Paper 90:

A Conceptual Evaluation of a kW-Scale sCO_2 Power Cycle for Waste Heat Recovery on a Heavy-Duty Diesel Engine Truck

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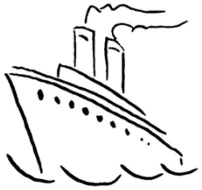
sCO_2 at the kilowatt (kW) scale

This part of the power generation market has not seen the same level of component and technology development as larger scale systems

State-of-the-art (SOA) technologies for kW-scale power generation include

- Organic Rankine Cycles (ORC) for WHR applications
- Conventional gasoline and diesel generators in the small/modular power generation market
- Stirling engines used for specialized Department of Defense applications



Mobile WHR applications for diesel IC engines include several industries:

- Long-haul trucks
- Locomotive
- Marine

For these markets, **power density**, **cost**, and **emissions** are key drivers

sCO2 cycles offer potential advantages with

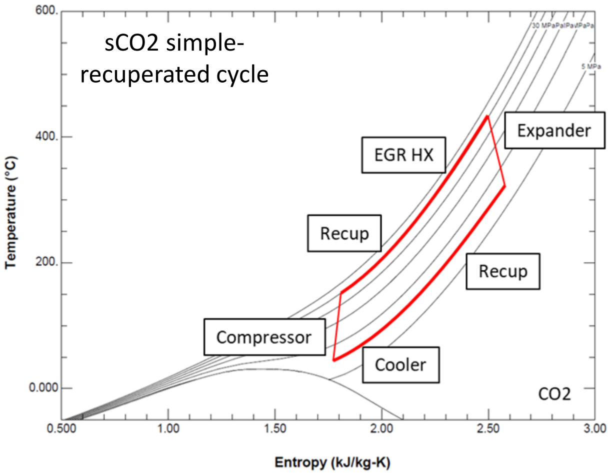
- higher system efficiencies enabling greater power output from the same waste stream
- higher fluid densities and favorable heat transfer characteristics enabling compact system components

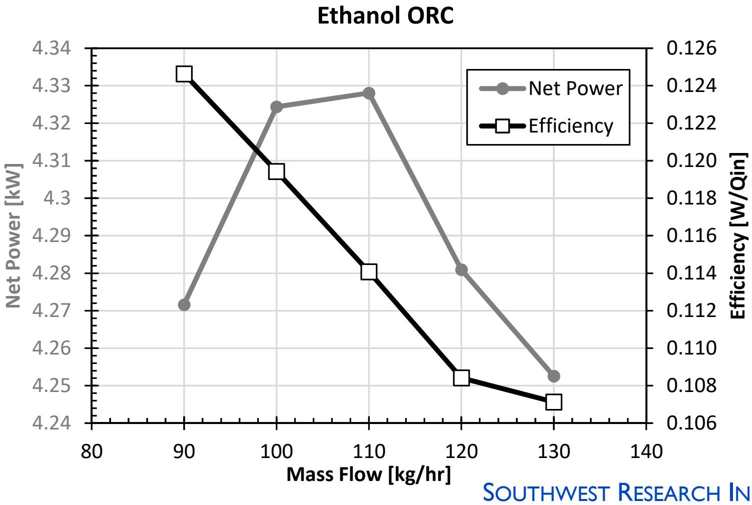
Comparison with the SOA

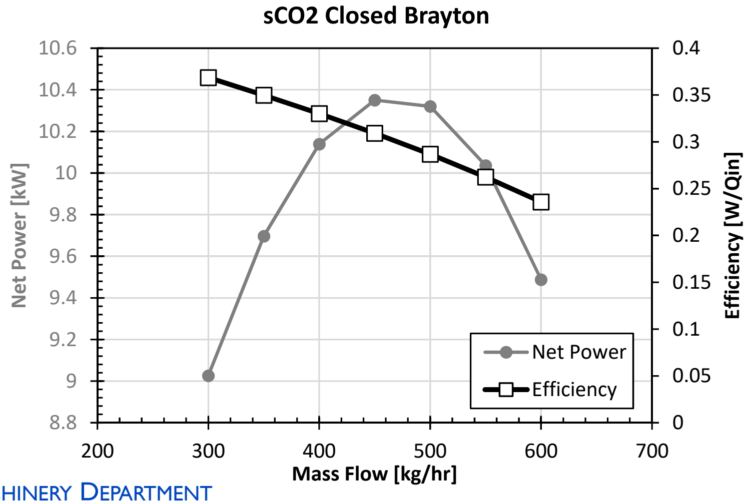
Waste Heat Source: 2027-era heavy duty truck engine Exhaust Gas Recycle (EGR)

Engine Speed = 1360 RPM
Torque = 1840 Nm
Power = 262 kW
EGR rate = 23%
EGR flow rate = 300 kg/hr
EGR Temp = 598 °C

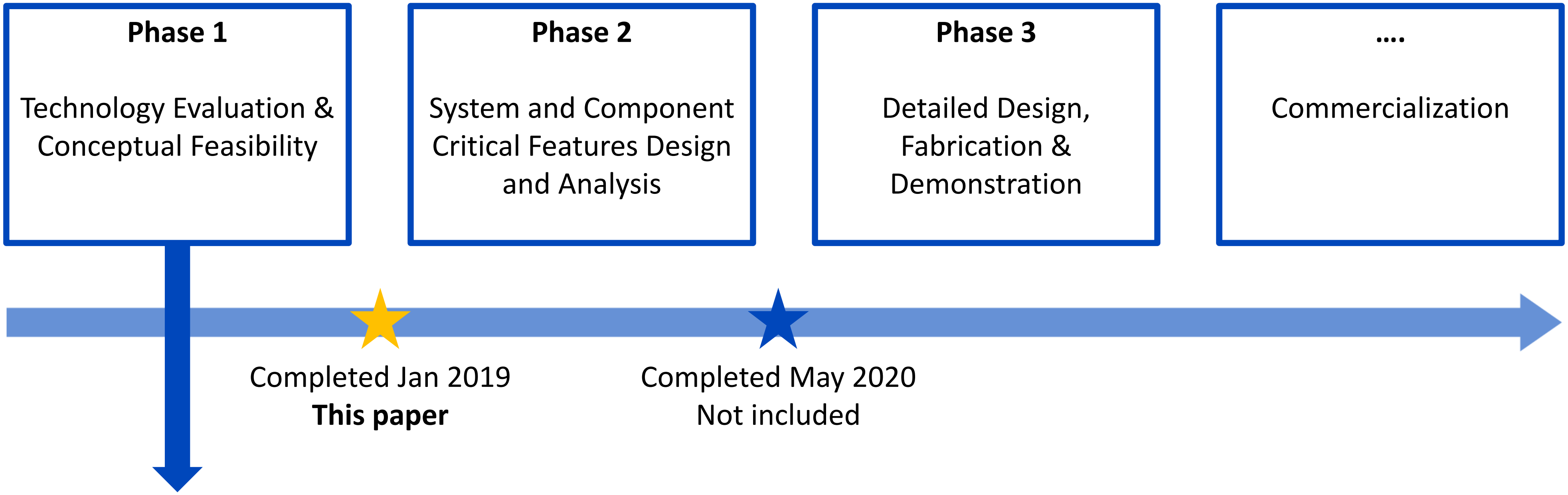
*The **sCO$_2$ system generates over twice the power** than the equivalent ORC system from the same waste heat source (and turbine inlet temperature)*







This kW-scale sCO2 WHR system development path



Phase 1

Technology Evaluation & Conceptual Feasibility

Phase 2

System and Component Critical Features Design and Analysis

Phase 3

Detailed Design, Fabrication & Demonstration

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Commercialization

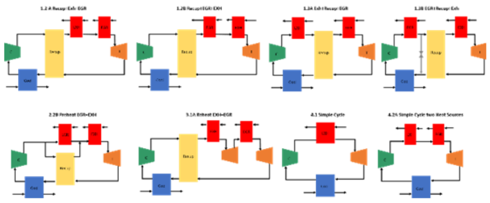
Completed Jan 2019
This paper

Completed May 2020
Not included

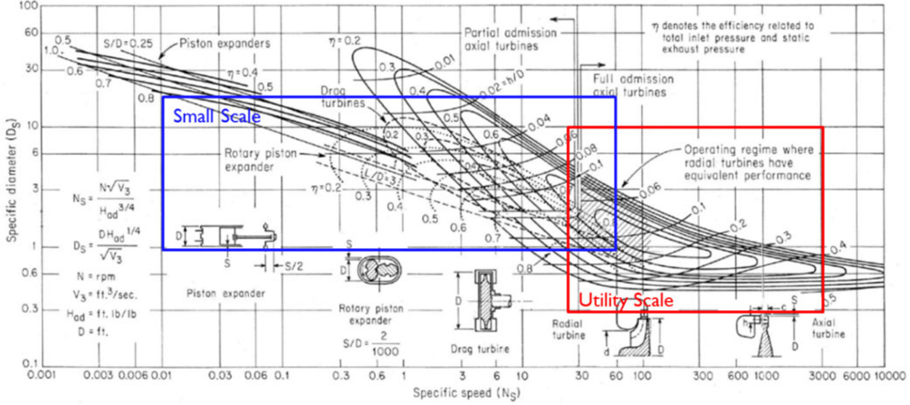
The overall objective of the project is the proof-of-concept of a scalable and efficient kW-scale sCO$_2$-based power generation system for waste heat recovery.

The Technology Evaluation & Conceptual Feasibility included the following major tasks

Cycle Architecture & Optimization

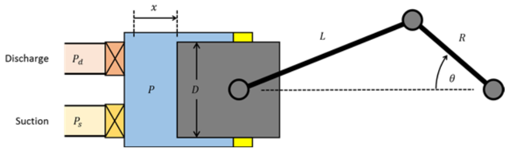


Machinery Selection



Technology Gap Assessment

Component Conceptual Design Feasibility



CYCLE ANALYSIS

WHR Application and Heat Source Definition

Heavy duty truck WHR application was chosen due to

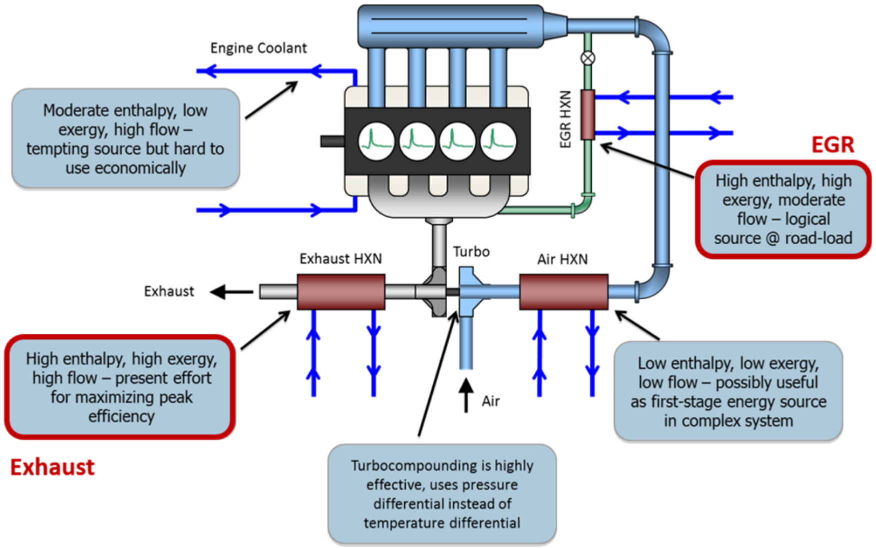
- Interest from the market
- Additional challenges for the mobile application

The potential engine heat sources were consider:

- **Exhaust stream**
- Turbocompounding stream
- Air HX
- **EGR HX**
- Engine Coolant

Design point waste heat stream conditions:

- SwRI test data from a representative current production engine
- Mode 6 (A75) from a standard 13-mode test



Moderate enthalpy, low exergy, high flow – tempting source but hard to use economically

High enthalpy, high exergy, moderate flow – logical source @ road-load

High enthalpy, high exergy, high flow – present effort for maximizing peak efficiency

Turbocompounding is highly effective, uses pressure differential instead of temperature differential

Low enthalpy, low exergy, low flow – possibly useful as first-stage energy source in complex system

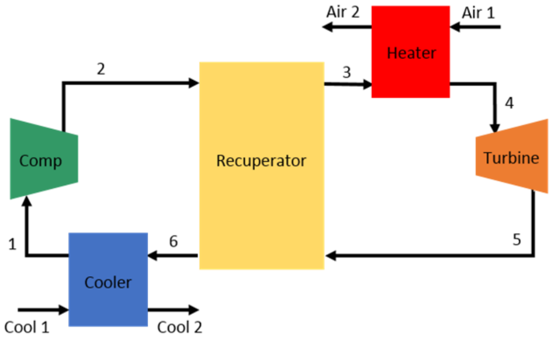
		Exhaust	EGR
Temperature	*°C*	376.4	506.2
Flow Rate	*kg/s*	0.276	0.064

Cycle Analysis & Baseline Cycle

npss
NUMERICAL PROPULSION SYSTEM SIMULATION
A Consortium of SwRI

Numerical Propulsion System Simulation (NPSS)

- Object-oriented, multi-physics, engineering design and simulation environment
- Commonly used for system analysis in the application areas of
    - Cycle design,
    - Steady-state and transient off-design performance prediction,
    - Test data matching for aerospace systems (i.e. engine performance models for aircraft propulsion),
    - Thermodynamic system analysis such as Rankine and Brayton cycles, and rocket propulsion cycles.
- Flow-network solver able to model a wide range of fluid-thermal problems extending past conventional Brayton cycles such as multi-phase heat transfer systems, refrigeration cycles, and closed power cycles
- Fluid Properties interfaced with NIST REFPROP



A **simple recuperated cycle** was used as the **baseline cycle** with

- Assumed component efficiencies
- Three high-side pressures: 250, 275, and 300 bar
- Heat rejection with coolant and ambient air, fixed
- Sweep of CO_2 flow rates

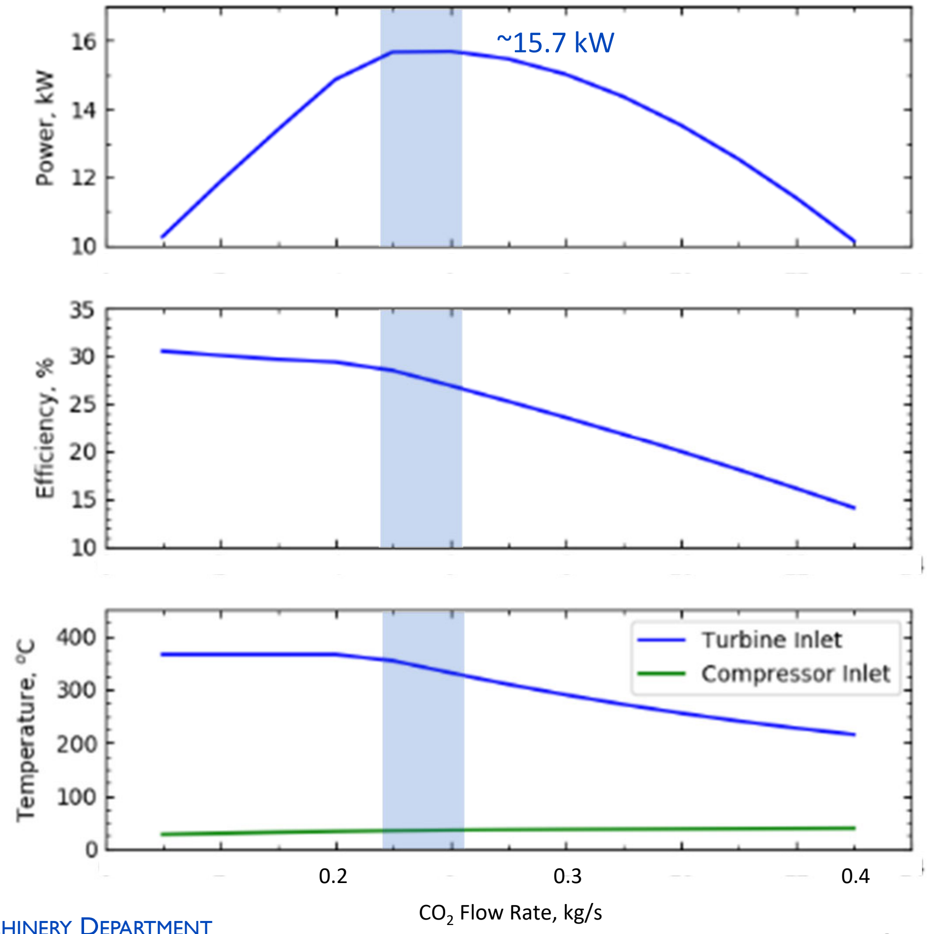
Compressor Efficiency	82%
Turbine Efficiency	89%
Heat Exchanger Effectiveness	95%
Pressure Loss	5%

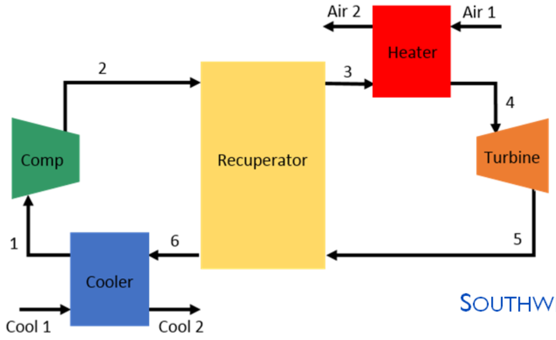
Simple recuperated cycle with a single heat source

- Net power output is the primary metric of interest
- Efficiency is informative about the utilization of heat input, but less useful as an end target

- The peak power cases (0.225-0.25 kg/s) corresponded with peak recuperator duty

- These conditions were used as a baseline for initial component scaling, off-design considerations, and for comparison with other cycle architectures

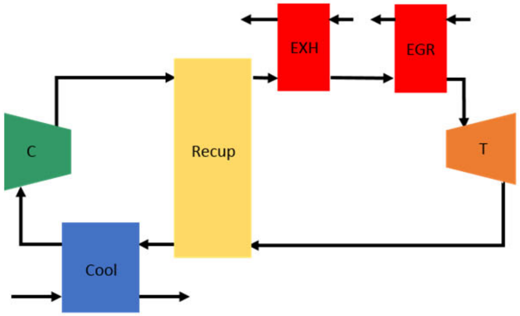




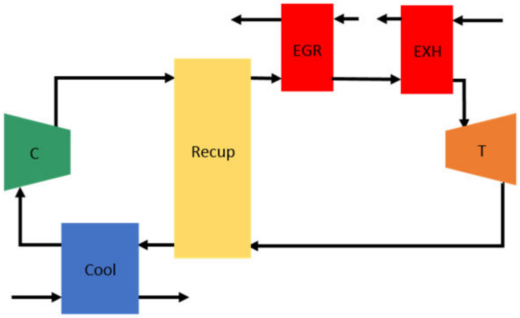
Cycle Architecture Evaluation

How can both the exhaust and EGR flows be used as heat sources? Can this be leveraged while still keeping complexity and size low?

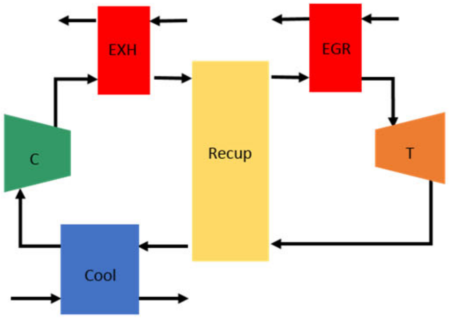
1.2 A Recup+Exh+EGR



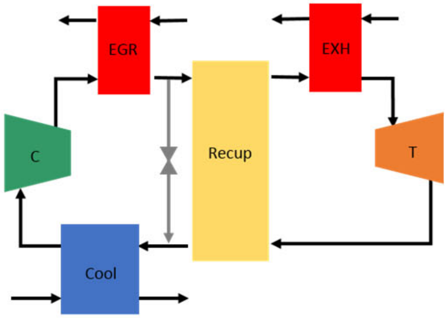
1.2B Recup+EGR+EXH



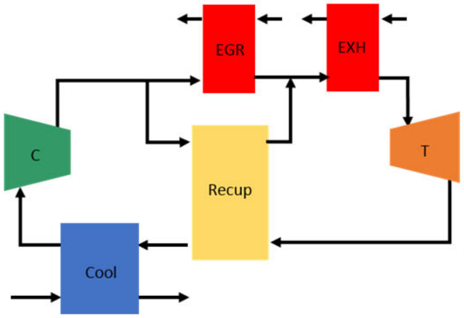
1.3A Exh+Recup+EGR



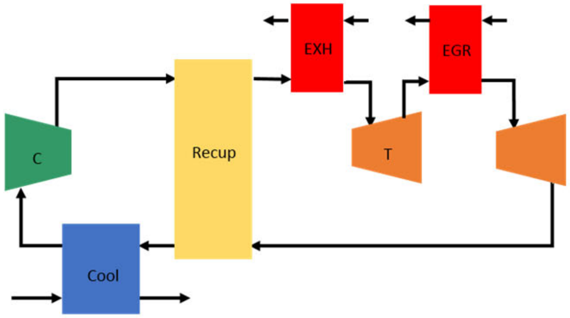
1.3B EGR+Recup+Exh



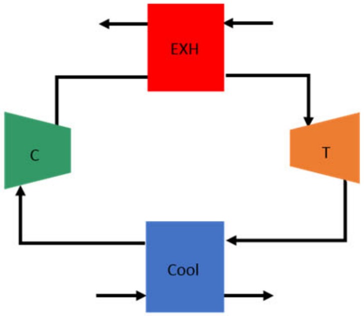
2.2B Preheat EGR+EXH



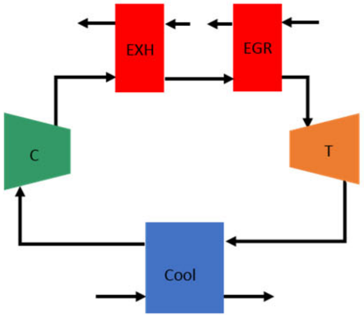
3.1A Reheat EXH+EGR



4.1 Simple Cycle



4.2A Simple Cycle two Heat Sources



Sweeps similar to those for the baseline cycle were completed

Maximum net power output case results are summarized and compared

	Name	Efficiency	Net Power [kW]	Heat Input [kW]	Recup Duty [kW]	CO2 Flow [kg/s]	T low [°C]	EXH T out [°C]	EGR T out [°C]	No. of Major Hardware Items
1	**Simple Recuperated**									
1.1	Recup Exh	28.5%	15.7	55	48	0.2268	34.6	290	-	5
1.2A	Recup+Exh+EGR	30.5%	17.58	56	56	0.2268	34.41	210	380	6
		27.1%	18.5	70	53	0.2948	36.78	185	320	6
1.2B	Recup+EGR+Exh	29.5%	16.5	56	50	0.2268		255	200	6
		26.0%	17.5	67	47	0.2948	35	225	185	6
1.3A	Exh+Recup+EGR	10.0%	10.5	105	34	0.19	36.8	80	235	6
1.3B	EGR+Recup+Exh	21.8%	16.7	76	36	0.2268	28.8	215	75	6
		20.7%	17.4	85	36	0.272	31.1	185	80	6
2	**Preheat**									
2.2B	Preheat EGR+Exh	26.6%	17.9	68	6	0.295	36.8	270	105	6
		25.5%	17.9	70	7	0.315	39	265	110	6
3	**Reheat**									
3.1A	Reheat Exh+EGR	27.0%	18	68	60	0.2948	38.5	200	260	7
		28.5%	17.7	62	63	0.25	37	220	285	7
3.1B	Reheat EGR+Exh	25.0%	16.7	68	72	0.2948		345	235	7
4	**Simple Cycle**									
4.1	Simple Exh	15.0%	13	86	-	0.181	30.6	80	-	4
4.2A	Simple Exh+EGR	15.1%	15	100	-	0.2041	30.7	80	385	5
		16.6%	15.1	104	-	0.25	30.7	80	295	5

Comparison of Cycle Architectures

● **Simple Recuperated Cycle with Two Heat Sources**

- Included all combination orders of the recuperator, exhaust HX, and EGR HX
- 18.5 kW net power output
- Fully replace EGR HX

▲ **Preheat Cycle**

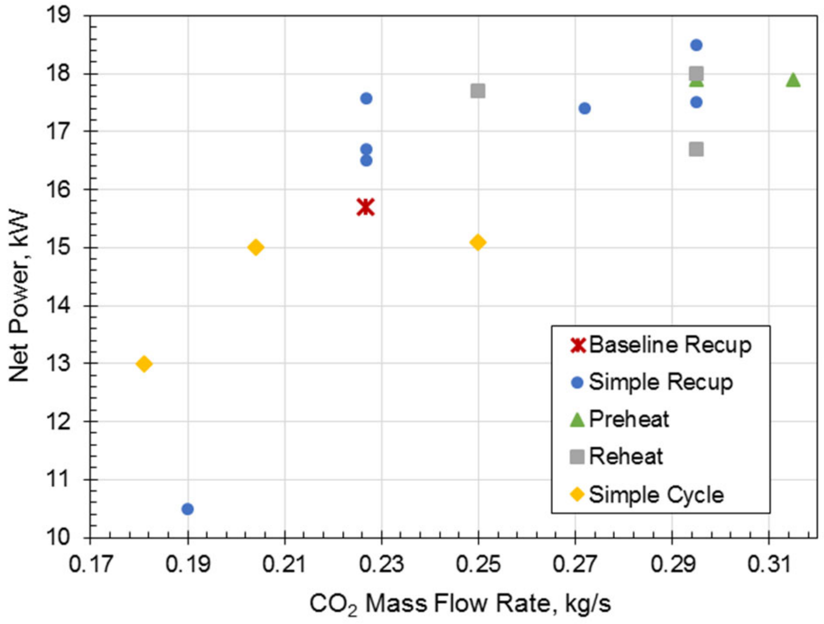
- EGR in parallel with the recuperator was the only configuration to converge
- Split fraction was also swept

■ **Reheat Cycle**

- Reasonable performance
- Added complexity

◆ **Simple Cycle**

- Simple cycle with one heat source produced 13 kW



Other factors:

- Recuperator duty
- EGR exit temperature
- Complexity

MACHINERY SELECTION AND FEASIBILITY

How difficult would turbomachines be?

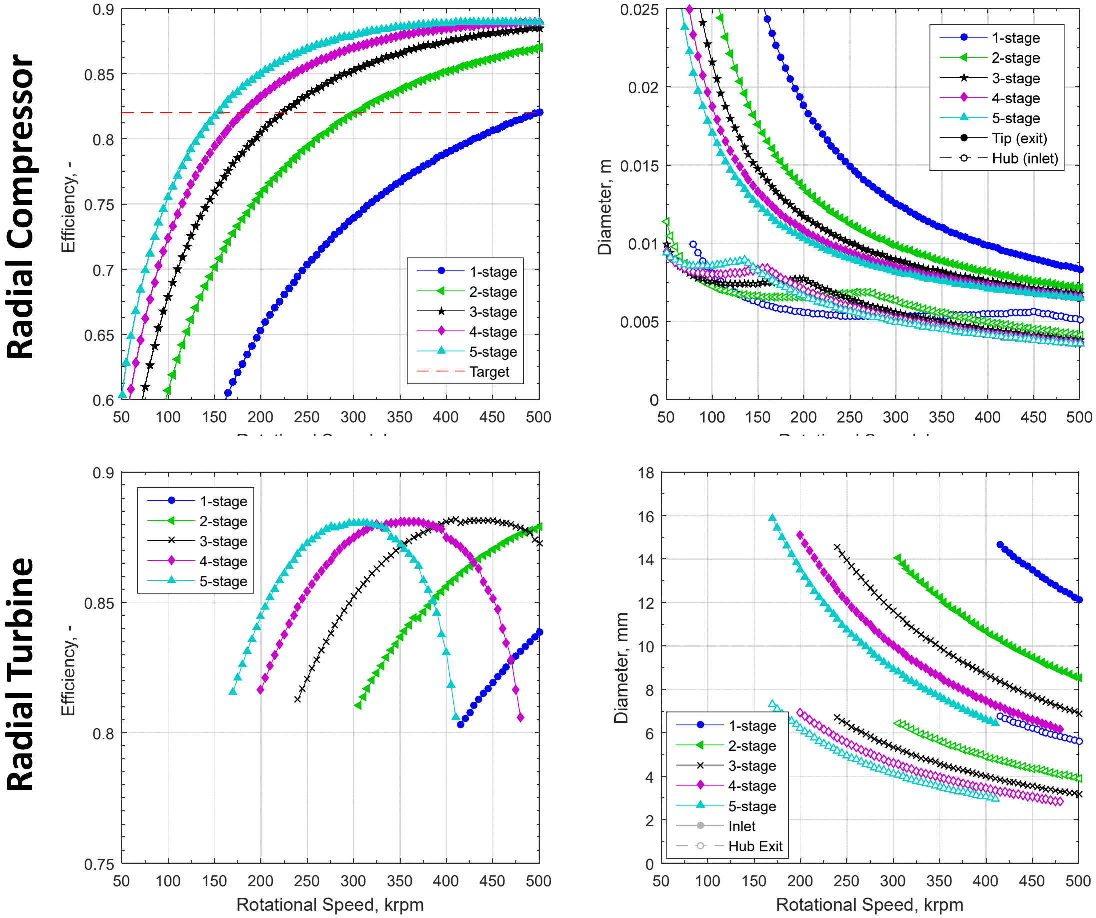
Conceptual design point sweeps here complete based on historical non-dimensional sizing (Balje, 1981 and Aungier, 2000)

- 1 to 5 stage machines
- 50,000 to 500,000 rpm

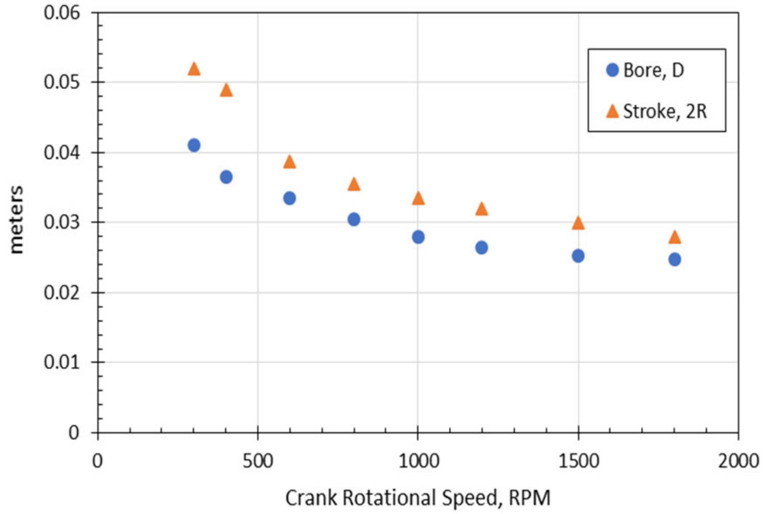
Result indicate

- High speeds beyond most sub-component technologies
- Small diameters
- High stage counts

Underscore the necessity for alternative machinery architectures for sCO2 power cycles at this scale



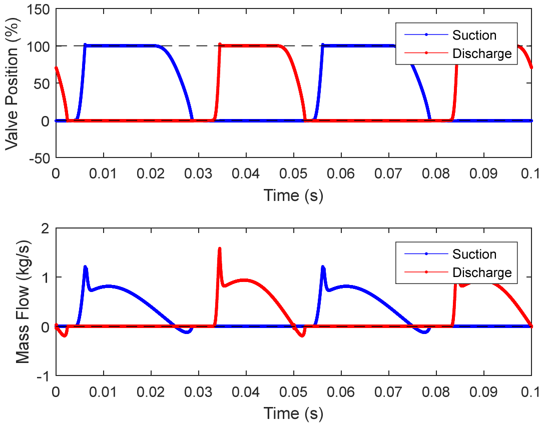
Reciprocating Compressor

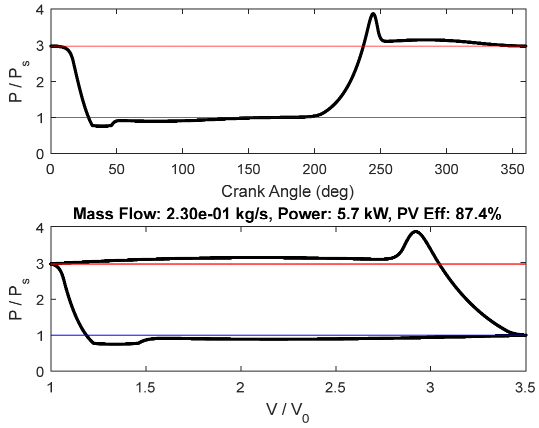


A single cylinder reciprocating compressor in-house SwRI model was used to estimate reciprocating compressor sizing

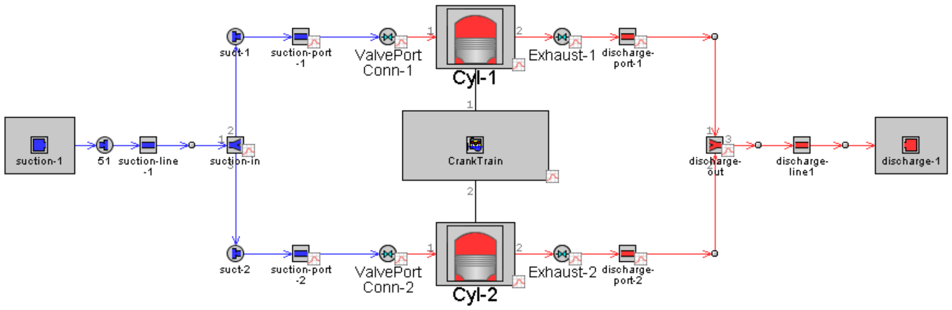
Sweeps of crank speeds were completed

The compressor is small but of the same magnitude as high pressure reciprocating compressors for other applications





Piston Expander



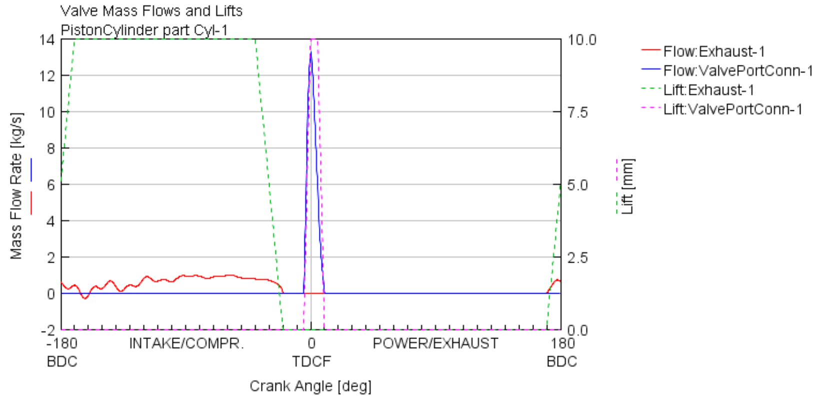
GT-Power, an engine simulation program, was used to build an initial model of a 2-piston expander with piston-ported valves

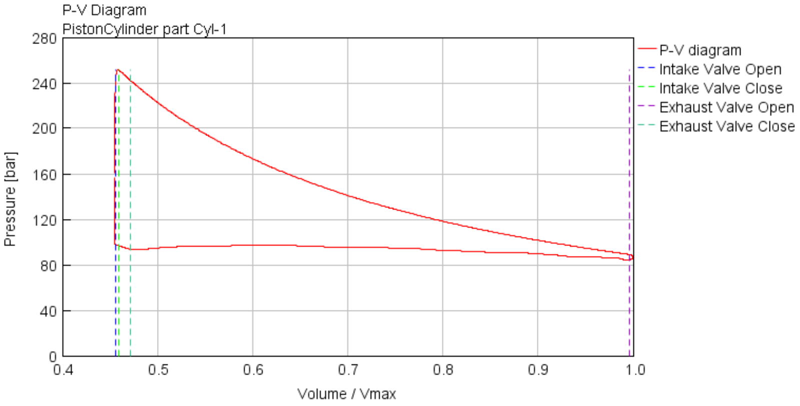
The modeled expander performance has a predicted output of approximately 30 kW at a shaft speeds ranging from 1000 to 5000 rpm

Volumetric efficiency is on the order of 39 percent. The relatively low volumetric efficiency is explained by the fact that the expander completes its cycle every revolution (two piston strokes) resulting in a very short intake process while the piston is near TDC

Example breathing characteristics and PV performance shown

- Sweeps of valve timing were completed
- Performance was particularly sensitive to the exhaust valve timing





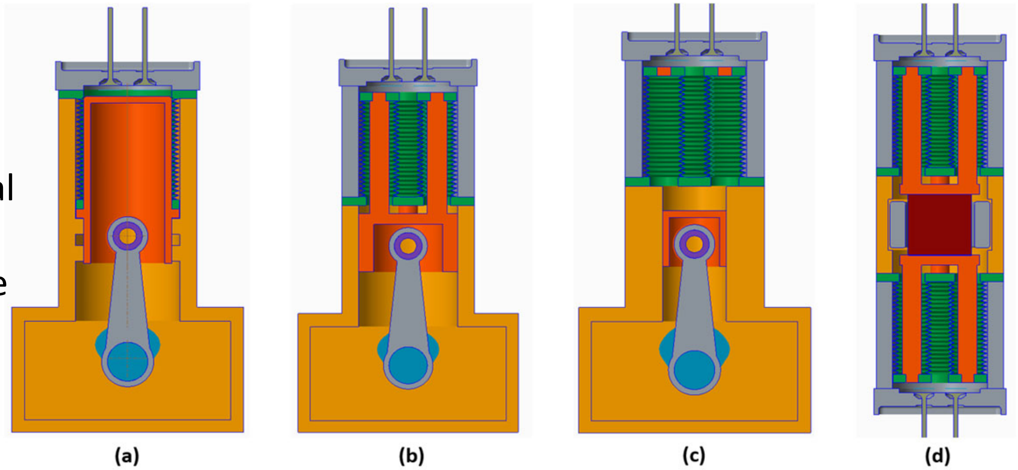
Piston Expander Mechanical Design Concepts

Diaphragm Expander

- A very large bore, short stroke configuration was anticipated for diaphragm seal life
- A range of bore-to-stroke ratios from 0.62:1 (31mm/50mm) to 19:1 (95mm/5mm) were evaluated
- Over this wide range expander performance remained quite robust.
- Volumetric efficiency improved with increasing bore, to a maximum of 41.85 percent at 67mm/10mm.

Bellows-type Piston Expander configurations using

(a) a conventional piston-crank-slider mechanism
(b) multiple pistons to reduce bellows diameter and stresses
(c) high-pressure hydraulic fluid coupling between the crank/piston and bellows to reduce the pressure differential across the bellows
(d) a linear alternator configuration flooded with high-pressure hydraulic fluid to minimize bellows pressure differential while providing direct electrical generation from the reciprocating motion



Technology Gaps & Further Development

1. High-Temperature Expander Detail Design including
    1. sealing,
    2. lubrication
    3. valving
2. Heat Exchanger Optimization and Integration
    1. Heater
    2. Cooler
    3. Recuperator
3. Robust Minimal Inventory Control System Development
4. Engine Integration
    1. On-truck location
    2. Shaft coupling

This kW-scale sCO2 WHR system development path

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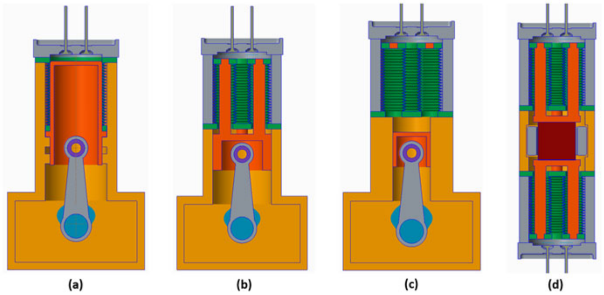

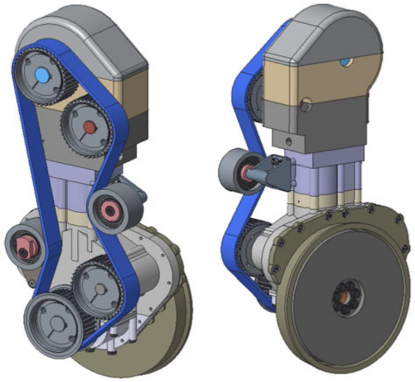

The core Phase 1 question was answered:

There is at least one suitable mechanical device which can provide good expander and compressor performance at the power scale required by these small applications

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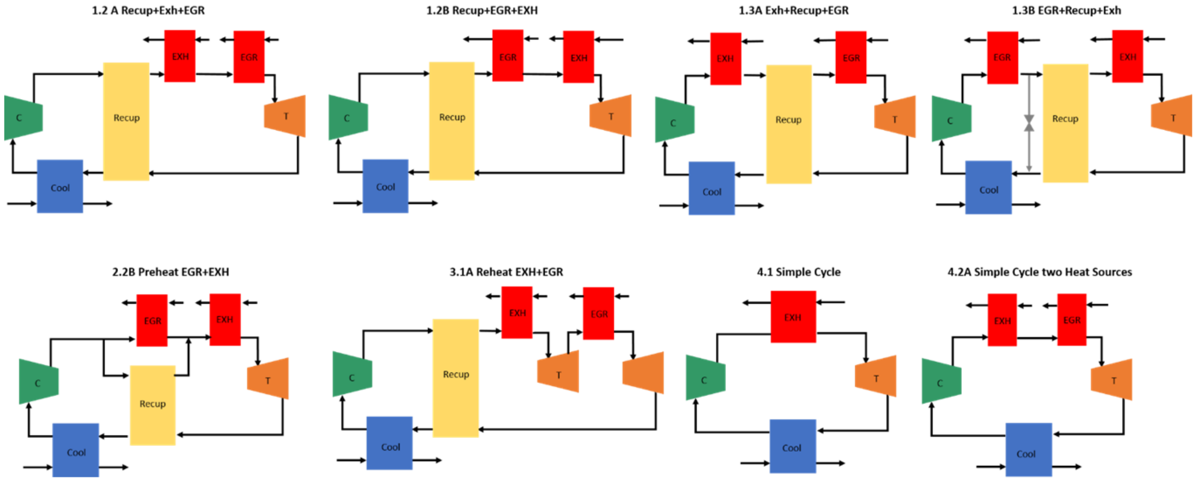
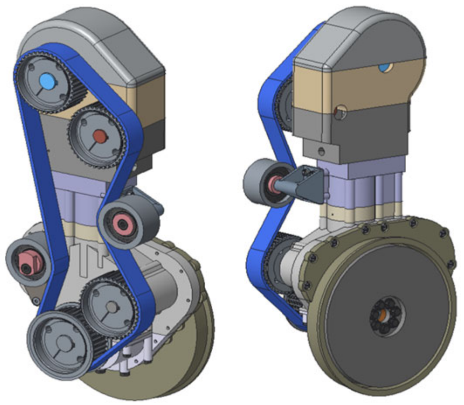




ADDITIONAL MATERIAL

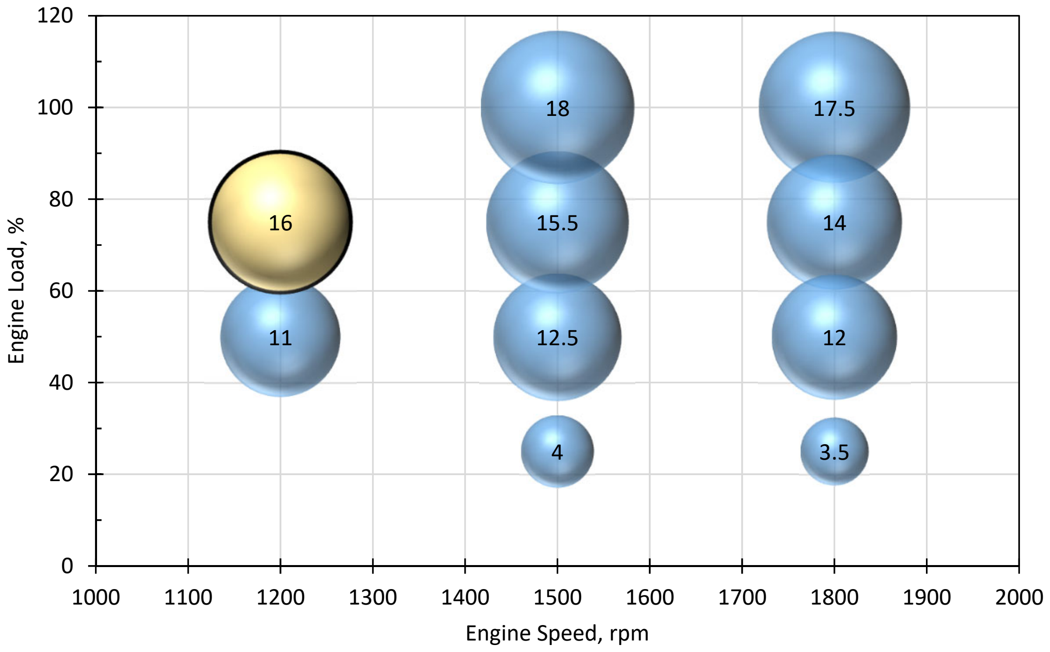
Off-design cycle performance

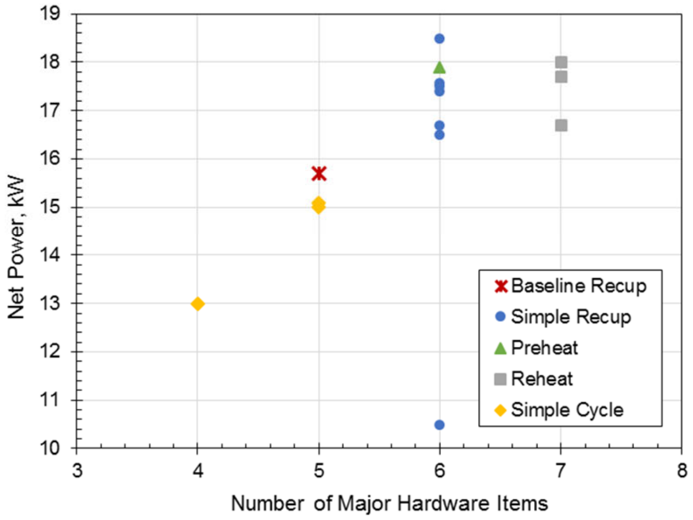
Off-design analyses for the HD truck cycle were completed using engine conditions from a 13-mode steady state test matrix that covers the full operating range of the engine in the truck

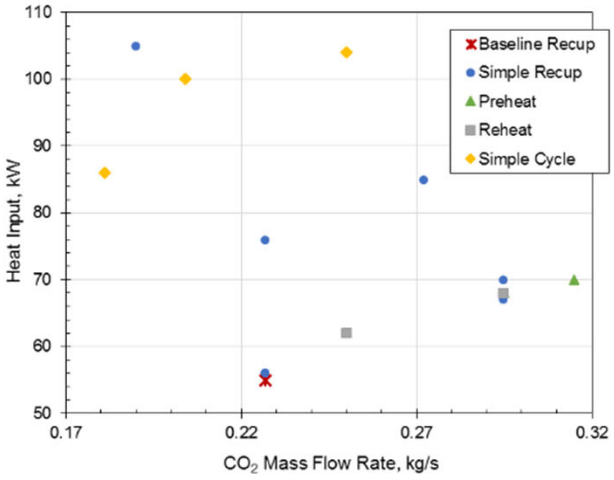
From the baseline cycle design-point conditions,

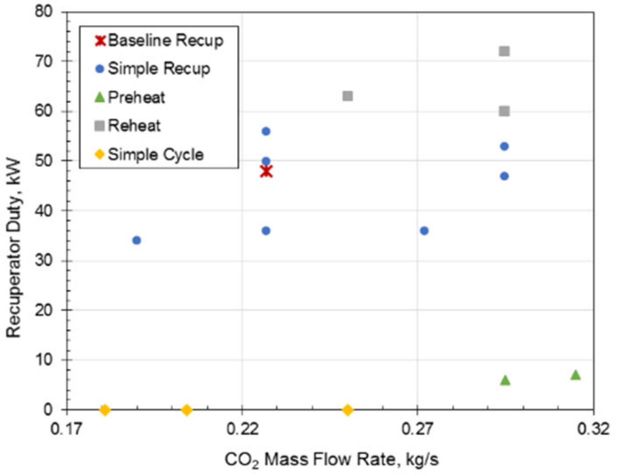
- CO_2 mass flow rate for this case was 0.272 kg/s and was held constant
- the heat source temperature and flow rate were altered
- Low-side and high-side cycle pressures were held constant, as well as the cooler air flow.

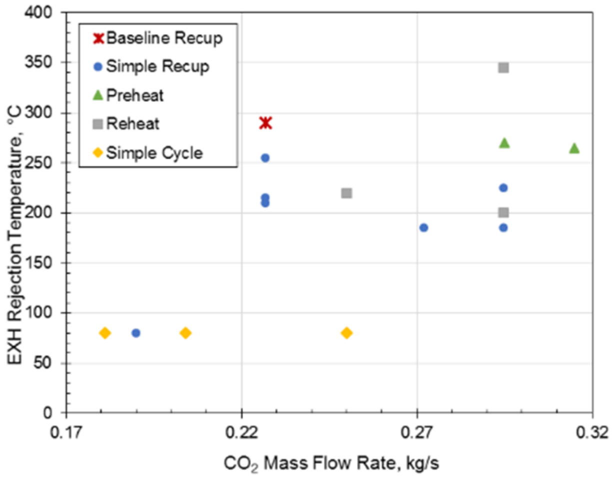
The aim was to understand how the simple recuperated sCO_2 WHR recovery cycle would respond with only a change in the exhaust heat source conditions




Net Power, kW
19
18
17
16
15
14
13
12
11
10
3
4
5
6
7
8
Number of Major Hardware Items
✖ Baseline Recup
● Simple Recup
▲ Preheat
■ Reheat
◆ Simple Cycle


Heat Input, kW
110
100
90
80
70
60
50
0.17
0.22
0.27
0.32
CO_2 Mass Flow Rate, kg/s
✖ Baseline Recup
● Simple Recup
▲ Preheat
■ Reheat
◆ Simple Cycle


Recuperator Duty, kW
80
70
60
50
40
30
20
10
0
0.17
0.22
0.27
0.32
CO_2 Mass Flow Rate, kg/s
✖ Baseline Recup
● Simple Recup
▲ Preheat
■ Reheat
◆ Simple Cycle


EXH Rejection Temperature, °C
400
350
300
250
200
150
100
50
0
0.17
0.22
0.27
0.32
CO_2 Mass Flow Rate, kg/s
✖ Baseline Recup
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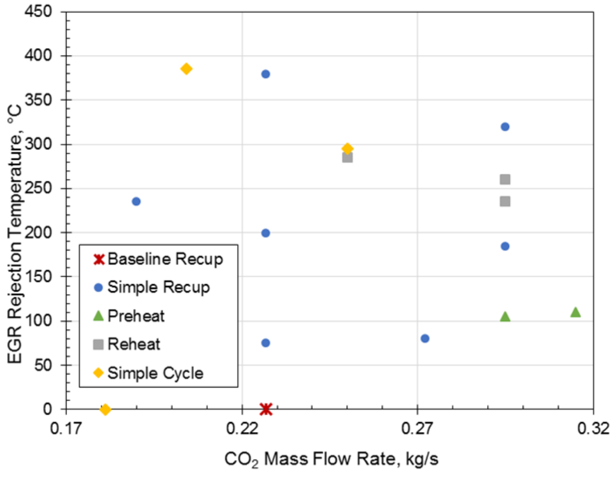
◆ Simple Cycle


EGR Rejection Temperature, °C
450
400
350
300
250
200
150
100
50
0
0.17
0.22
0.27
0.32
CO_2 Mass Flow Rate, kg/s
✖ Baseline Recup
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