Performance comparison of supercritical CO2 versus steam bottoming cycles for gas turbine combined cycle applications

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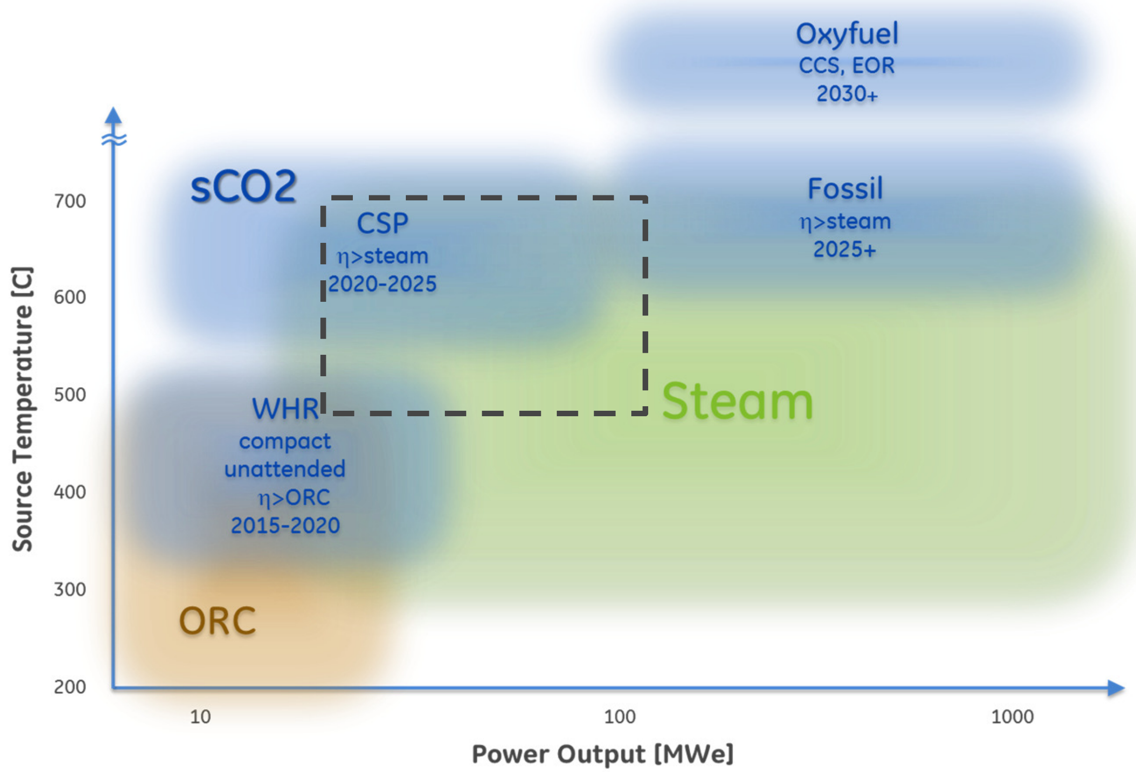
Imagination at work.

Focus on gas turbine bottoming cycle

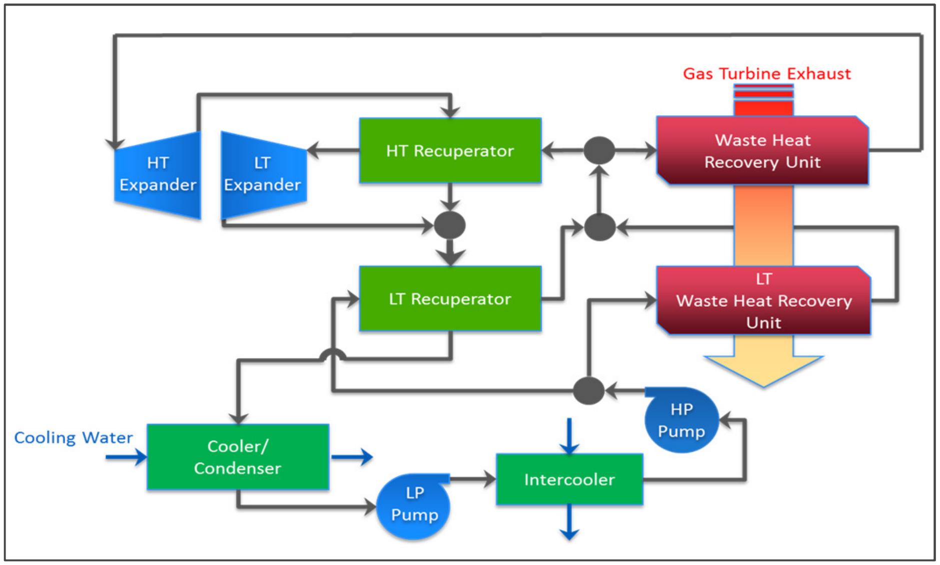


Courtesy of Douglas Hofer, GE Global Research

When is sCO2 bottoming cycle performance attractive vs. steam?

- GT size/type…aeroderivative vs. heavy-duty
- GT exhaust temperature…500°C to 700°C+
- Steam bottoming cycle type…2PNR and 3PRH

Focus on cycle with dual split and expansion



- Best compromise between waste heat utilization and 1st law efficiency
  → maximized power output and CC efficiency

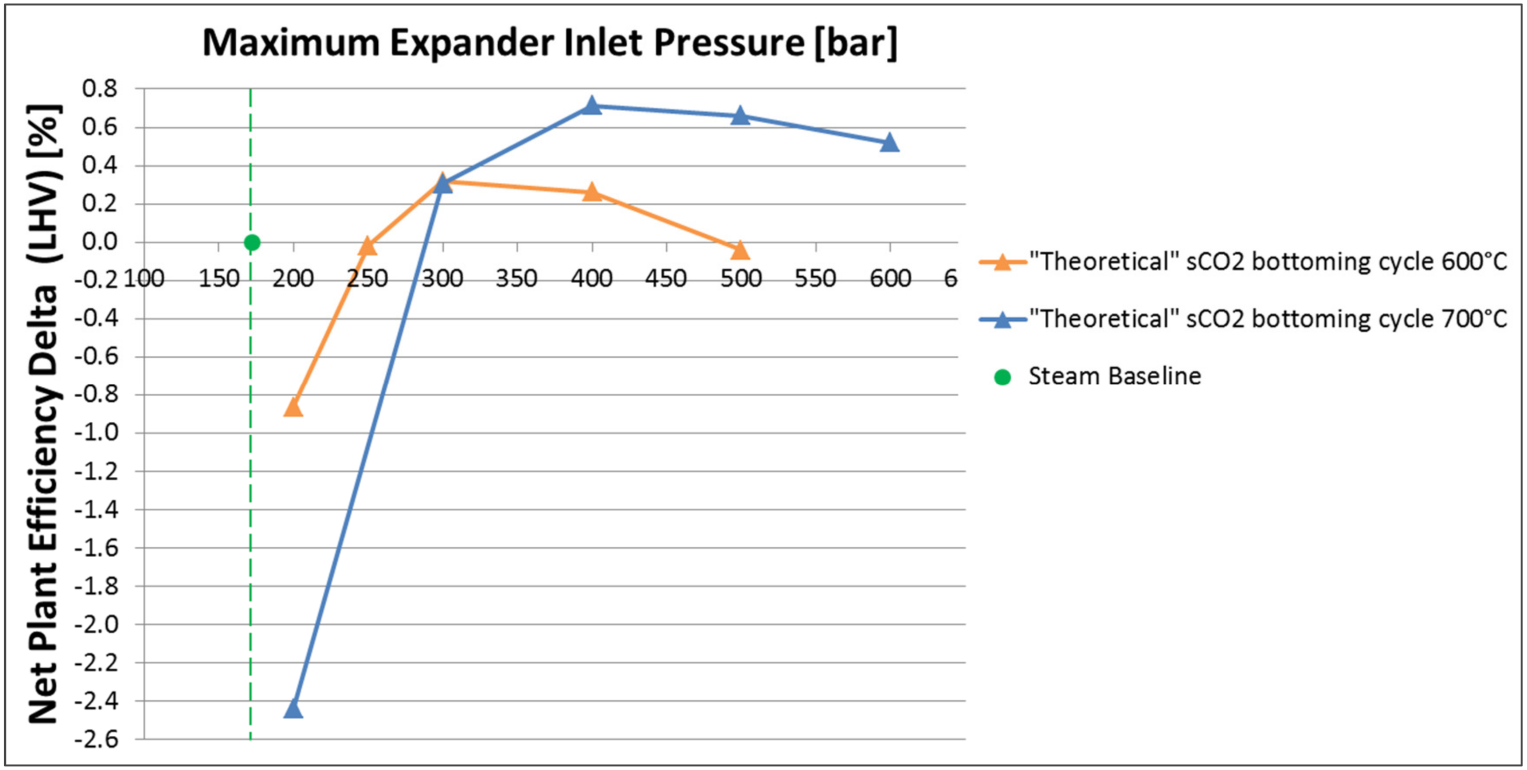
Reference	GT exhaust T	sCO2 vs. steam
Kimzey (1)	625°C	-13% bottoming cycle power
Cho et al. (2)	580°C	+0.7%pts CC efficiency (58.4% steam)
Kimzey (1)	471°C	+9% bottoming cycle power

Assumptions in the heavy-duty GT case

GT Type	[-]	H-Class	
Configuration	[-]	2x1, 3PRH	
Case	[-]	**Current** exhaust temperature range	**Theoretical higher** exhaust temperature range
Exhaust temperature	[°C]	650-700	700-750
Steam maximum temperature	[°C]	600	700
CC net efficiency	[%]	62-62.5	>62.5

- Key assumptions for sCO2 “theoretical” case:
  - **Very high expander and pump isentropic efficiencies: 95%**
  - Rankine cycle: condensed state at the coldest point
  - Intercooled pumps
  - 4°C hot and cold end approach on recuperators (high effectiveness)
  - Split ratio and intercooling pressure optimized to maximize power
  - Same UA as steam for waste heat recovery unit and condenser

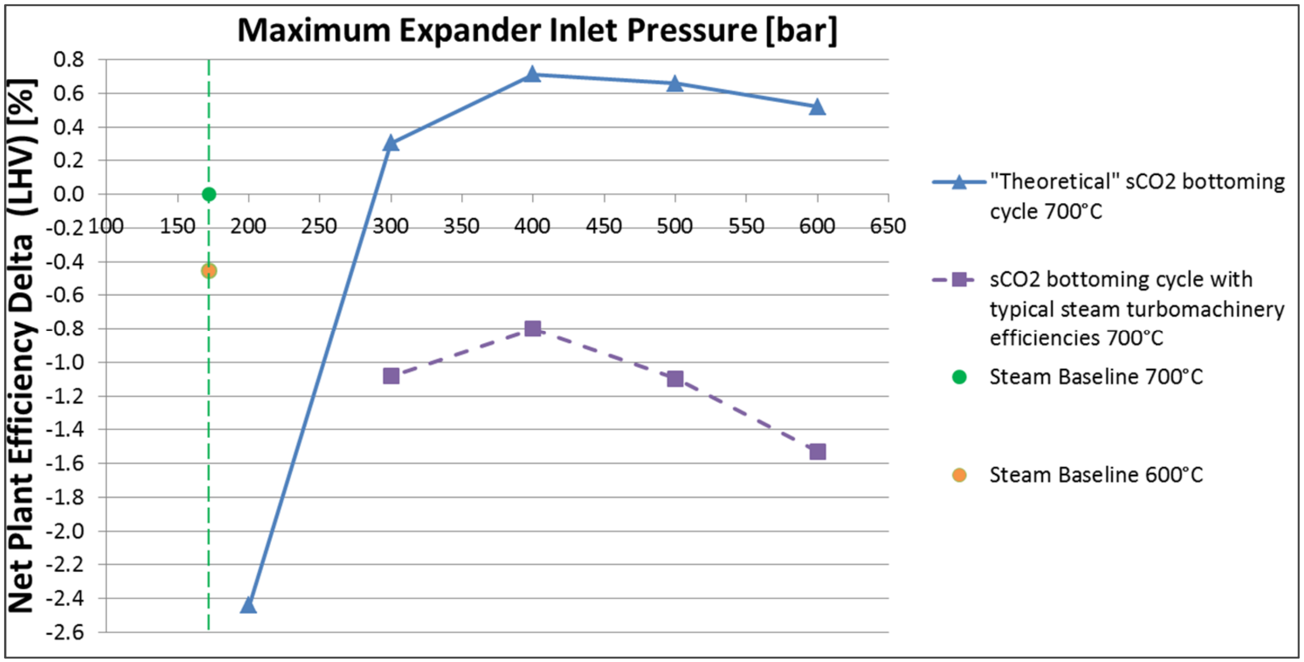
Performance comparison in the HDGT case



- High pressure level needed for sCO_2 to compete with steam
- Reference: max. 350bar in AUSC coal power plants research programs

Beyond the „theoretical“ case

- **sCO2 turbine and pump isentropic efficiencies reduced to usual level found in steam bottoming cycle**
  - Expander overall eff.: 91%→89%
  - Pump overall eff.: 93%→77%



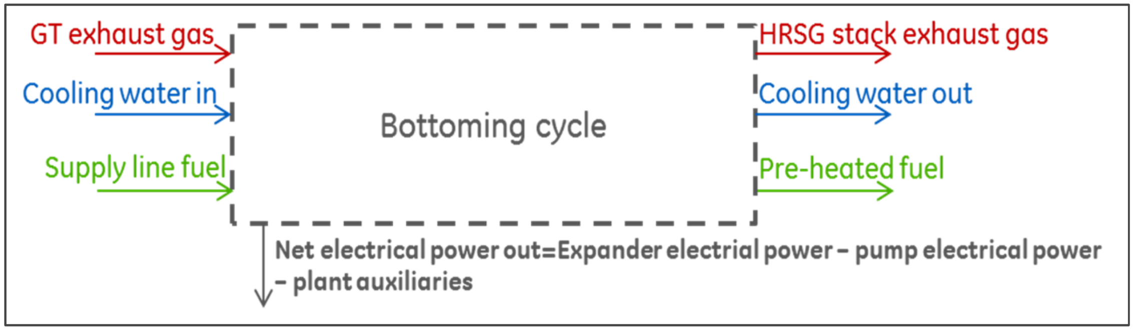
- 700°C CC efficiency of sCO2 at the optimum pressure is 0.8%pts lower than 700°C steam
- 600°C steam shows 0.4%pts higher net CC efficiency than 700°C sCO2

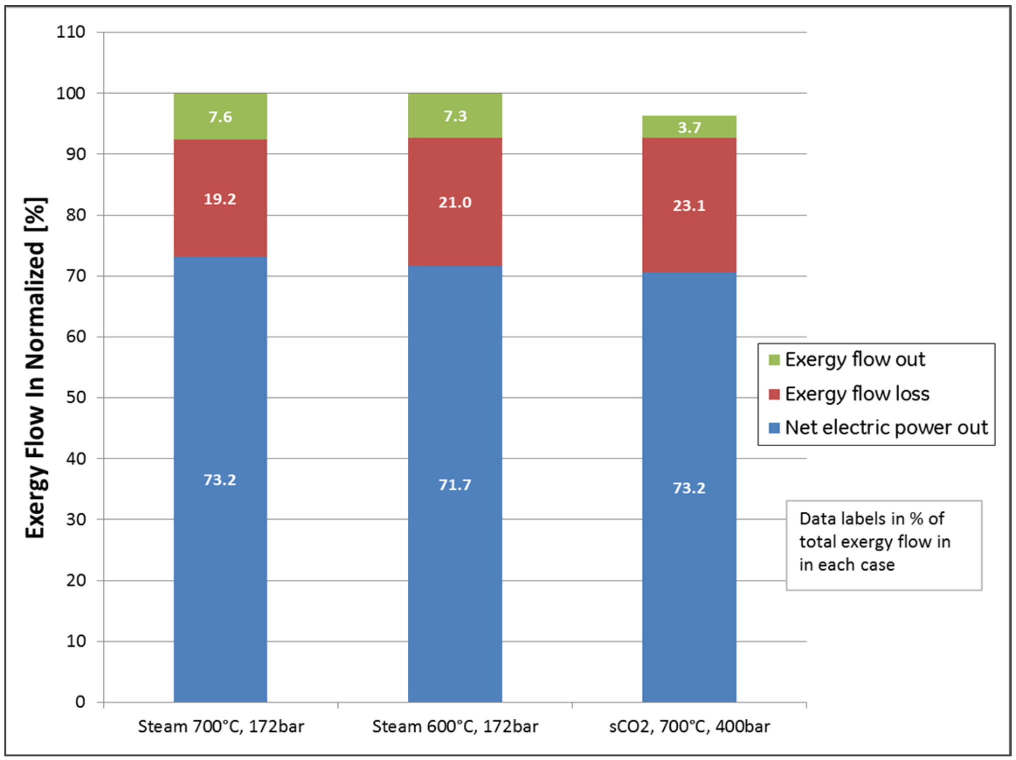
Detailed performance comparison

		Steam, 700°C, 172bar	Steam, 600°C, 172bar	sCO2, 700°C, 400bar
Net CC efficiency delta	[% pts]	0.00	-0.45	-0.80
WHRU thermal duty	[%]	100.0	101.5	96.9
Difference to reference stack T	[°C]	0	-11	+29
Bottoming cycle first law efficiency delta	[% pts]	0.00	-1.50	-0.26
Net electrical power	[%]	100.0	98.0	96.4
Expanders electrical power	[%]	100.0	98.3	123.6
Pumps electrical power	[%]	100	113	1632
HT expander inlet volume flow	[%]	100.0	98.9	118.5
Condenser inlet volume flow	[%]	100.0	107.5	0.2

- Poorer waste heat utilization in sCO2 case
  - Lower WHRU thermal duty
  - Higher stack temperature
- sCO2 1st law efficiency higher than steam at 600°C, lower at 700°C
  - Not high enough to compensate for lower utilization
- sCO2 larger gross power and lower condenser inlet volume flow

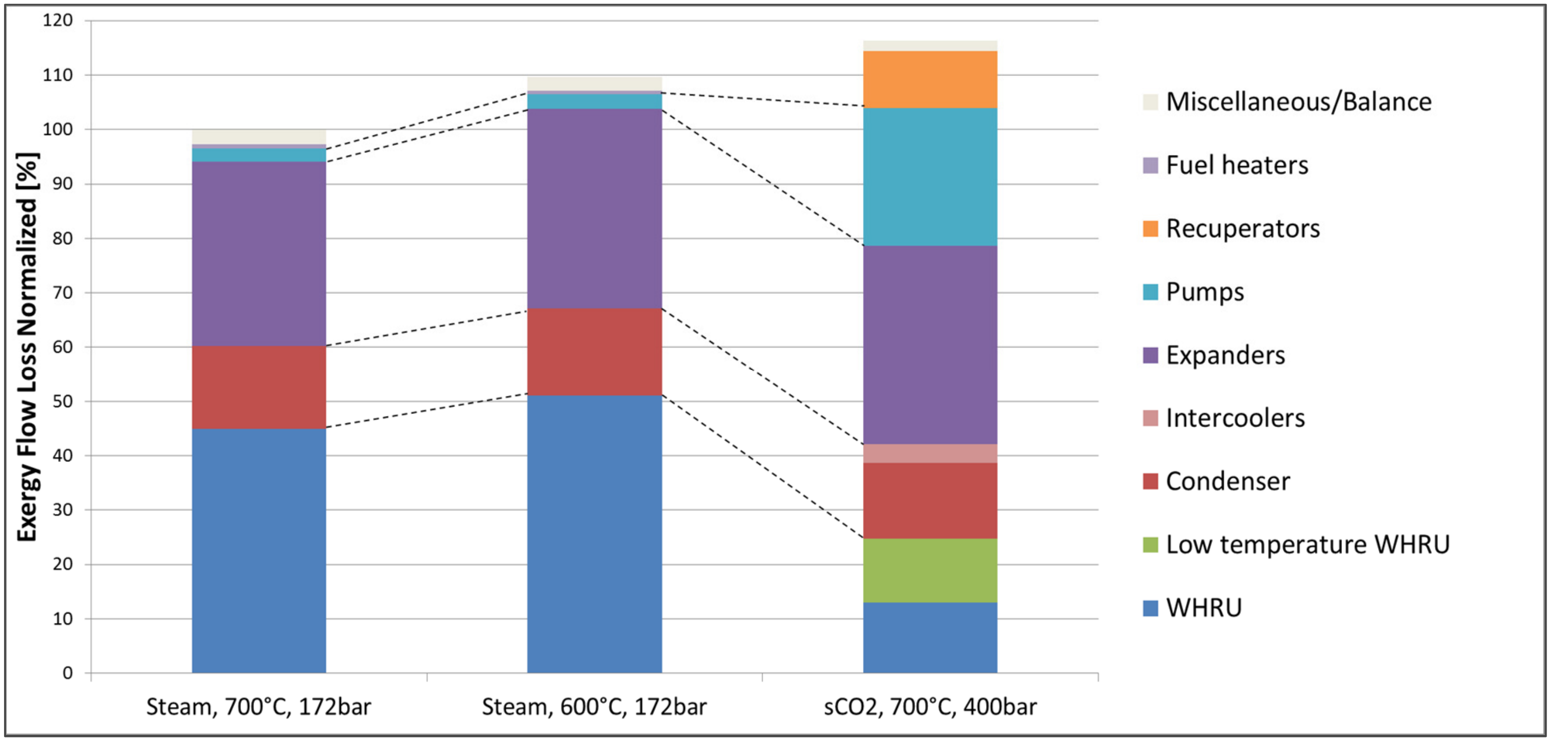
Exergy analysis





- Total incoming exergy flow lower in sCO2 case (fuel heating not considered for CO2)
- Net electric power output lower in sCO2 case due to higher exergy flow losses
- Exergy flow out lower for sCO2 because no heated fuel leaves the cycle unlike in the steam case
    - If fuel heating neglected in the steam cases, sCO2 has higher exergy outflow because of higher stack temperature

Exergy flow losses analysis



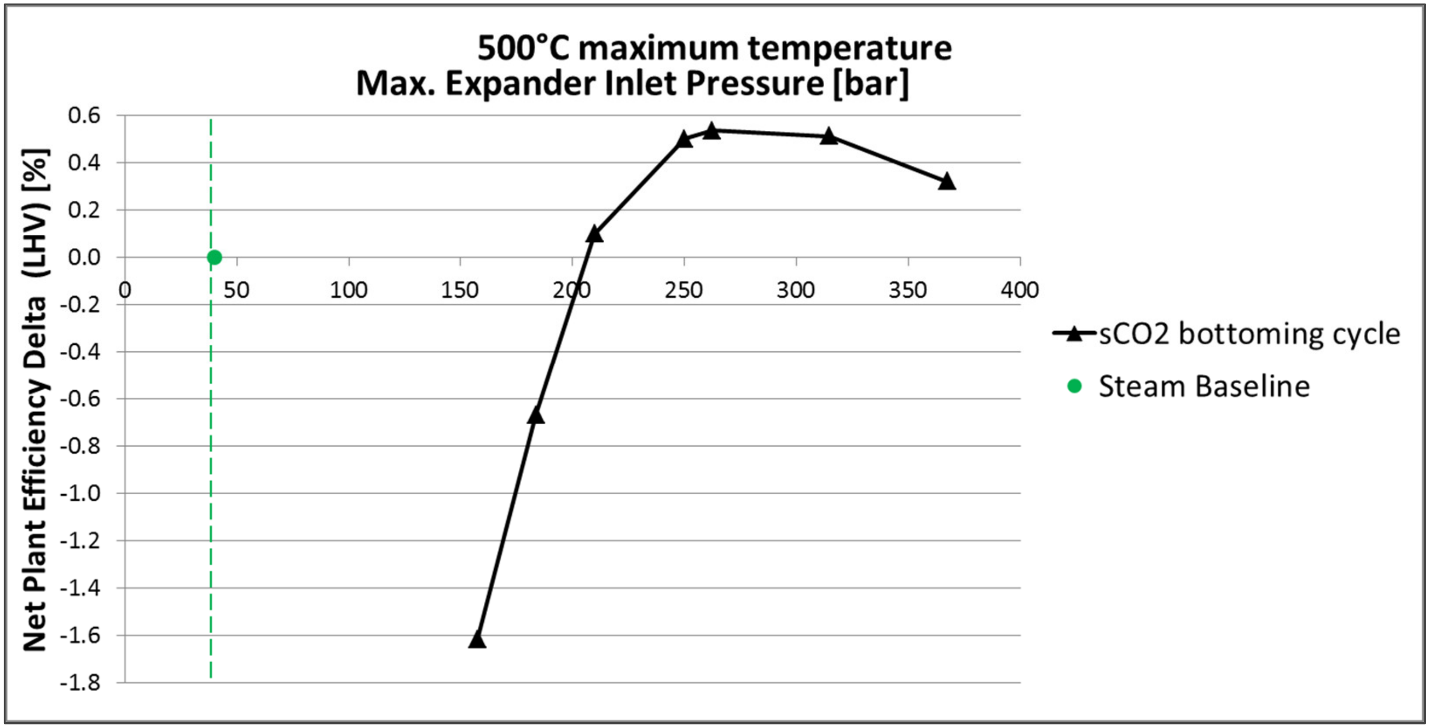
- WHRU exergy flow losses lower in sCO2 case due to a better temperature match during heat exchange
- Condensers losses similar in 3 cases
- Losses during expansion only slightly higher in sCO2 cases because of larger gross power
- Total exergy flow losses accounted above lower in sCO2 case
- This changes when considering pump exergy losses, much higher in sCO2 case
- Losses of sCO2 compared to steam further increased by recuperators exergy losses

Comparison in the aeroderivative GT case

GT Type	[-]	LM2500
Configuration	[-]	1x1, 2PNR
Exhaust temperature	[°C]	525-550
Steam maximum temperature	[°C]	500
CC net efficiency	[%]	52.5-53

- Key assumptions for sCO2:
  - **Reasonable expander and pump isentropic efficiencies at this scale (85% to 90% and 75% to 80% respectively)**
  - Rankine cycle: condensed state at the coldest point (air-cooled)
  - Non-intercooled pumps
  - 4°C hot and cold end approach on recuperators (high effectiveness)
  - Split ratio and intercooling pressure optimized to maximize power
  - Same UA as steam for waste heat recovery unit and condenser

Performance comparison in the Aero case



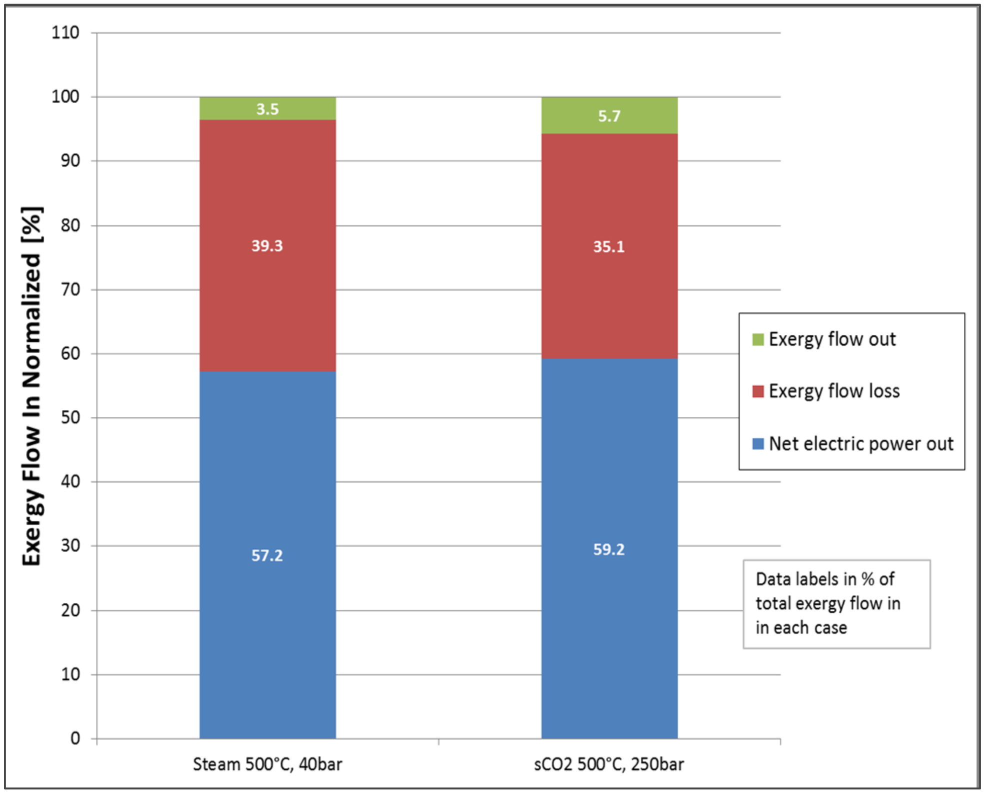
- sCO_2 outperforms steam when pressure higher than 200bar
- Optimum of 0.5%pts CC net efficiency gain over steam baseline reached at 250bar
- Despite more near-term design with less optimistic boundary conditions than for heavy-duty GTs, sCO_2 cycle shows superior performance to a steam bottoming cycle for aeroderivative gas turbines

Detailed performance comparison

		Steam, 500°C, 40bar	sCO2, 500°C, 250bar
Net CC efficiency delta	[% pts]	0.00	0.50
WHRU thermal duty	[%]	100.0	96.5
Difference to reference stack T	[°C]	0	+23
Bottoming cycle 1st law efficiency delta	[% pts]	0.00	+2.07
Net electrical power	[%]	100.0	103.5
Expanders electrical power	[%]	100	135
Pumps electrical power	[%]	100	3884
HT expander inlet volume flow	[%]	100.0	49.6
Condenser inlet volume flow	[%]	100.0	0.2

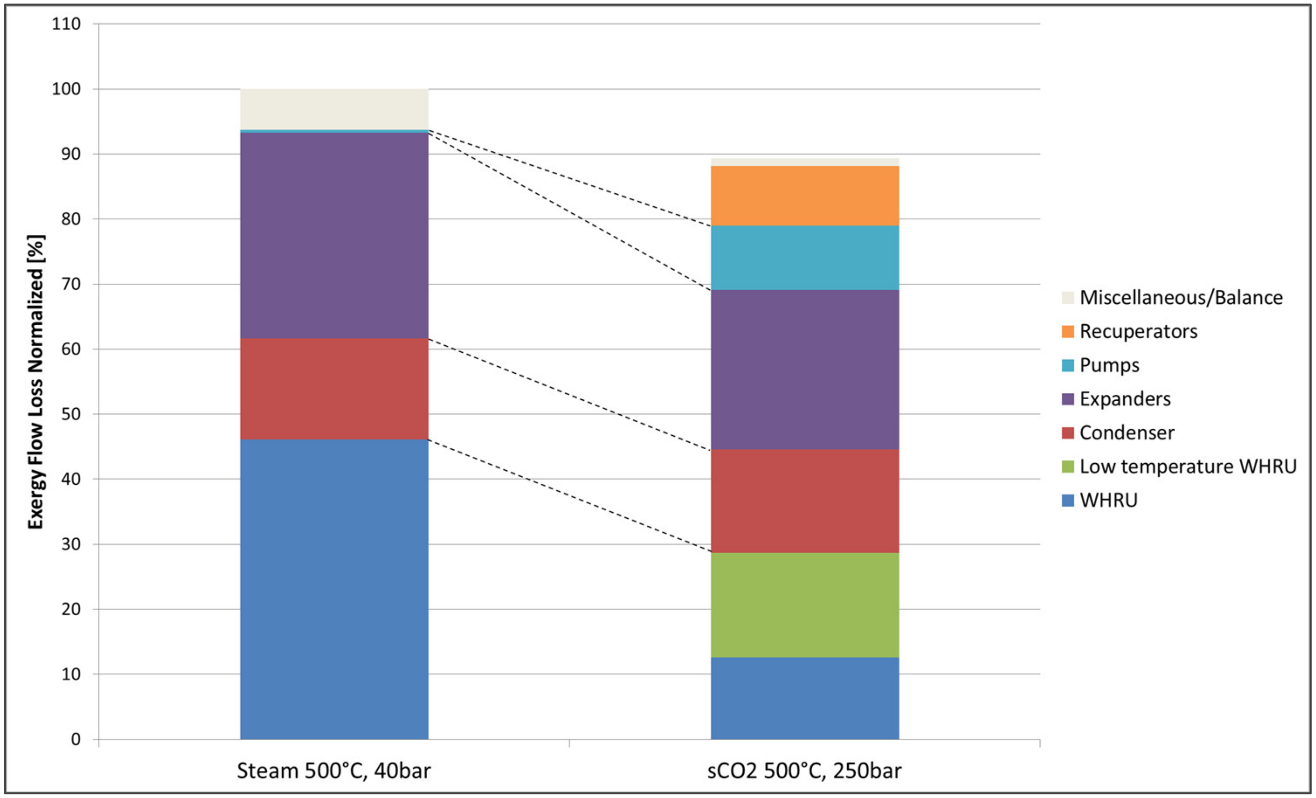
- Poorer waste heat utilization in sCO2 case
  - Higher stack temperature
  - Lower WHRU thermal duty
- sCO2 1st law efficiency higher than steam
  - Higher enough to compensate for lower utilization and result in higher CC efficiency
  - Key difference with heavy-duty GT
- sCO2 larger gross power and lower condenser inlet flow

Exergy analysis



- Same total incoming exergy flow in both cases
  - No fuel heating considered
- Net electric power output higher in sCO_2 case
  - Due to lower exergy flow losses
  - Despite higher stack temperature losses

Exergy flow losses analysis



- WHRU exergy flow losses lower in sCO_2 case
- Condensers and expanders losses similar in both cases
- **Unlike in heavy-duty GT case, reduction in WHRU exergy flow losses large enough to make up for the higher pumping and additional recuperator losses**
- Result in overall lower exergy flow losses in sCO_2 case

Conclusion

- sCO2 bottoming cycle considered in this paper will likely
  - achieve higher performance than 2PNR steam bottoming cycles typically paired with small aeroderivative gas turbines with reasonable pressure and turbomachinery efficiency levels
  - need very high component efficiencies and operating pressures to achieve higher performance than 3PRH steam bottoming cycles typically paired with large heavy-duty gas turbine
- Other factors need to be compared between sCO2 and steam bottoming cycles:
  - cost
  - footprint
  - operability
  - maintenance

References

(1) Kimzey G., Development of a Brayton Bottoming Cycle using Supercritical Carbon Dioxide as the Working Fluid, Gas Turbine Industrial Fellowship, University Turbine Systems Research Program, 2012

(2) Cho S.K., Kim M., Baik S., Ahn Y., Lee J.I., Investigation of the Bottoming Cycle for High Efficiency Combined Cycle Gas Turbine System with Supercritical Carbon Dioxide Power cycle, GT2015-43077, ASME Turbo Expo 2015, 2015