8^{th} sCO_2 Symposium

San Antonio, TX February 26-29, 2024

Cycles Tutorial

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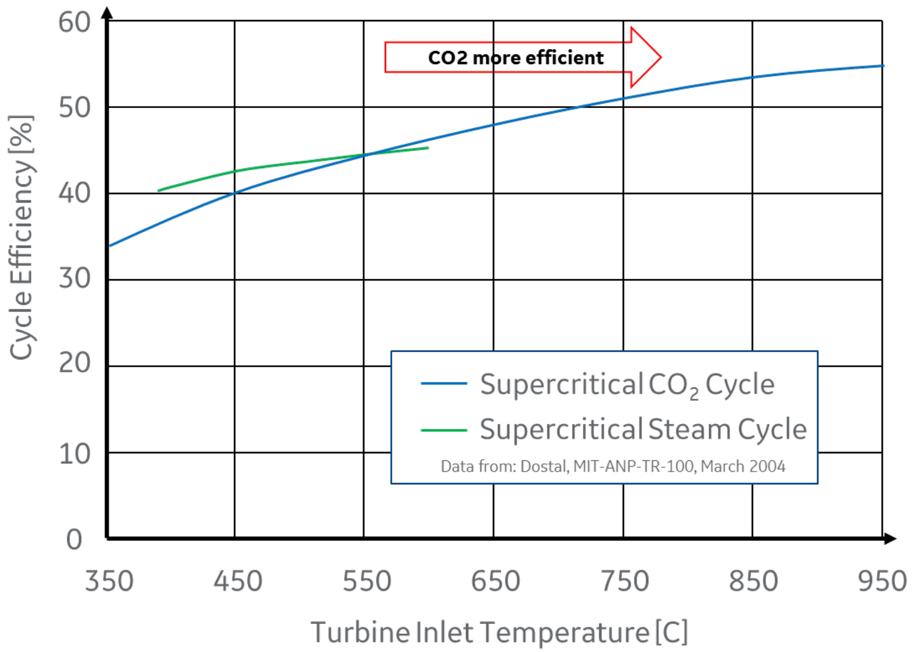
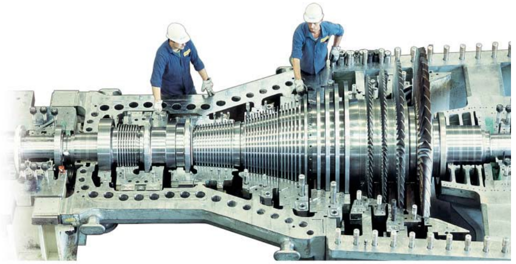
Motivation for CO_2 as a Working Fluid

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

Slide Courtesy of GE
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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.

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

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.

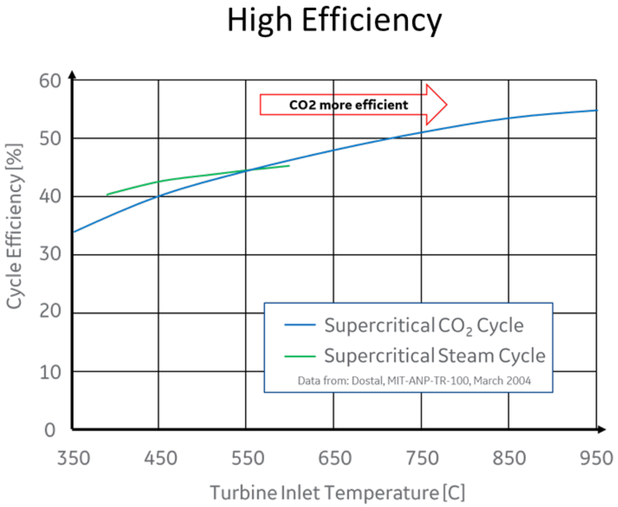
Nomenclature

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

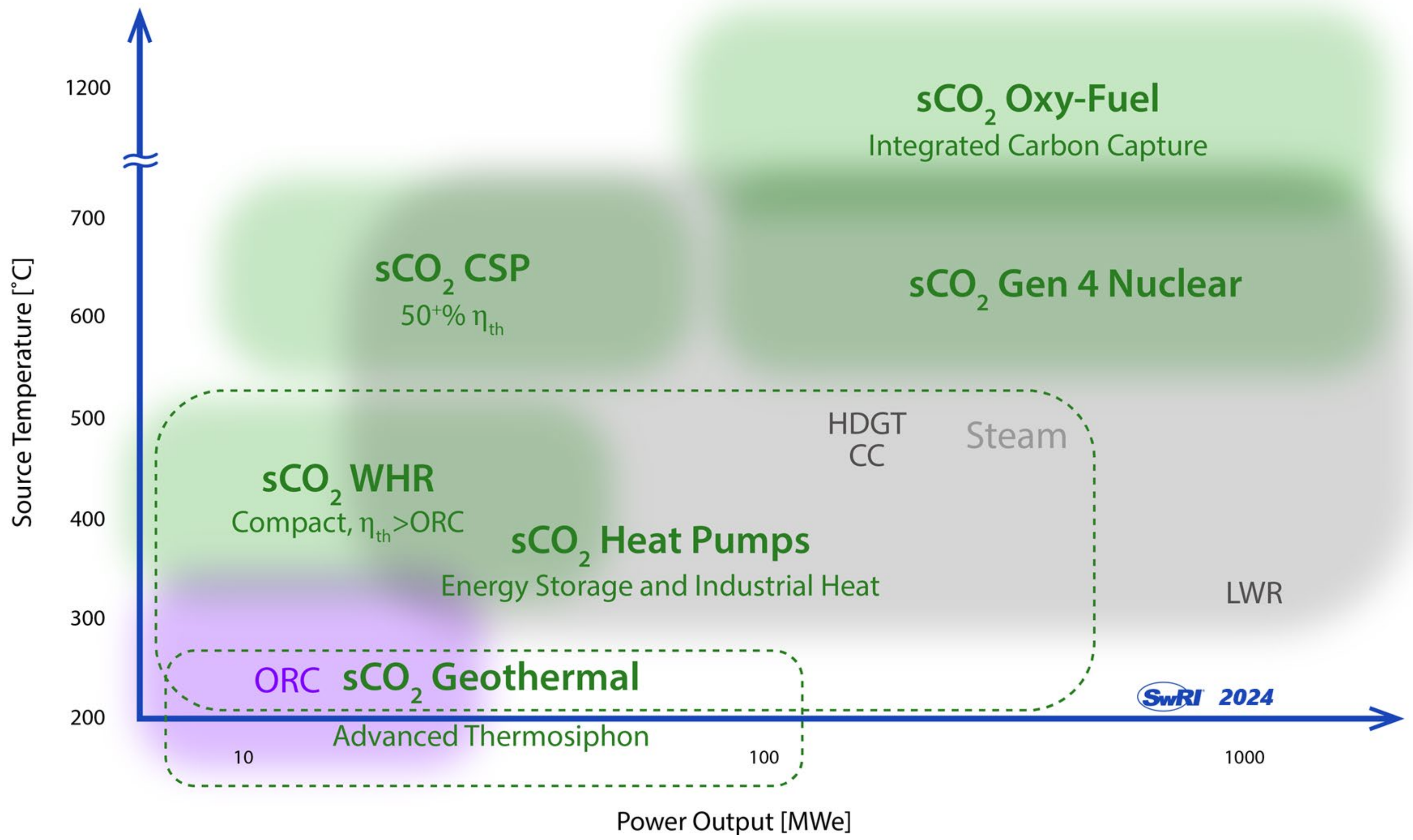
Subscripts

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

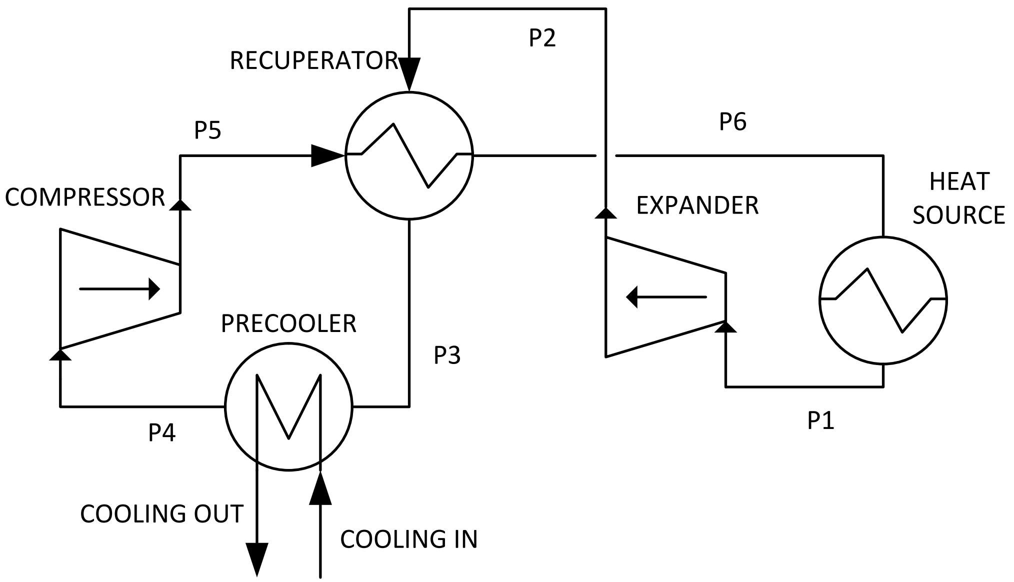
Dostal 2004

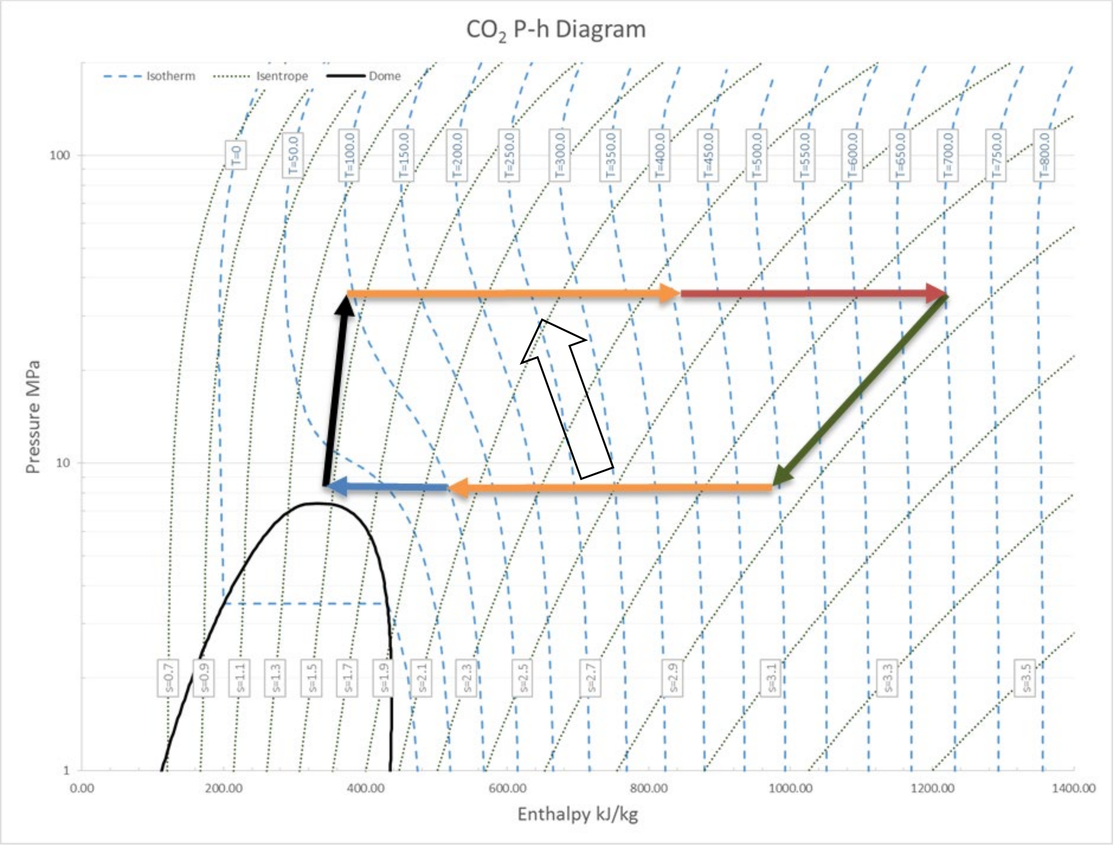


sCO2 Application Space



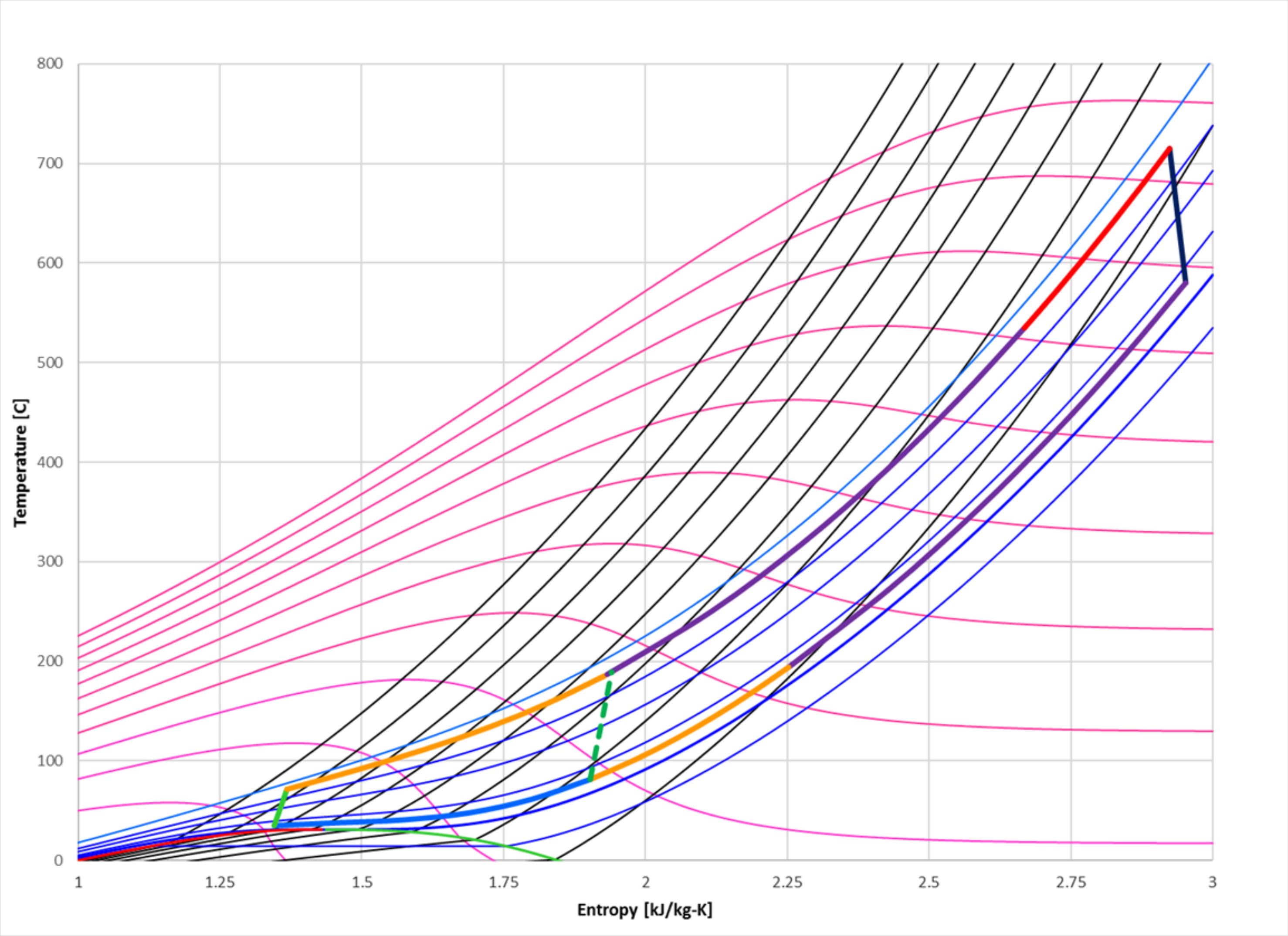
Basic Components in a Simple Recuprated Cycle





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
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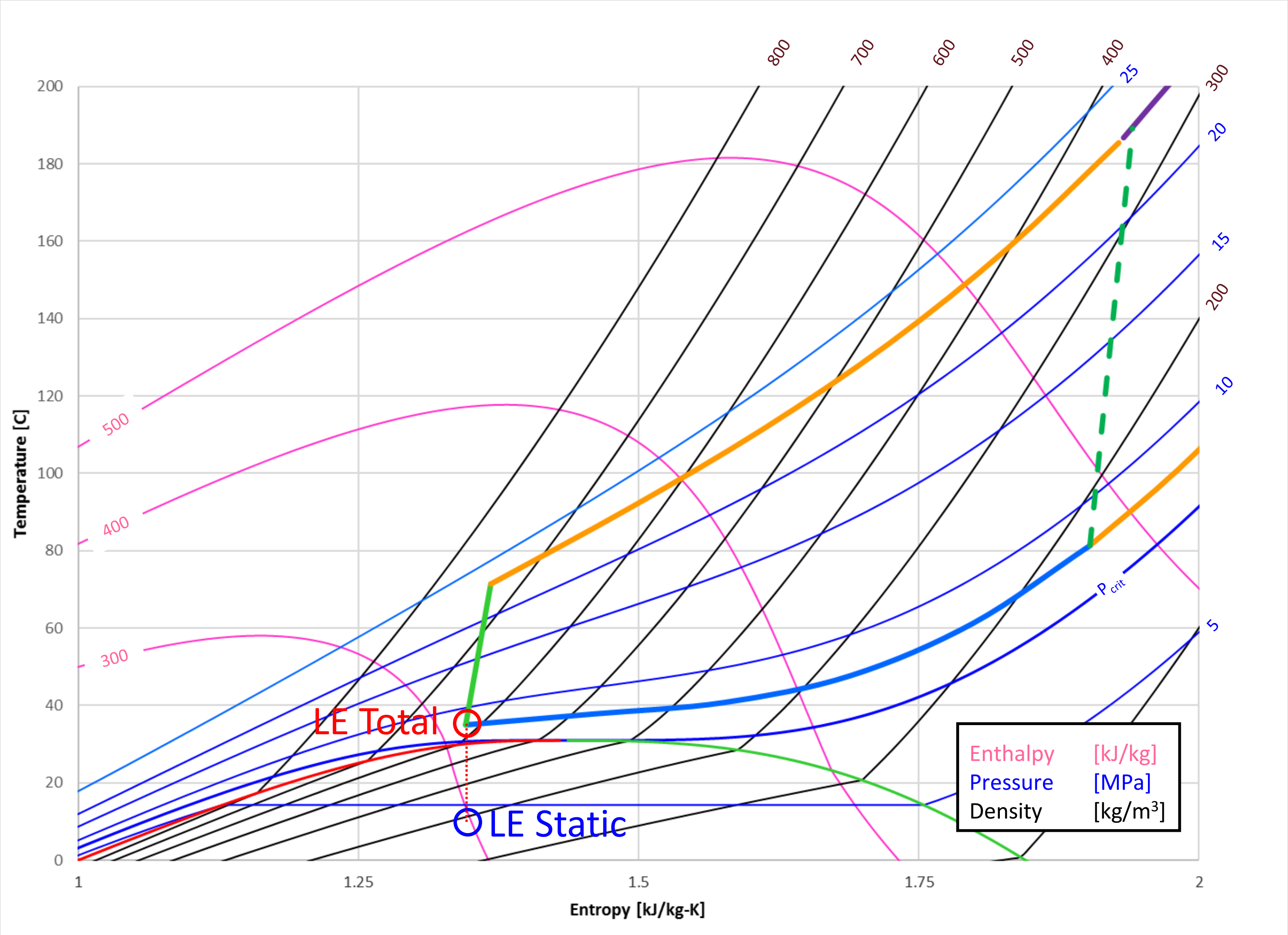
What's Special about CO_2?



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?

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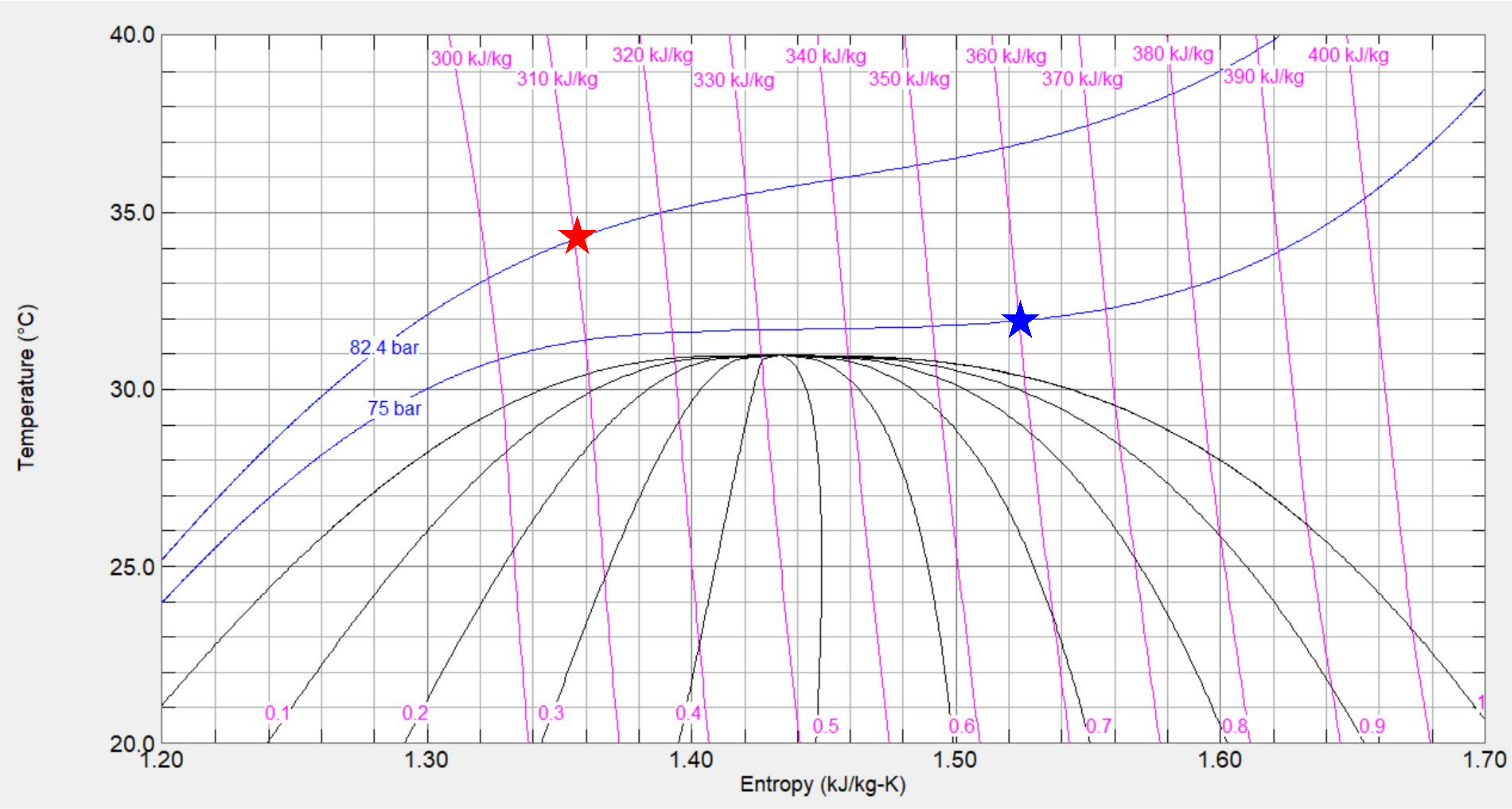
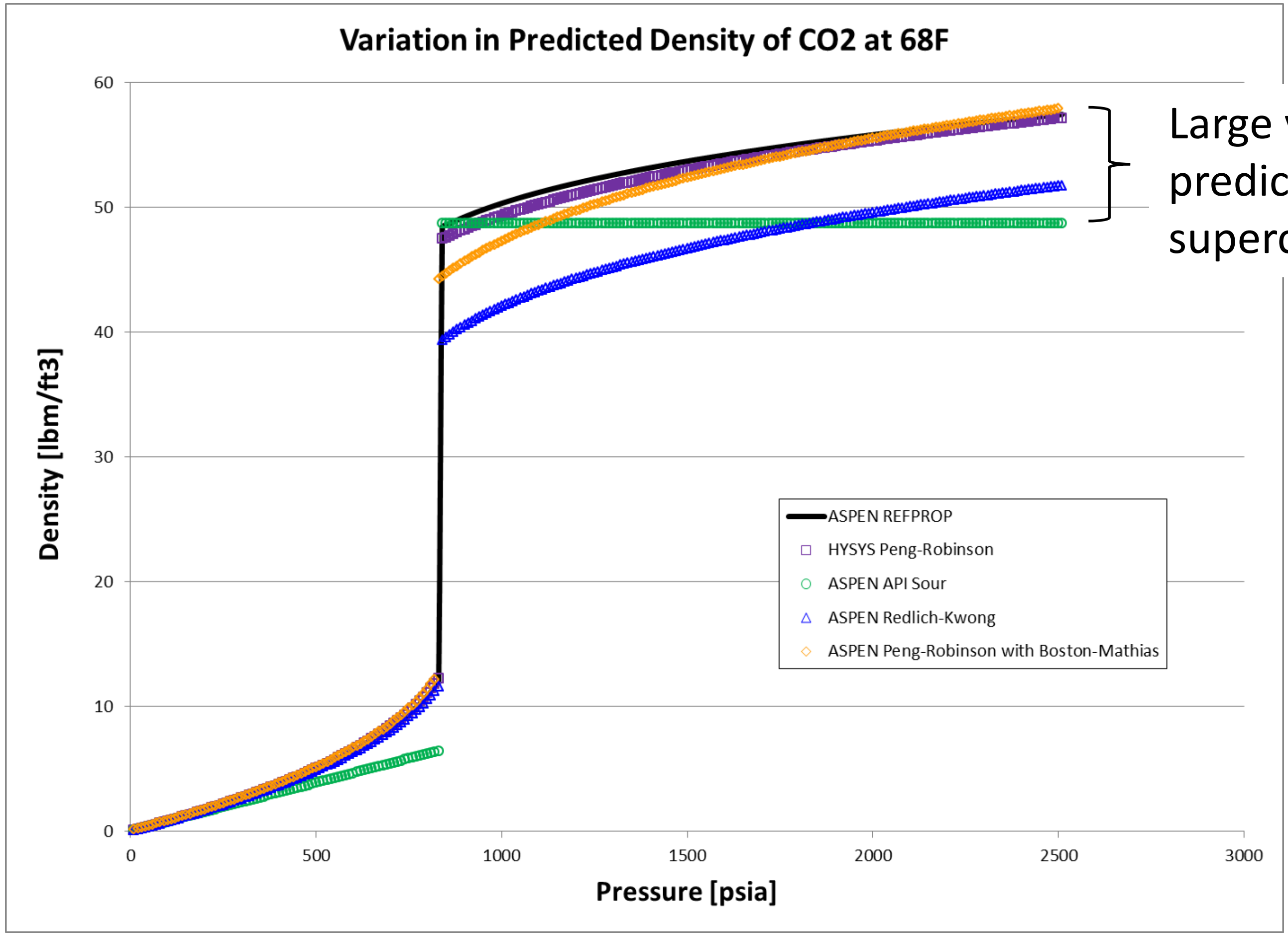


Common CO_2 Equations of State



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

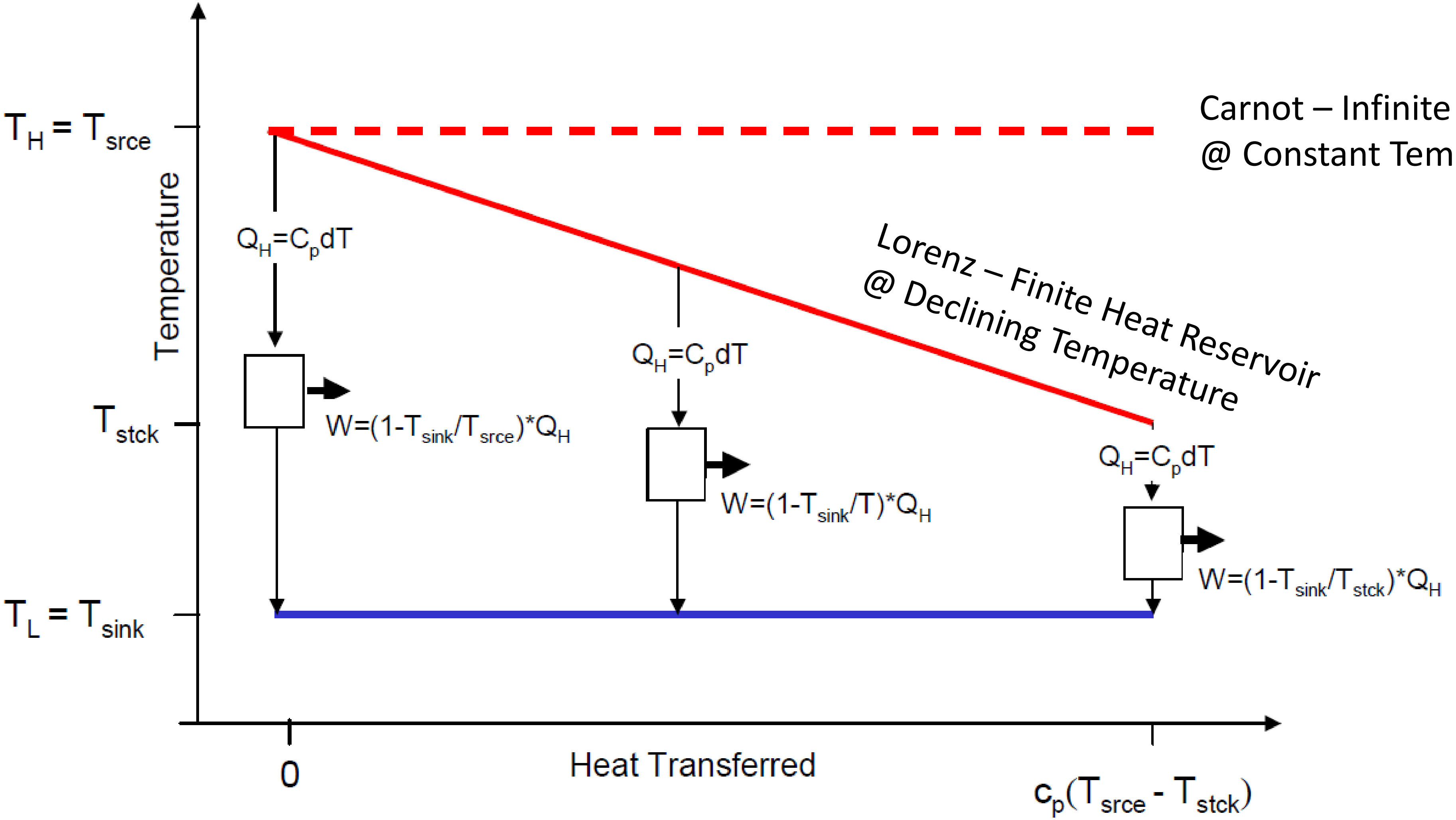


Image from Hofer & Gulen Efficiency Entitlement for Bottoming Cycles GT2006-91213

Lorenz 1895

Die Ermittelung der Grenzwerte der thermodynamischen Energieumwandlung.

Von Dr. H. Lorenz, Ingenieur in München.

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

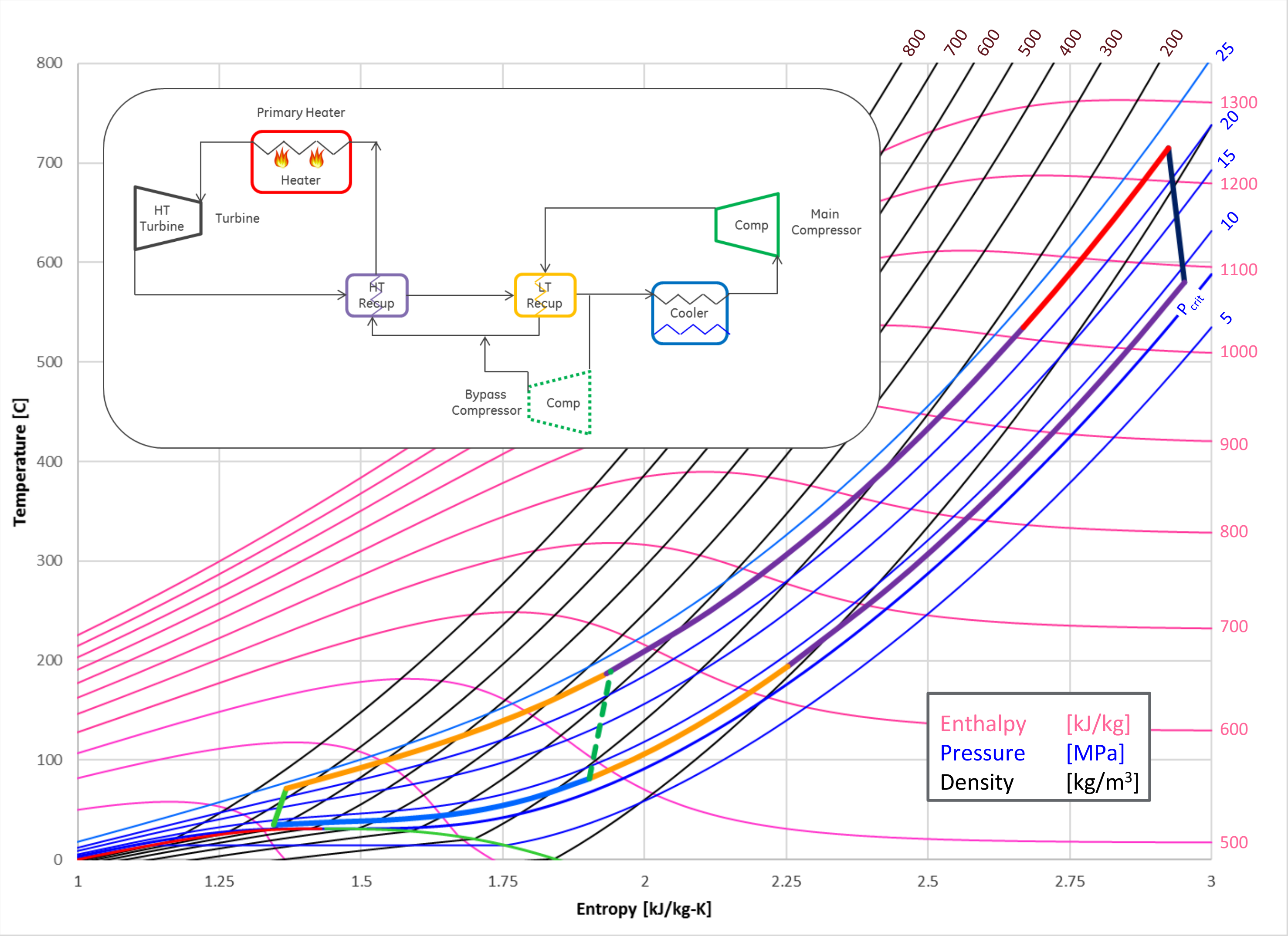
oder mit Rücksicht auf (11)

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

Applications and Architetures

High Temperature Cycles

sCO_2 Recompression Cycle



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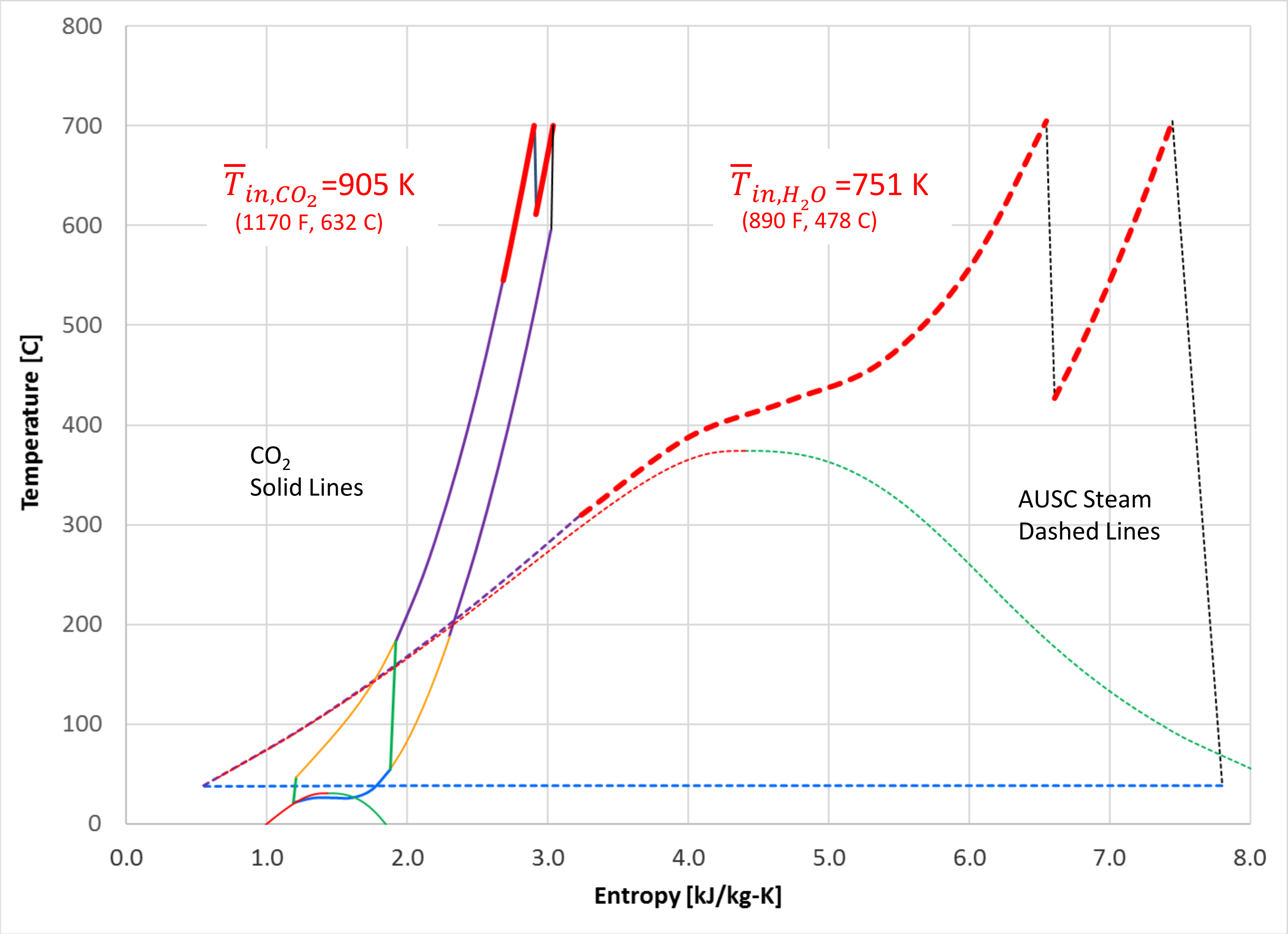
High Temperature Cycles – Thermodynamic Benefit



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

Image Courtesy of GE
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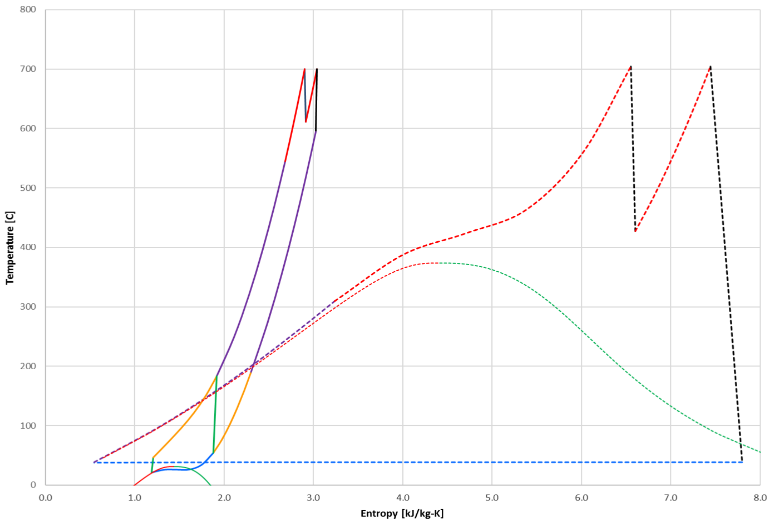
High Temperature Cycles – Flow Differences



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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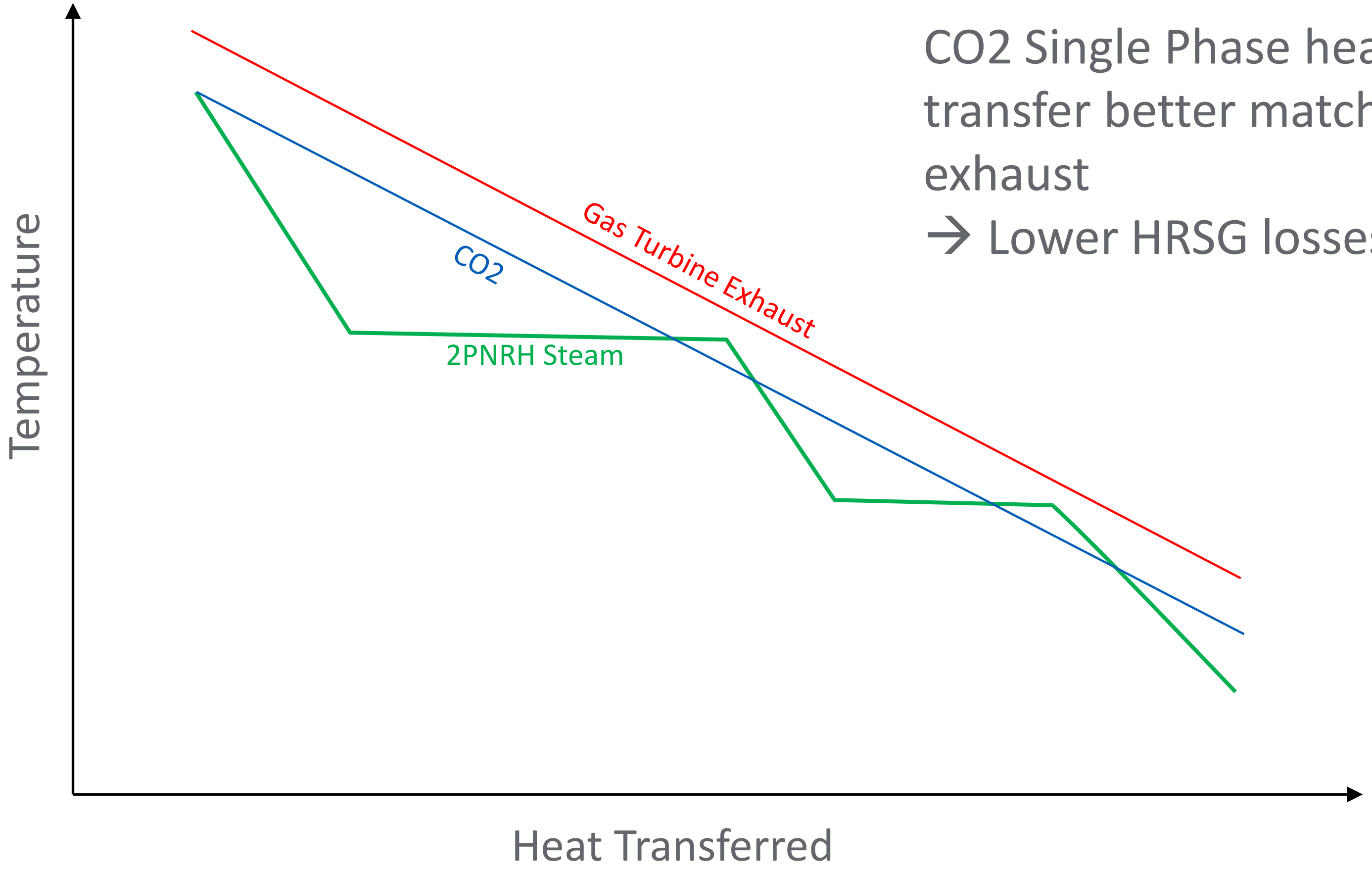
Waste Heat Recovery

Waste Heat Recovery – Thermodynamic Benefit



CO_2 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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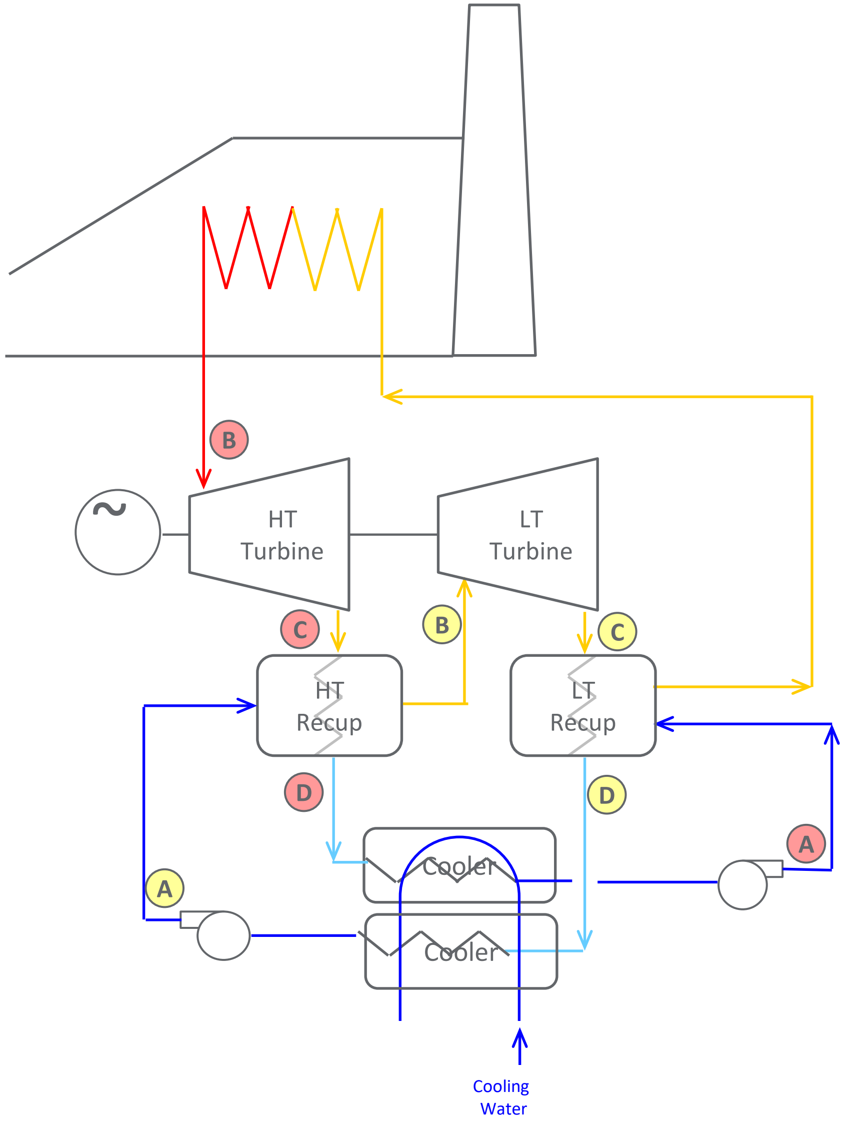
Cascaded CO_2 Bottoming Cycle



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
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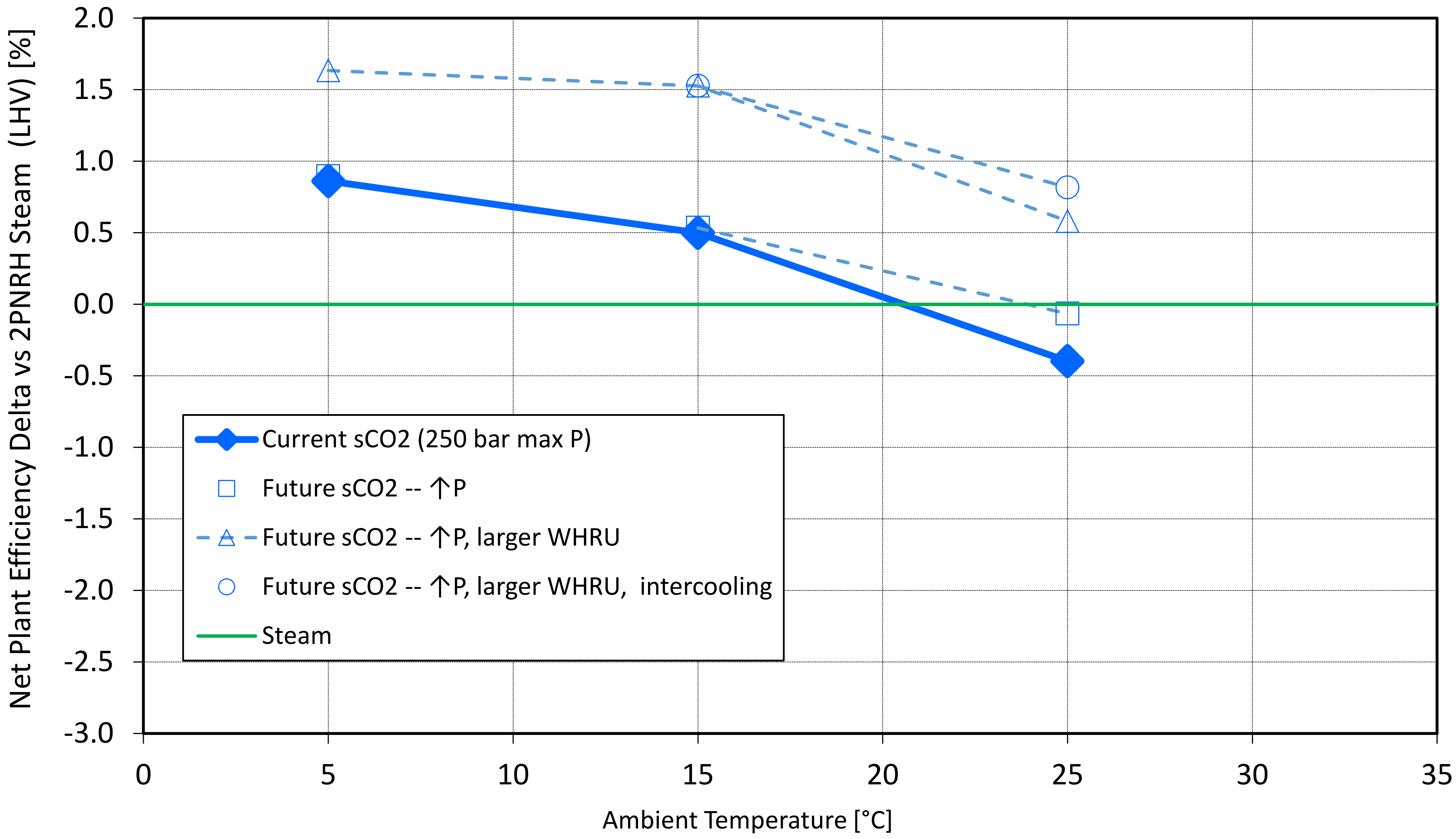
Impact of Ambient Temperature



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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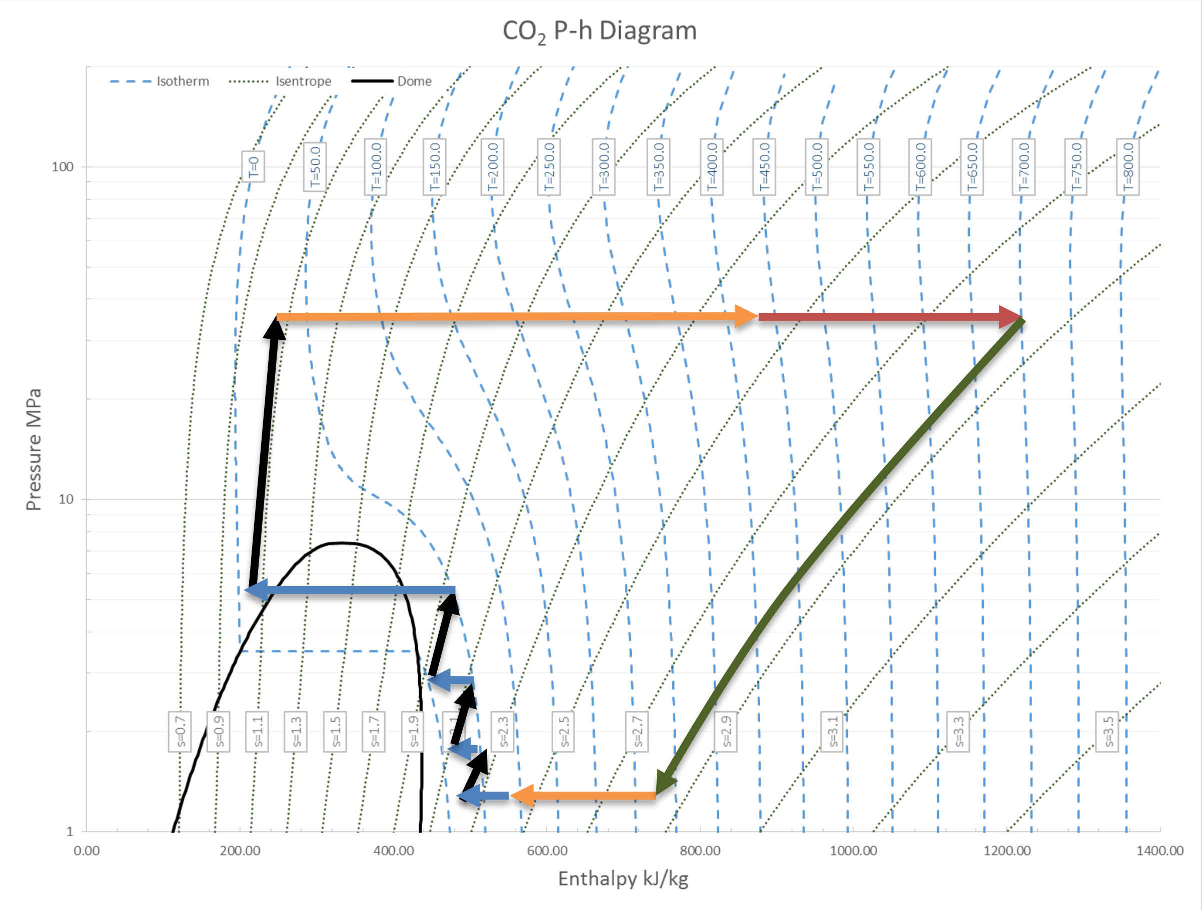
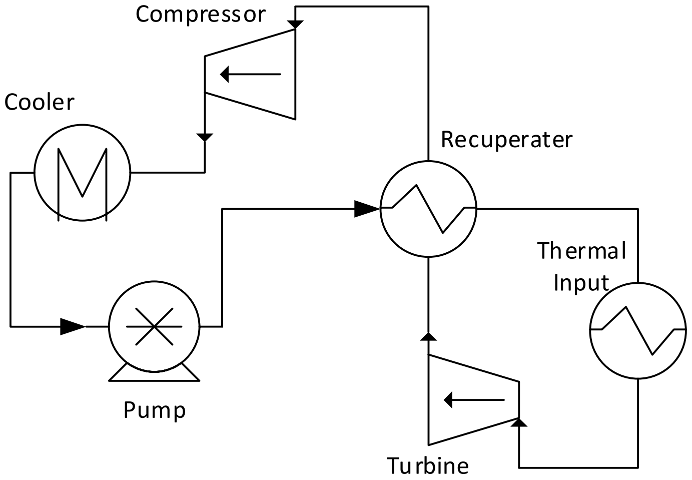
Condensing Cycles

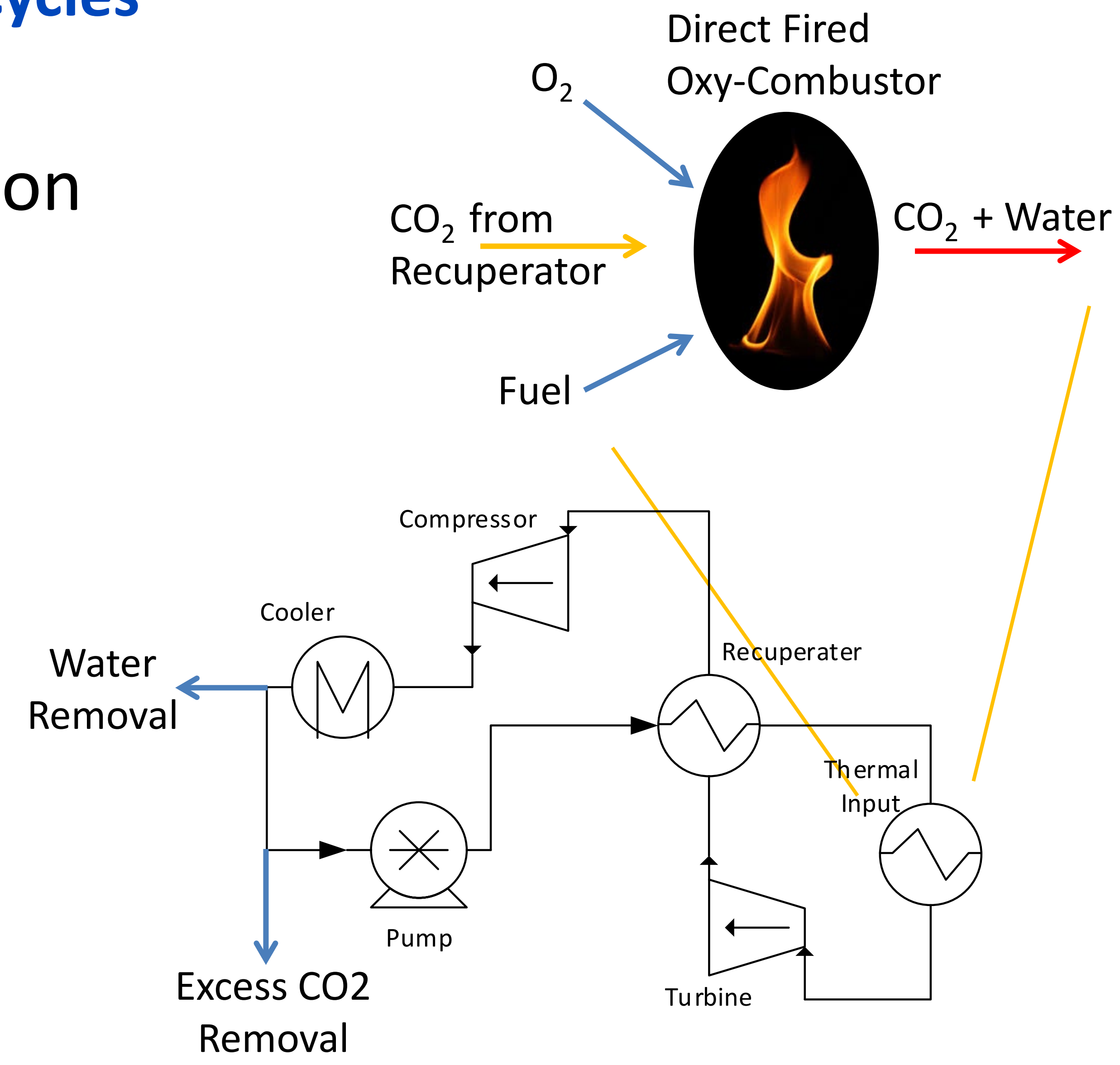


- Transcritical cycle
- Utilizes partial compression, condensation, and pumping to minimize compression work
- Must balance refrigeration / cooling requirements against compression requirements
- Can achieve efficiencies close to recompression cycle



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