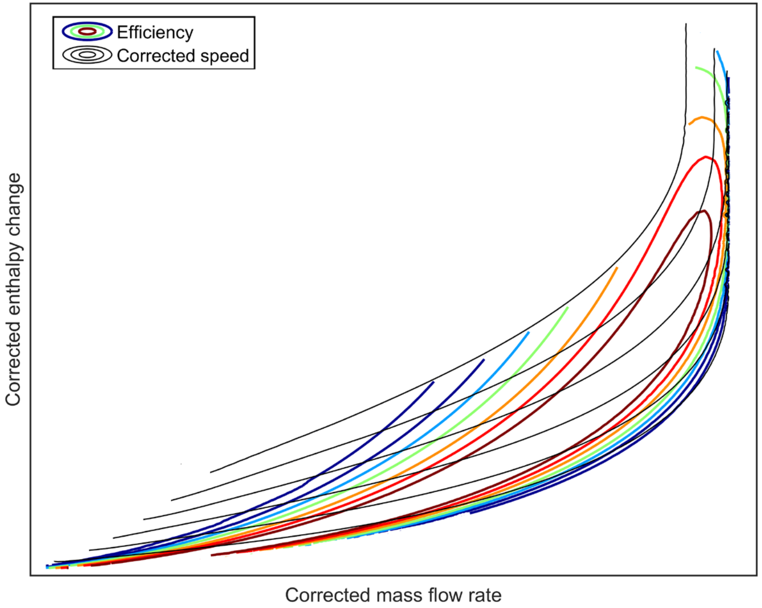
Real gas (and dense-ish) fluid effects are more subtle. See, for example:

Pham, H. S. *et al.*, 2016, International Journal of Heat and Fluid Flow, **61**, pp. 379–394.

Variable inlet guide vanes add another dimension to maps

ECHOGEN

Turbine performance modeling – maps



Given inlet conditions and speed, can output pressure ratio and efficiency

Standard approach bakes in ideal gas assumptions

Real gas fluid effects are more subtle. See, for example, BNI's approach documented in Appendix B of:

> Gavagnin, G., *et al.*, 2018, Applied Energy, **231**, pp. 660–676, url=https://linkinghub.elsevier.com/retrieve/pii/S030626191831451X

Variable inlet guide vanes add another dimension to maps

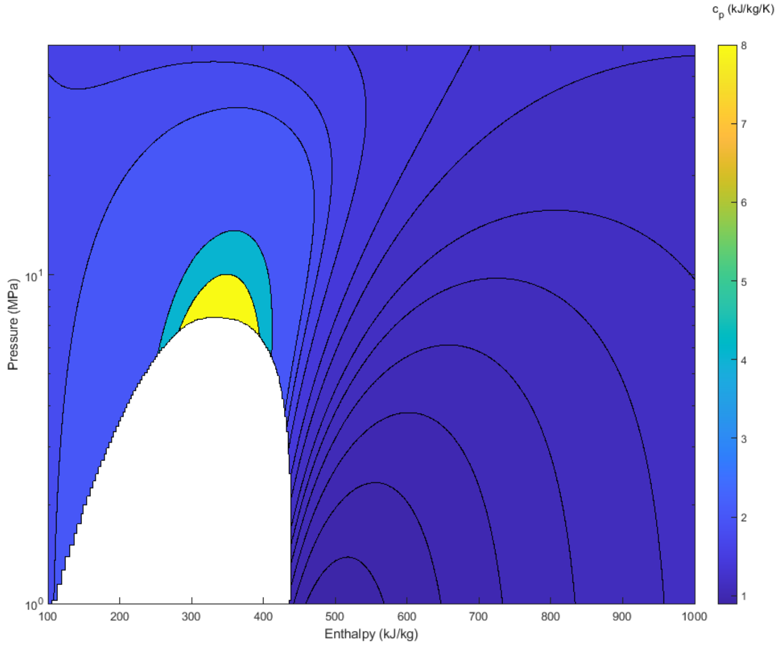
Heat exchangers

- 3 major categories:
  - Recuperators: Internal heat transfer (CO_2 to CO_2)
  - External heat addition: X to CO_2
  - External heat rejection: CO_2 to ambient (air or water)

- Common types for CO_2:
  - PCHE/DBHE
  - Multi-bank finned tube
  - Shell & tube

Heat exchanger component models

- Traditional approaches:
- Effectiveness/NTU
- UA LMTD
  - Both only work under constant c_p assumption
  - Bad assumption for CO_2 most of the time



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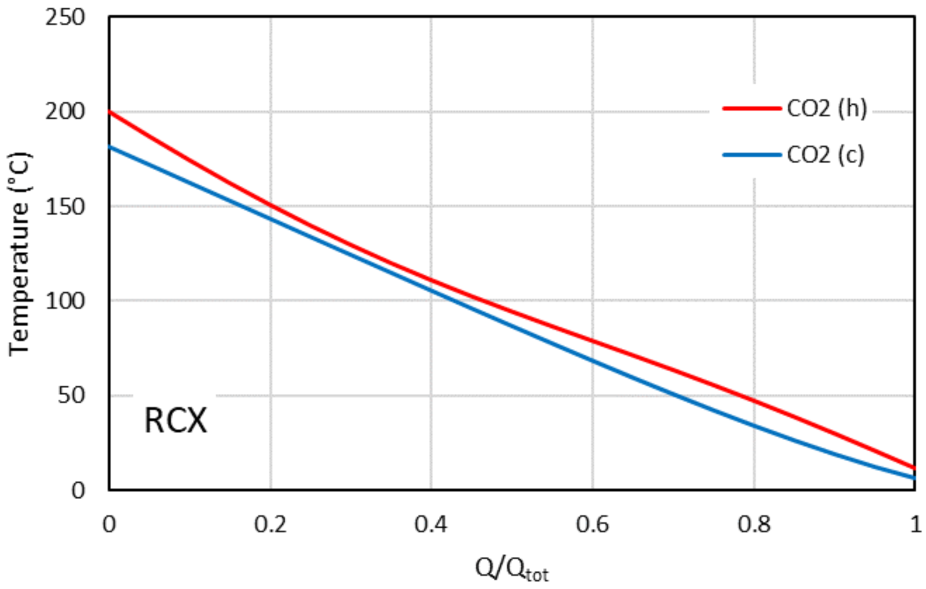
Heat exchanger design point modeling - basic concepts

TQ plot example:

- Slope of curve is proportional to $(\dot{m}c_p)^{-1}$
- Ideal fluid with constant c_p, curves would be straight lines
- Real fluids have curves, and if the streams cross…



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Heat exchanger design point modeling – more basics

- Heat transfer can be modeled as:

$$Q_h = -Q_c = \frac{\Delta T}{R} = UA\,\Delta T,$$

where R is a thermal resistance term that includes heat transfer area

and UA is the “conductance” of the heat exchanger

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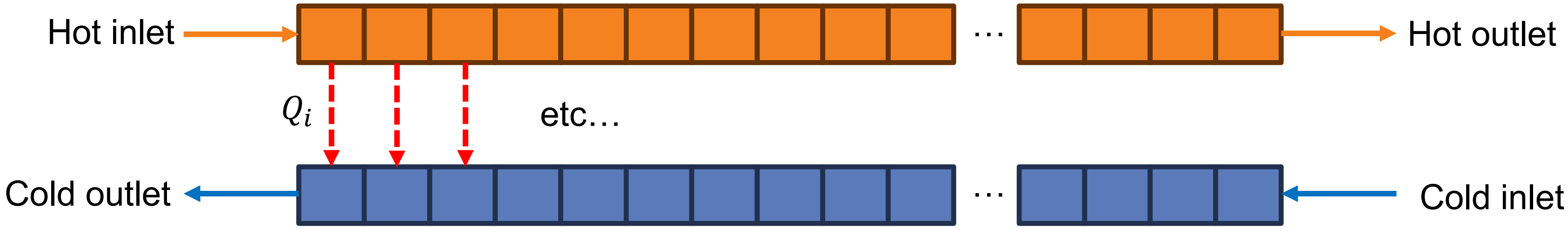
Heat exchanger design point modeling – one way

- Divide (discretize) heat exchanger into I segments



- By making I large enough, c_p of the fluids are reasonably constant in each interval, so we can write

$$Q = \sum_{i=1}^{I} dQ_i = \sum_{i=1}^{I} UA_i \Delta T_i$$

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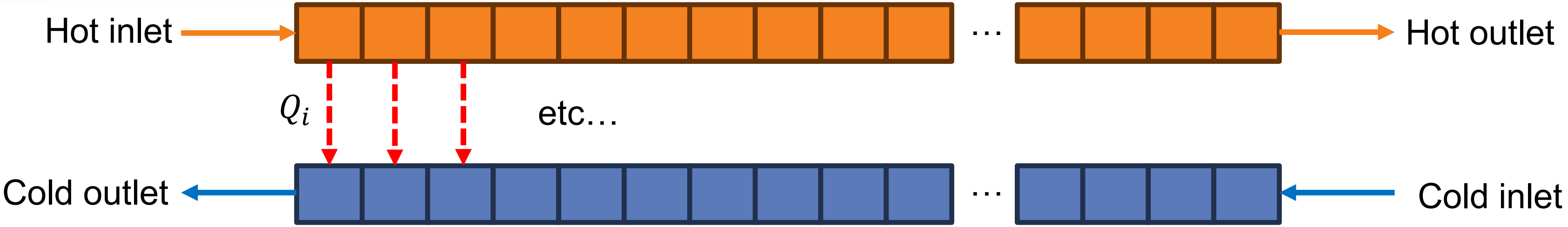
Heat exchanger design point modeling – Design mode

- Option 1: Specify a UA target
  - Given a set of inlet conditions $(p, T, \dot{m})$, assume a set of outlet conditions such that $Q_h = -Q_c$
  - Calculate $Q_{total} = \dot{m}\Delta h$
  - Divide both sides into I segments of equal Q_i
  - Calculate $T_i = T(p, h_i)$
  - From preceding slide, $UA = \sum UA_i = \sum \frac{dQ_i}{\Delta T_i}$
  - Iterate on outlet conditions until UA=target value
- Other options include setting an effectiveness target or minimum temperature difference – iterative processes are similar
- Specified dP – check with suppliers to find out what is reasonable

Heat exchanger design point modeling continued



- Concept only applies to counterflow geometry
  - DBHEs are close to pure counterflow, but not exactly
- Cross-flow heat exchangers are common in air-to-CO_2 applications
  - If number of passes is > 3 or 4, counterflow isn't a bad approximation
  - More complex discretized solutions are possible, but might not be worth the effort – consider an effectiveness limit based on NTU

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Heat exchanger modeling – performance

- Similar to design-point specified-UA technique
- Correct UA to off-design conditions

$$R = R_{conv,h} + R_{cond} + R_{conv,c}$$

$$R_{conv,h} = \left(h_{conv,h} A_{h,h}\right)^{-1}$$

...etc

using your favorite heat transfer correlation(s) $\left(Nu = hd/k = f(Re, Pr, \ldots)\right)$, ratioed to design point

- Correct dP using a similar approach:

$$dP = \frac{f(Re)}{d}\frac{\rho}{2}U^2 L$$

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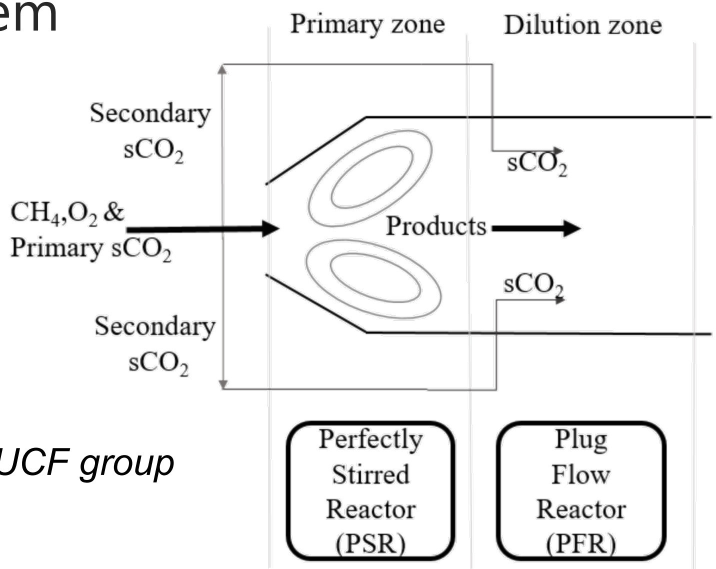
Oxy-fuel combustion in CO_2

- Standard equilibrium solvers (e.g. Chemkin, ASPEN) will manage $T(h_{in}, X_f, X_o)$ problem
- Kinetics simulations – wide-open problem!



From UCF group

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Other modeling tips

- Heat losses can be significant, especially in small systems – consider including in modeling
- Piping pressure drops are also important, model similarly to heat exchanger dP
- Control valves are frequently necessary to adjust flow splits and have non-zero dP even when full open
- Secondary / auxiliary loads need accounting
  - Cooling fans, pumps, etc.
  - Seal gas conditioning systems
  - Lube oil cooling
  - Generator, gearbox, VFD, switchgear losses

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Fluid properties

- Span & Wagner EOS for CO_2 is industry standard
- New EOS has been created (Harvey and Lemmon, 2022 sCO_2 Symposium) – release date TBD (but not imminent)
- REFPROP, CoolProp both use S&W
  - Generally good for pure CO_2
  - Mixture properties less certain
  - Execution speed can be problematic
  - 2-D table lookups a common approach
    - Interpolation behavior near saturation line can be a problem
    - $h_{lookup}=h(T_{spec},P)$, $T_{lookup}=T(h_{lookup},P)$, … $T_{lookup}-T_{spec} \neq 0$!
      - Not necessarily the end of the world, but can be a problem if you expected that behavior

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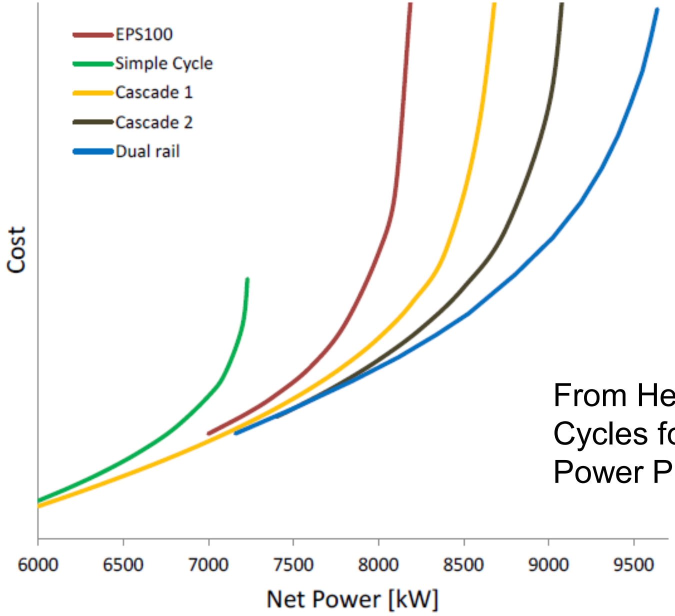
Cost modeling

- Core function of practical modeling – cost/performance trades
  - Key cost variables:

    Cycle complexity
    Heat exchanger sizing



From Held, 2015, "Supercritical CO_2 Cycles for Gas Turbine Combined Cycle Power Plants," *Power Gen International*.

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Cost modeling (continued)

- Typical cost models:

$$C = A + B \cdot SP^n$$

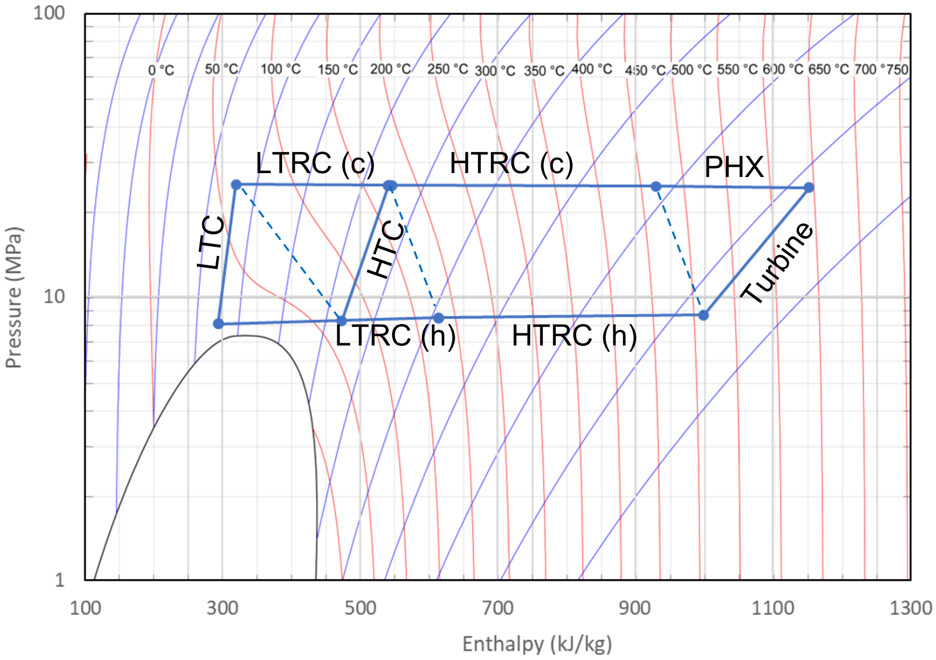
where "SP" is a scaling parameter (e.g., UA for heat exchangers, power output for turbines, etc.), A, B and n are fit parameters

- Weiland paper is an excellent source for fit parameters – recognizing that 2024 prices are 1.3 to 1.5X vs 2019

Weiland, Lance and Pidaparti, "sCO_2 Power Cycle Component Cost Correlations from DOE Data Spanning Multiple Scales and Applications," *ASME Turbo Expo 2019,* Paper GT2019-90493

Modeling tools and techniques

- Hand draw on a PH diagram – great way to get into the ballpark



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Modeling tools

- Spreadsheet models
  - Good way to get started, teaches you how to set up a problem, what's missing
  - Anything more complicated than an unrecuperated cycle will require iteration
  - Potentially error-prone
- MATLAB, Python
  - Largely still "roll-your-own" approach
  - Much larger selection of optimization methods
  - Also error-prone

State	Outlet	Inlet	h (kJ/kg)	P (MPa)	T (°C)	s (kJ/kg·K)	w (kg/s)	Q/W (kW)
1	ACC	LTC	293.60	8.05	31.73	1.303	599.28	-107687
2	LTC	LTRc	319.78	24.99	62.15	1.312	599.28	15690
31	LTRc	Mix	546.07	24.79	175.36	1.900	599.28	135611
32	HTC	Mix	540.83	24.79	172.05	1.888	363.18	
33	Mix	HTRc	544.09	24.79	174.11	1.896	962.46	
34	HTRc	PHX	929.87	24.59	468.17	2.566	962.46	371299
4	PHX	HPT	1153.06	24.34	645.93	2.838	962.46	214809
5	HPT	HTRh	999.45	8.65	512.82	2.858	962.46	-147843
54	HTRh	LTRh	613.67	8.45	178.86	2.225	962.46	-371299
6	LTRh	ACC	473.29	8.25	70.13	1.871	962.46	-135111
32	Compr outlet (check		540.83	24.79	172.05	1.888	363.18	24530

Wt	-147843	kW		**Aux loads & losses**	
Whtc	24530	kW		Generator	-1478
Wltc	15690	kW		Motor/VF	-402
Wnet	100643	kW		Misc	-50
Qh	214809	kW		Heat sour	-512
Qc	-107687	kW		Coolant	-4538
Eta	46.85%			Total	-6981
UAltr	21636	0.976352			
UAhtr	15395	0.985902			
UAacc	18907	0.945277			
UAphx	1648	0.575364			

Modeling tools - Constraints

- Non-linear optimizers need boundaries – and a good initial guess

Variables	x	LB	UB	x0
P1	8.05	7	12	8
P2	24.99	10	25	25
T1	31.73	28	60	30
T31	175.36	100	400	170
T34	468.17	200	600	450
T4	645.93	600	650	640
wltc	599.28	5	800	600
whtc	363.18	0.1	400	350

Constraints		LB	UB
LTC margin	1.052	0.3	
UALTR	21636		40000
UAHTR	15395		30000
UAACC	18907		20000
UAPHX	1648		10000
Wnet	100643	100000	100001
HX cost	52,970,721		54,000,000

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Modeling tools – not an exhaustive list!

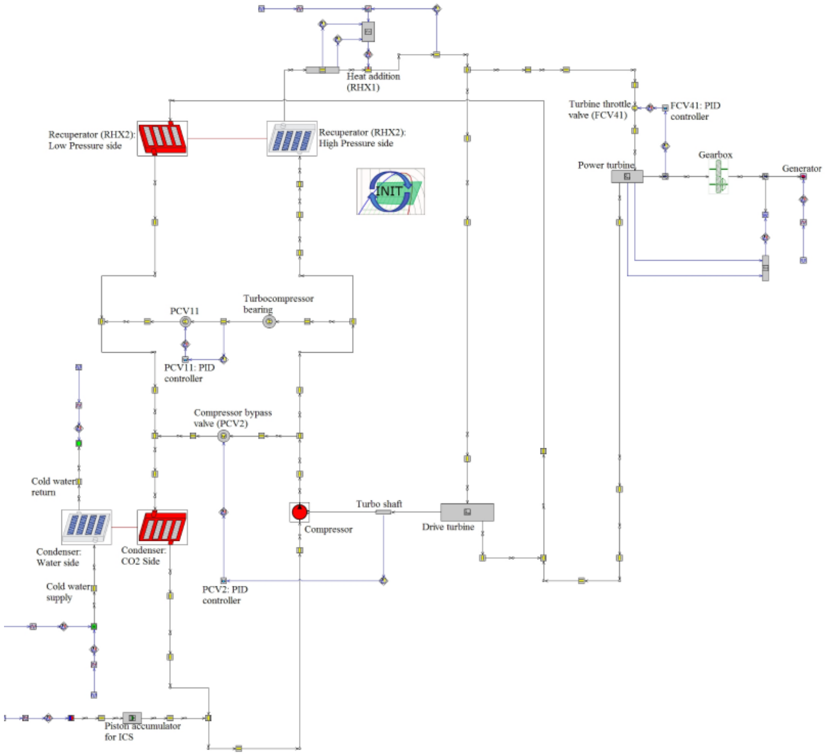
- Commercial codes
  - ASPEN, HYSYS, Ebsilon, GateCycle, IPSEPro, AxSTREAM, CYCAL, NPSS, Flownex, Modelon
- Open-source codes
  - OpenMDAO/PyCycle, T-MATS, others?

Transient modeling

- Simulation purposes:
  - System responses to changes in power demand, heat supply, heat sink temperature
  - Control system design and optimization
  - Startup/normal shutdown
  - Event simulation – e-stops, PRV releases, pipe ruptures
  - Define transient BCs for components, transient stress analysis
- Significantly more challenging numerically, computationally
- Requires at least a rudimentary control system layout
- Requires component off-design models
- Steady-state models + transient terms
  - Thermal inertia
  - Mechanical inertia
  - Volume dynamics (fluid inertia)

Transient model example

- Model construction similar to steady-state model
- Setting up BCs and ICs crucial to finding valid solution



From Avadhanula and Held, "Transient Modeling of a Supercritical CO_2 Power Cycle and Comparison with Test Data," *ASME Turbo Expo 2017, GT2017-63279*

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Transient modeling tools – not an exhaustive list!

- Commercial codes
  - Aspen, NPSS, Flownex, Modelon, GT-SUITE, AME-Sim, TRNSYS, MATLAB-Simulink, Dymola
- Open-source codes
  - Modelica (language, not a program)

Q&A