Development of A Near-isothermal Compressor for Transcritical Carbon Dioxide Cycle

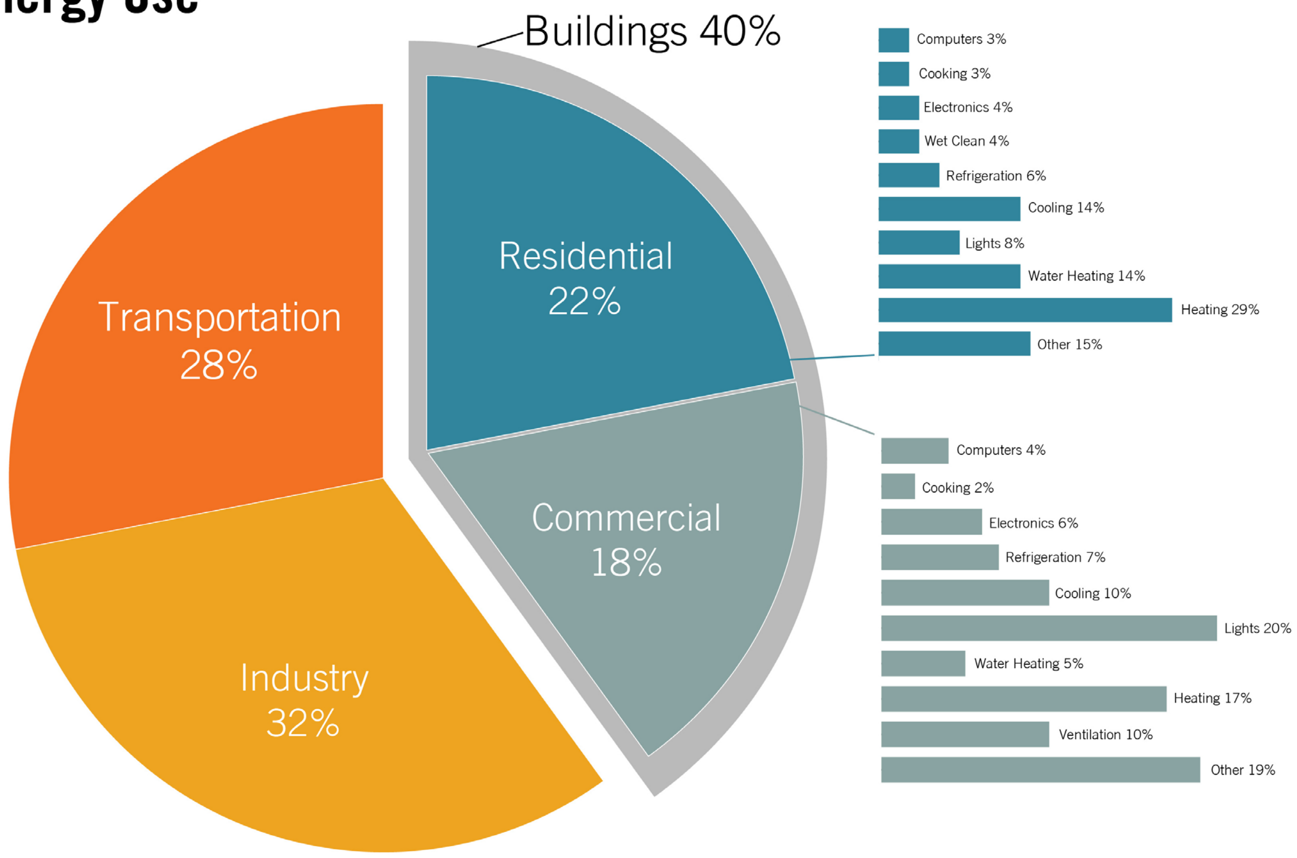
Cheng-Yi Lee, Timothy Kim, Jan Muehlbauer,

Yunho Hwang*, Reinhard Radermacher

Center for Environmental Energy Engineering
Department of Mechanical Engineering, University of Maryland,

Introduction: Building Energy Use

U.S. Energy Use

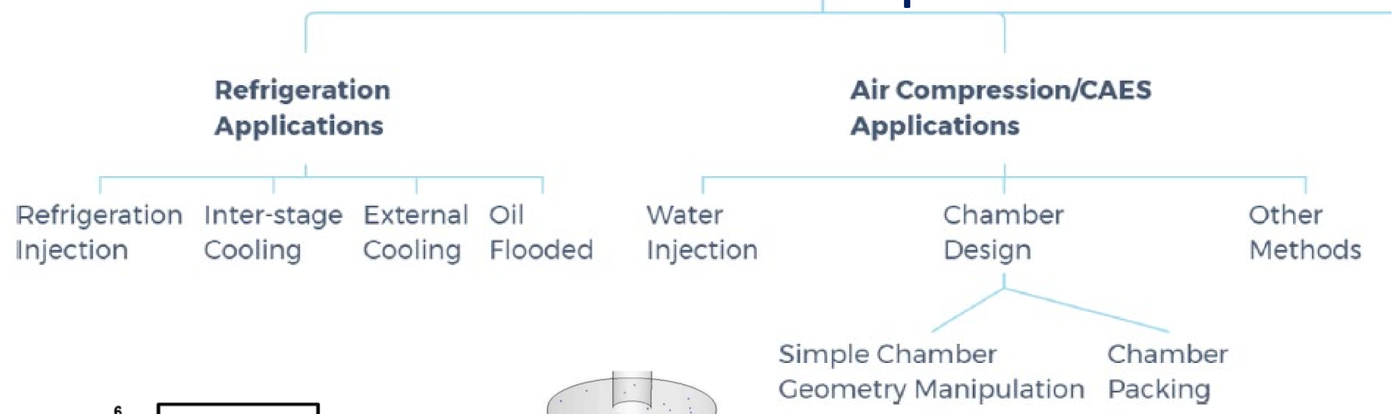


https://www.pae-engineers.com/news/articles/the-time-is-right-for-zero-energy-buildings

- The buildings sector accounts for about **76% of electricity use and 40% of all U. S. primary energy use** and associated greenhouse gas (GHG) emissions
- About **19% of all U.S. primary energy use** is related to heating, air conditioning, and refrigeration.

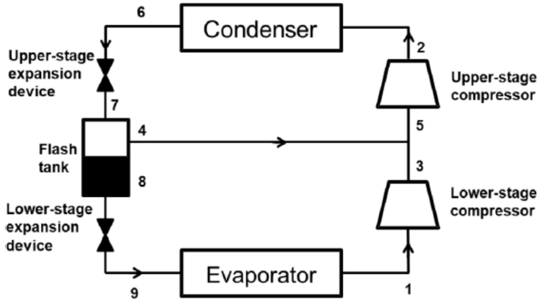
Approach

Isothermal Compression



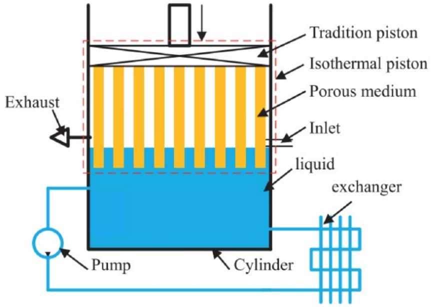
Design Considerations

- Liquid Piston
- CO_2 Transcritical Cycle

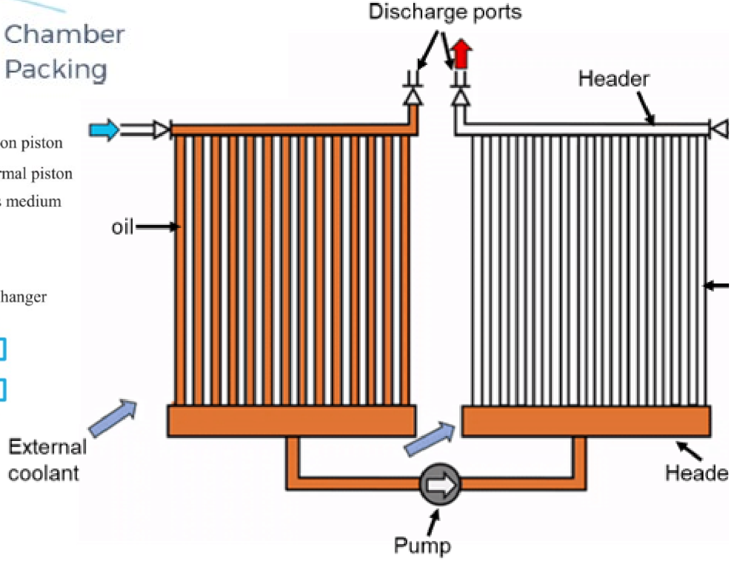


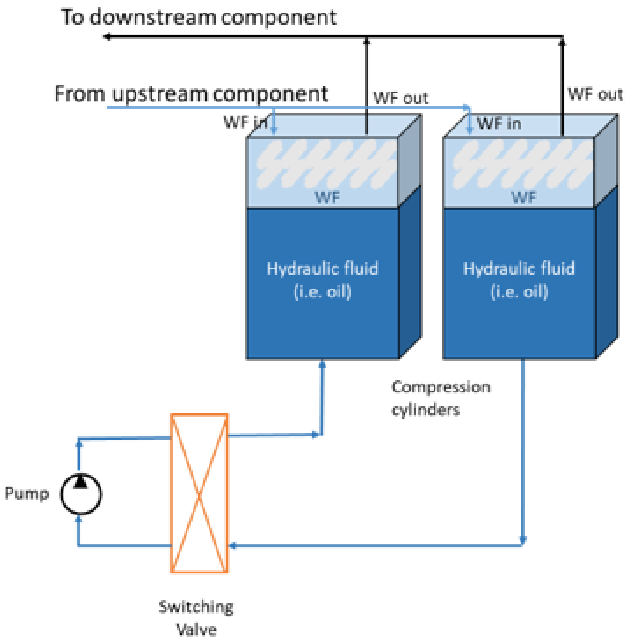
Refrigerant Injection

Liquid Spray



Porous Medium



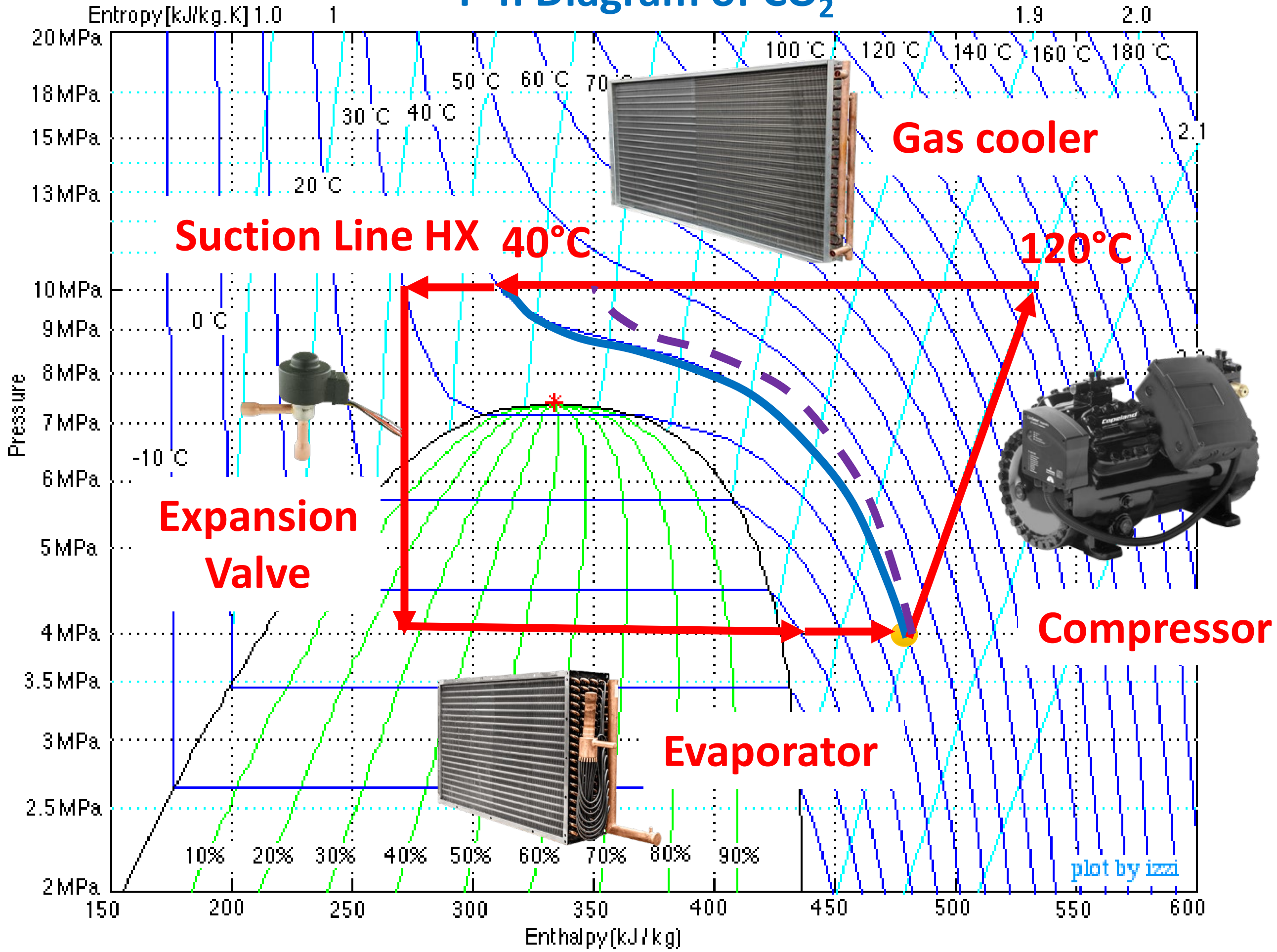


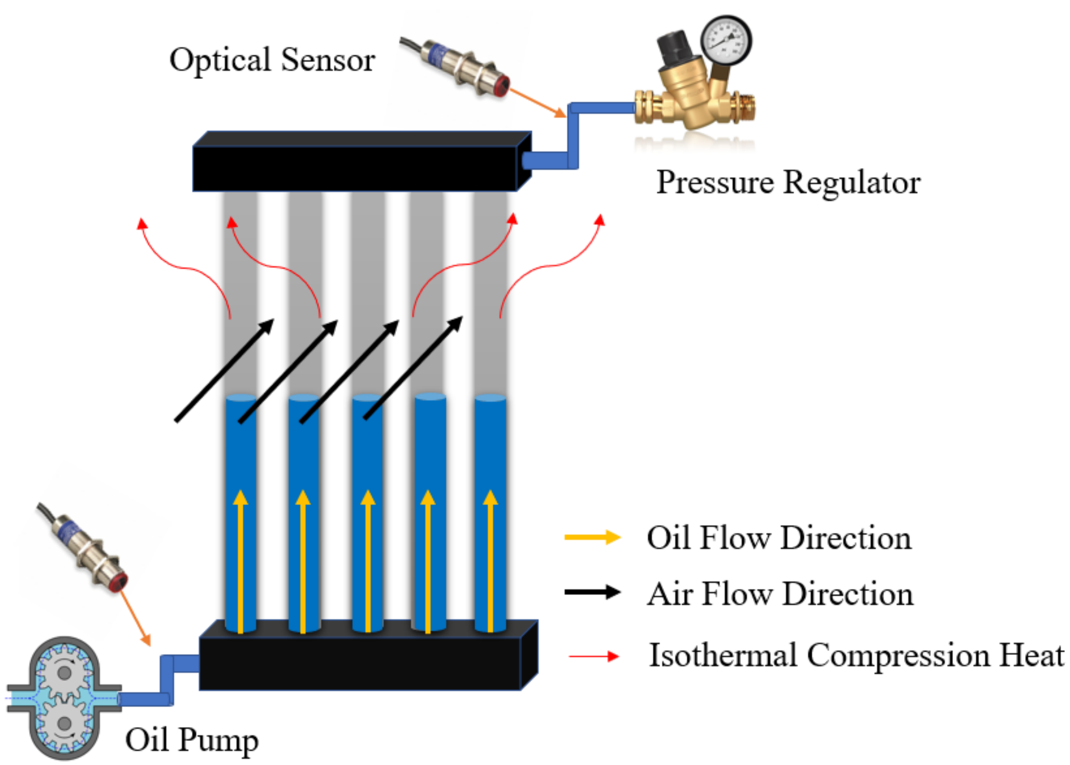
Kim et al., IJR, 144 (2022) 145–162

Compressor Within Heat Exchanger | Heat Exchanger Within Compressor

Introduction: Isothermal Compression Mechanism

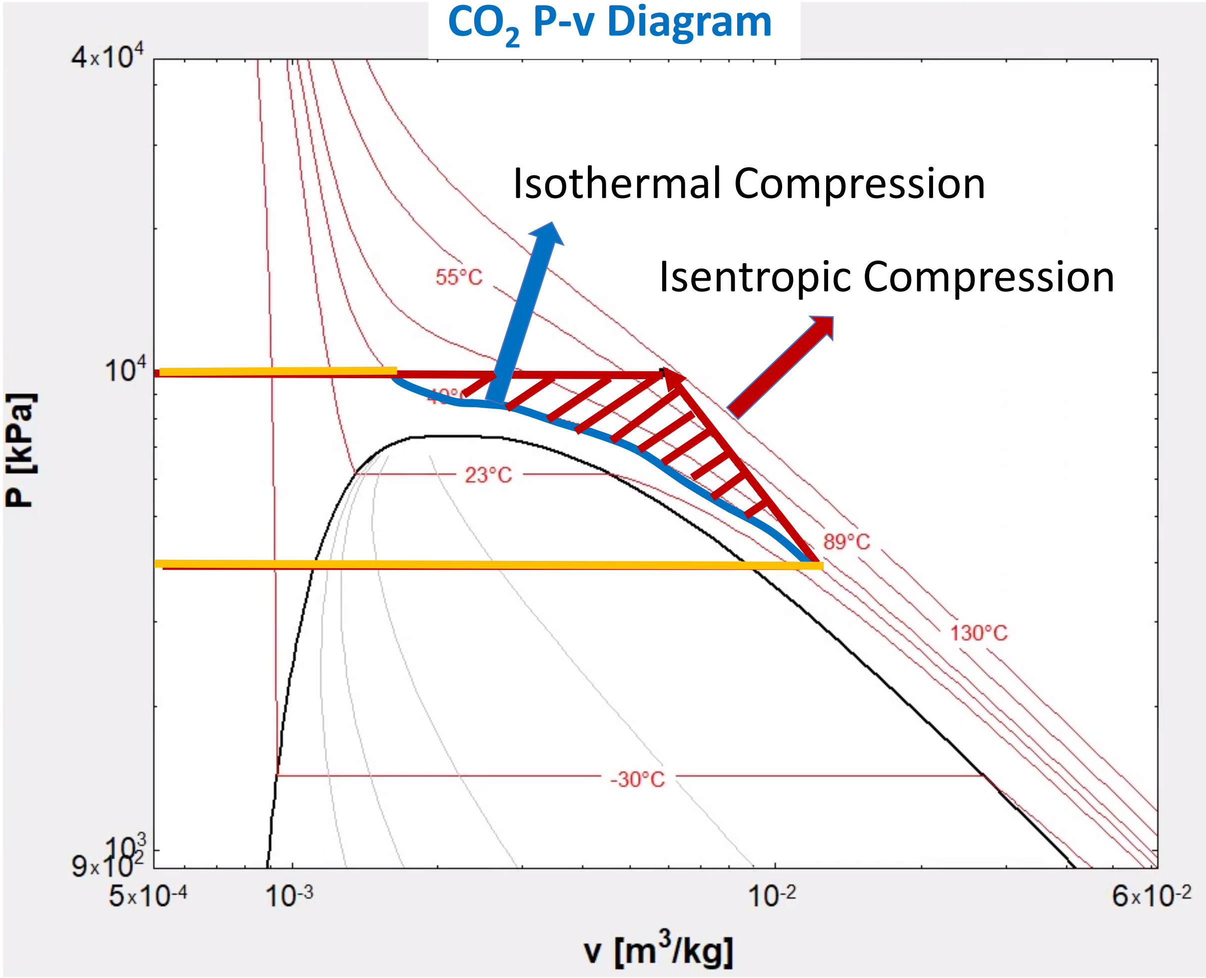
P-h Diagram of CO_2





Isothermal Compression Mechanism

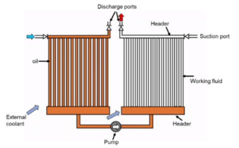
Introduction: Isothermal Compression Mechanism



- Isentropic Compression Work = 48.5 kJ/kg
- Isothermal Compression Work = 32.9 kJ/kg
- Ideal Work Saving = 15.6 kJ/kg
- Work Saving Percentage = 32.1%

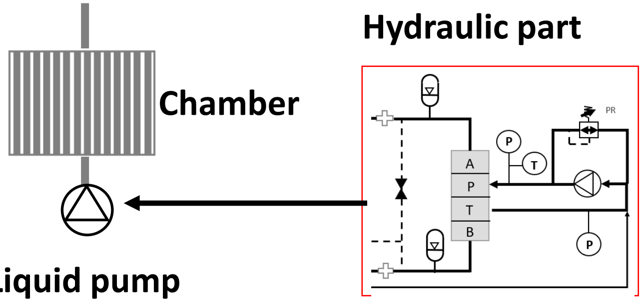
Approach

Chamber Concept



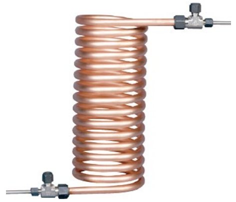
Generic EES Code

Isothermal compressor



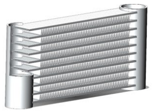
EEV

Electric heater



Subcooler

HX Design and Selection

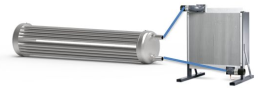


Microchannel HX (1.3 mm)

Potential Isothermal Chamber Design

Plate Fin HX (5 mm OD)

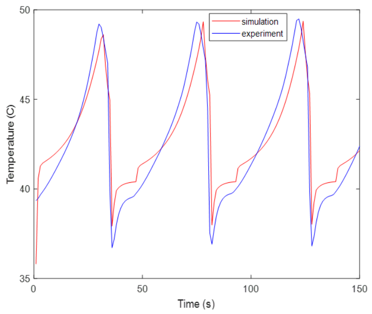
Finned Tube HX (4.8 mm OD)



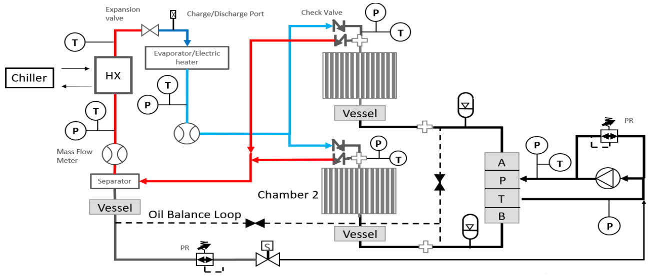
Shell and Tube HX
Water as Secondary Loop

Small Diameter Round Tube HX
(0.1 – 0.9 mm OD)

Transient Model Validation



Isothermal Compression Experiment



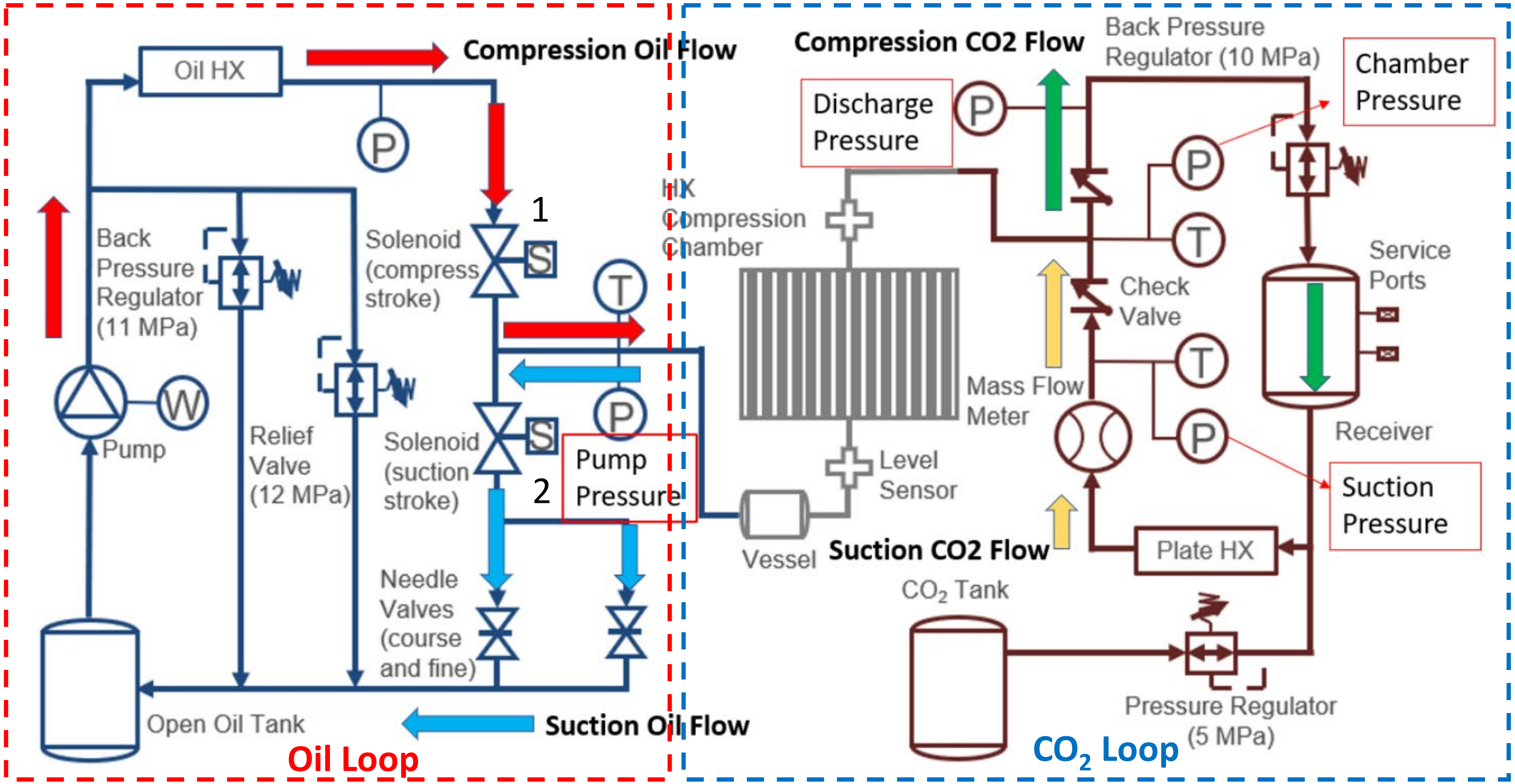
Objectives

- Design an open-loop liquid piston isothermal compressor that can deliver 1.75 kW of cooling capacity.
- Achieve at least 85% of the isothermal efficiency by integrating the compressor with the gas cooler

Compression Chamber Design Parameters

Parameter	Unit	Value
Compressor Suction Temperature	°C	40
Evaporating Temperature	°C	5
Mass Flow Rate	g/s	13.3
Cooling Capacity	kW	1.75
Operating Pressure	kPa	4,000 -10,000
Cycle Frequency	Hz	0.1
Chamber Heat Transfer Area	m^2	0.279

Experimental Facility



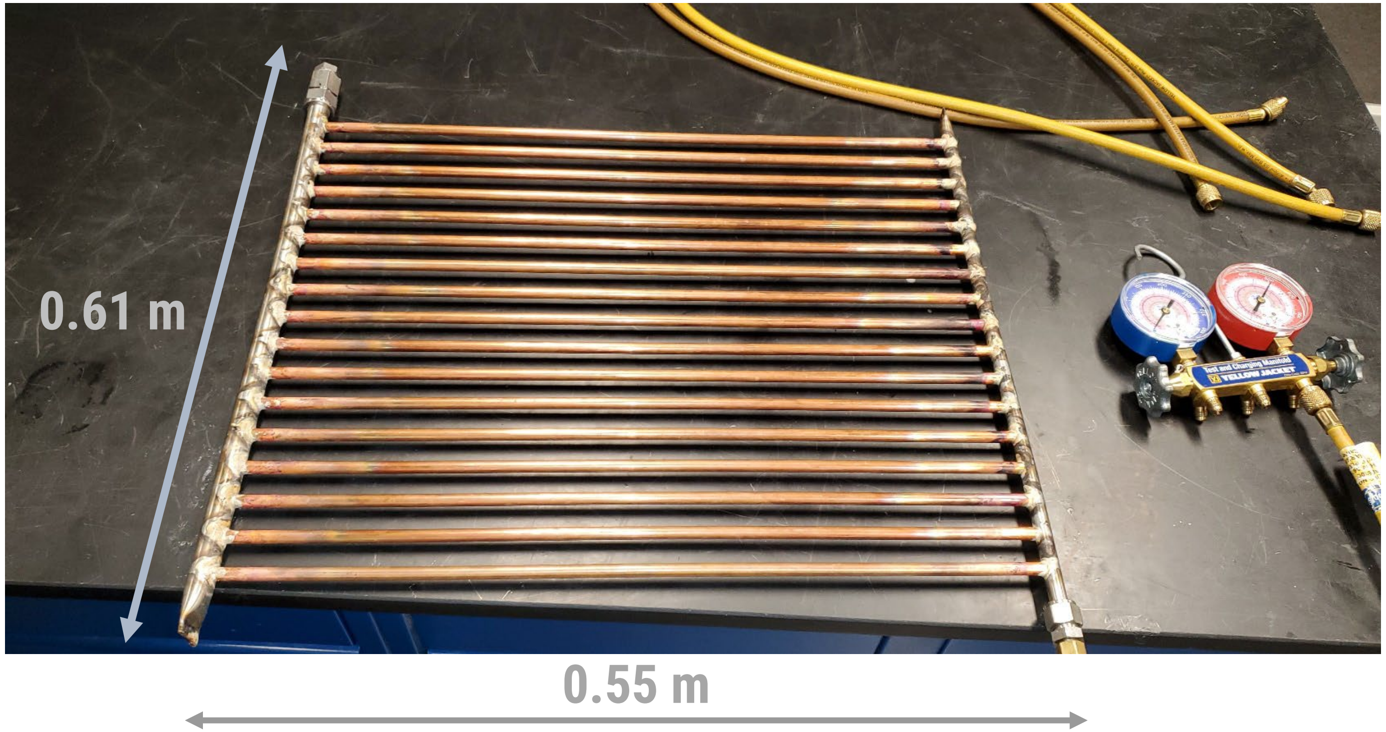
Compression Process

- Solenoid 1 open
- Solenoid 2 closed
- Oil pumped into the compression chamber
- CO_2 exits at discharge after reaching desired pressure

Suction Process

- Solenoid 1 closed
- Solenoid 2 open
- Oil drains back to oil tank
- CO_2 enters chamber from suction side at desired pressure and temperature

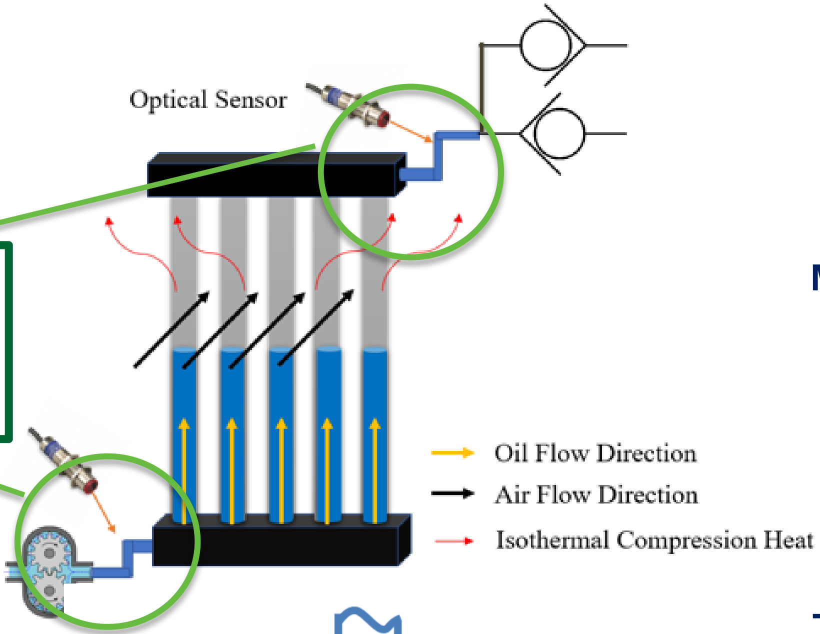
Experimental Facility – Comp. Chamber/HX



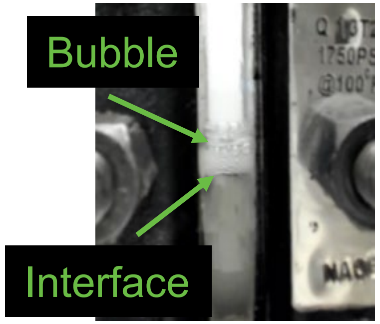
*Pressure Tested at 17,000 kPa

Parameter	Unit	Value
Upper Header Diameter	mm	10.9
Lower Header Diameter	mm	16.6
Tube Diameter	mm	6.2
Tube Pitch	mm	19
Center to Center Tube Pitch	mm	23.8
Heat Exchanger Length	mm	609.6
Heat Exchanger Width	mm	548.6

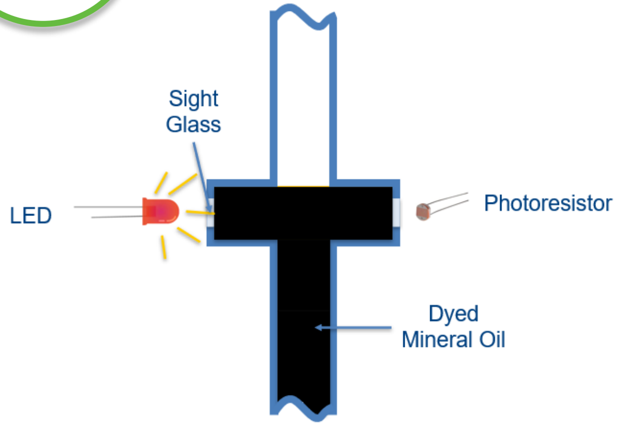
Experimental Facility



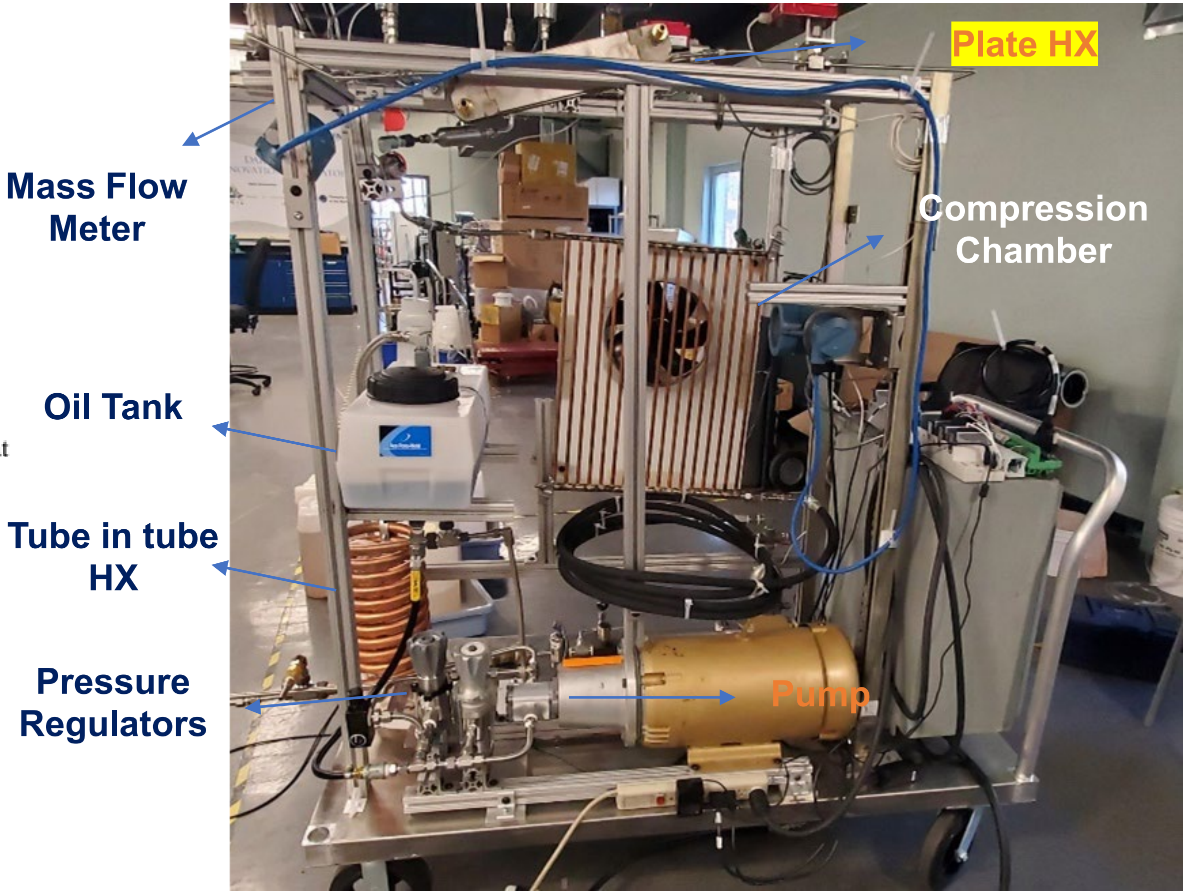
The determination of the compression start, and end is crucial



CO_2 Bubble Formation



Optical Sensor



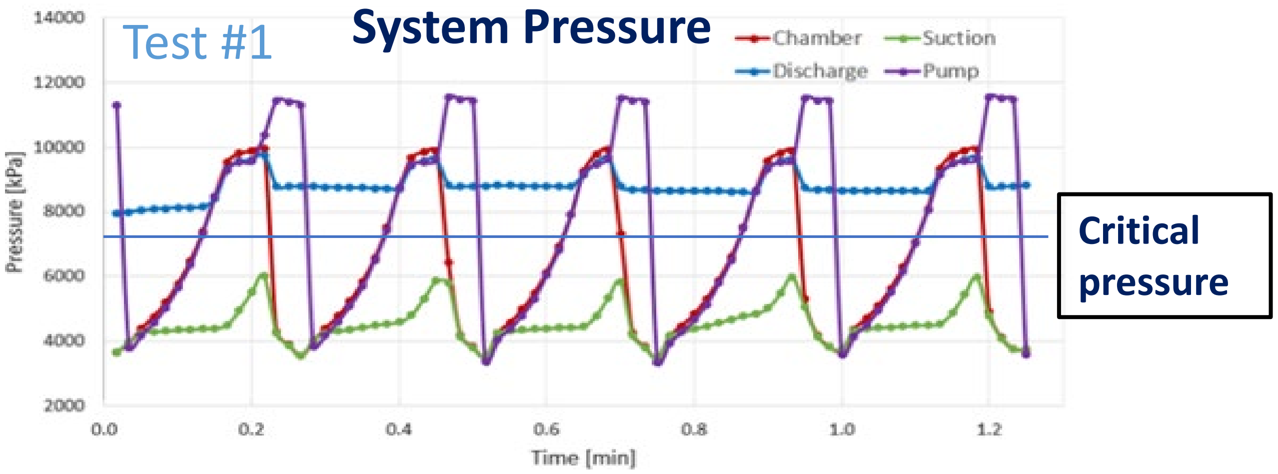
Test Facility Picture

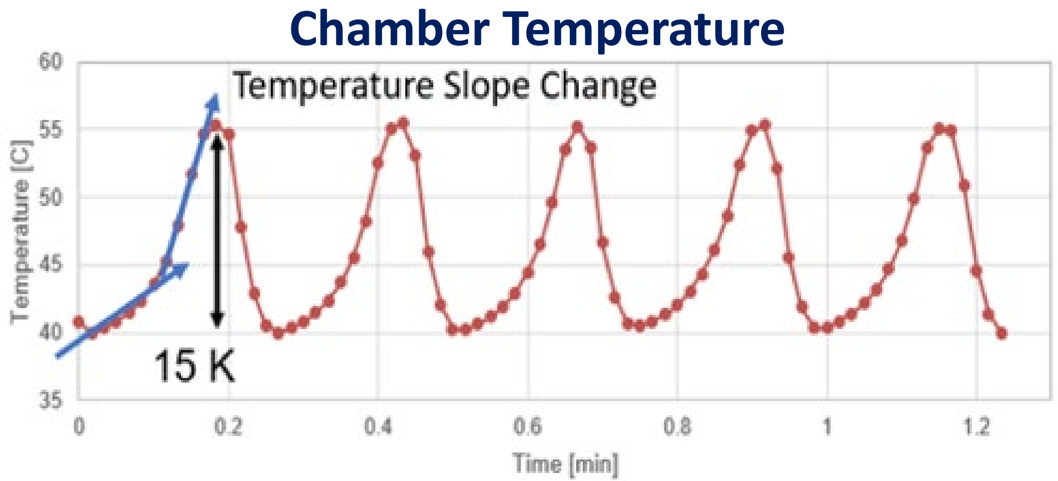
Test Matrix

Test Number	Initial Chamber Temperature [°C]	Initial Chamber Pressure [kPa]	Final Chamber Pressure [kPa]	Pump Speed [RPM]	Water mist spray (35°C)	Axial Fan Number
#1	40	4,000	10,000	900	No	1
#2	40	4,000	10,000	270	No	1
#3	40	4,000	10,000	270	Yes	2

* Test room temperature: 35°C, Humidity: 20%

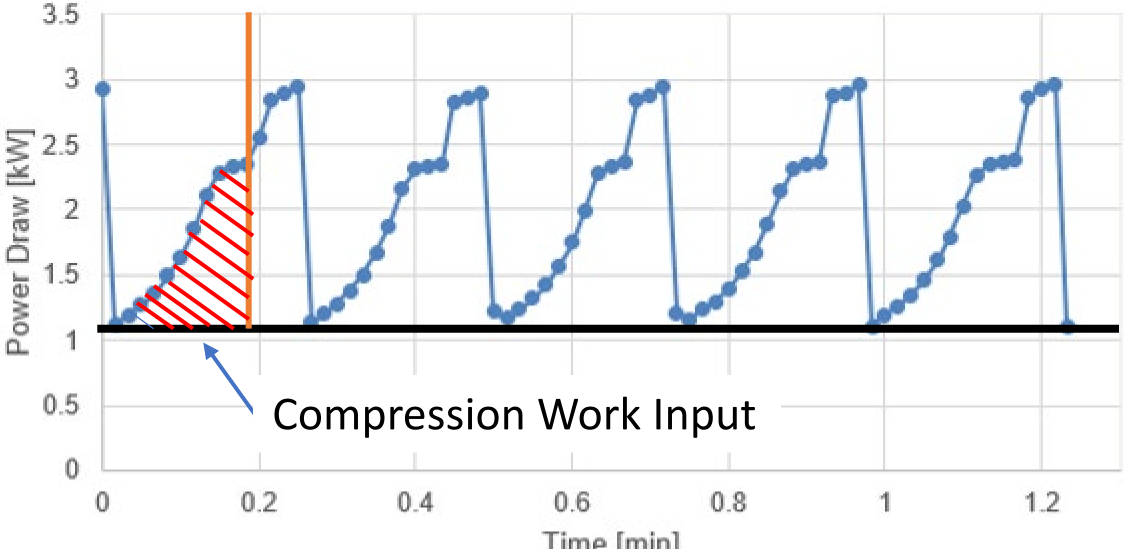
Results and Discussion - Test #1

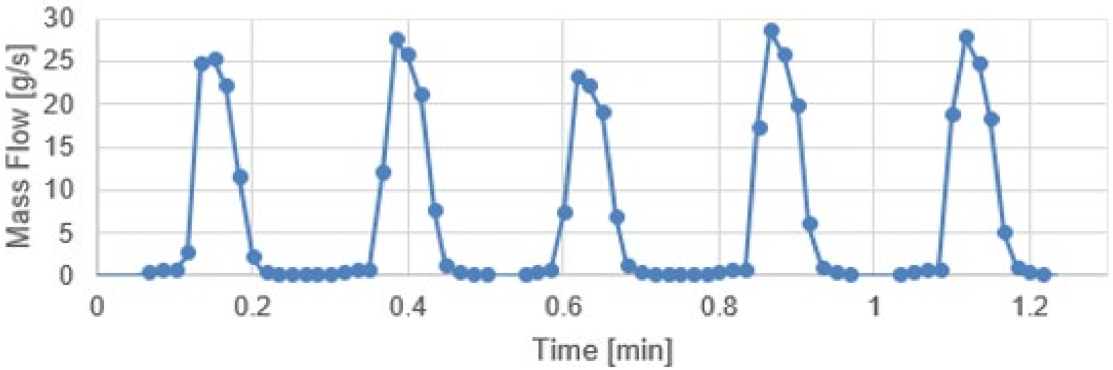




- The CO_2 refrigerant was successfully compressed over the critical pressure.
- While the isentropic compression would result in a 117°C, the current compression increased to 55°C with a 15 K lift.
- A noticeable slope change was observed in the temperature chart, which resulted from the reduction of the heat transfer area and the larger enthalpy difference at a higher-pressure range.

Results and Discussion- Test #1





Isothermal Efficiency

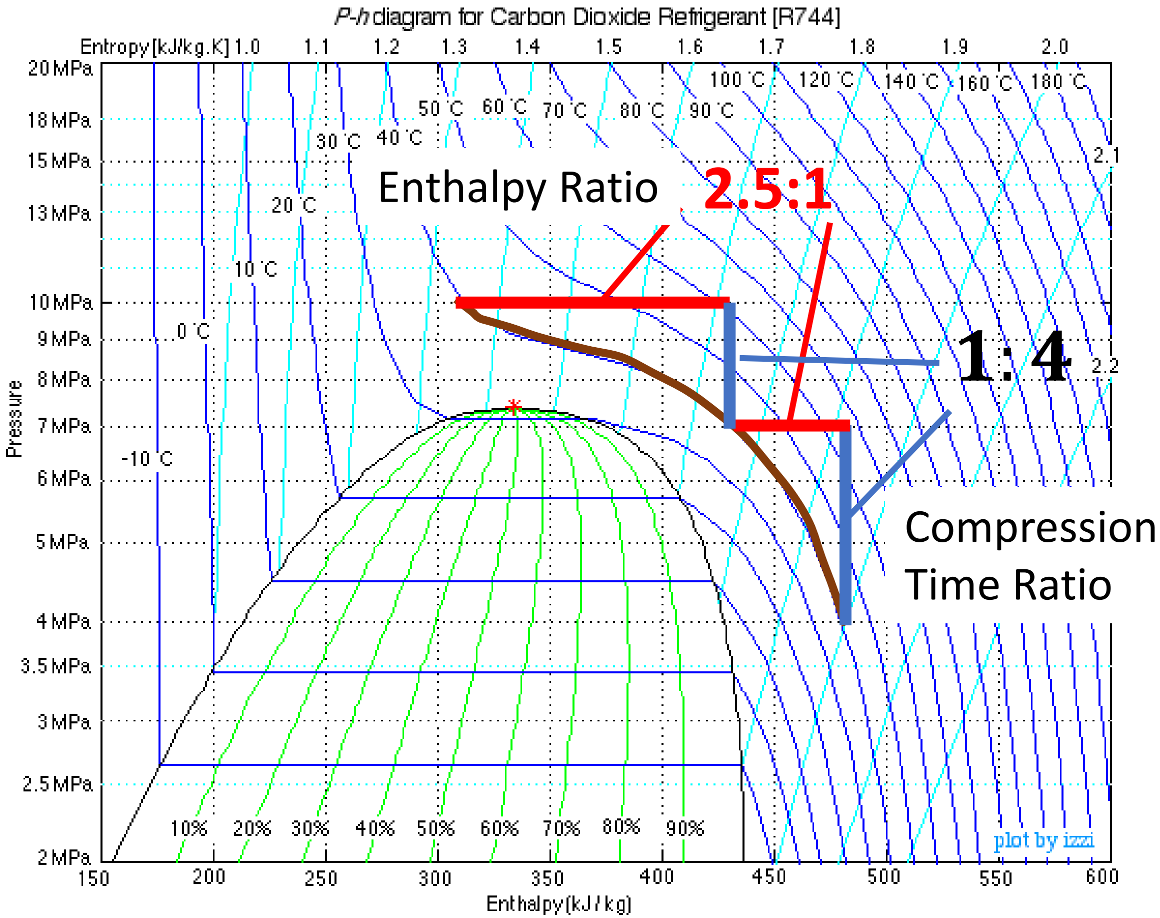
$$\eta_{iso} = \frac{W_{isothermal}}{W_{actual}} = \frac{Mass_{chamber} \cdot w_{isothermal}}{W_{integrated} \cdot \eta_{motor+pump}}$$

Motor and Pump Efficiency

$$\eta_{motor+pump} = \frac{W_{actual\ work}}{W_{integrated\ work}}$$

- The isothermal efficiency of Test #1 can only achieve 59.7 %.
- The cooling capacity reached the design capacity of 1.75 kW based on the measured mass flow rate.

Results and Discussion - Test #1

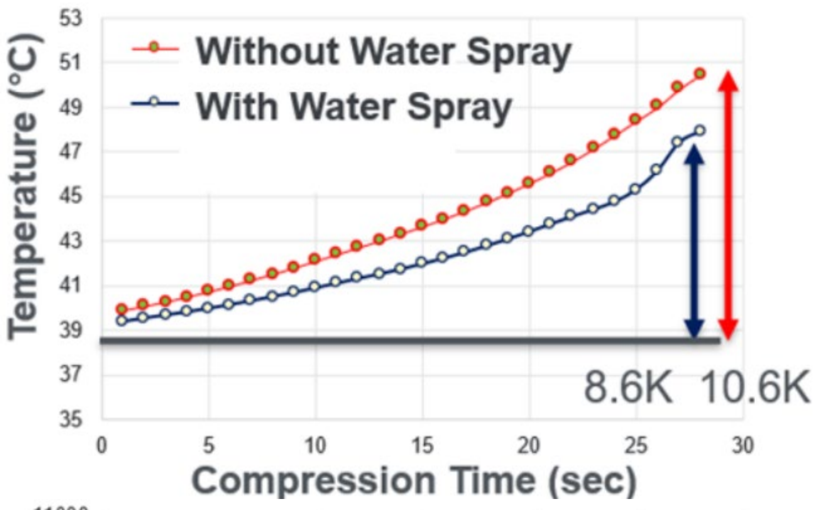


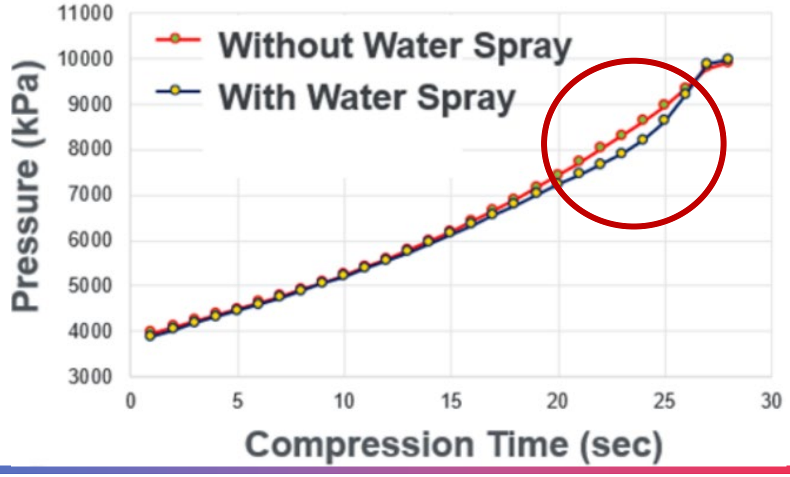
CO_2 P-h Diagram

- Challenges for liquid piston isothermal compressor:

1. Heat transfer area decreases as the liquid piston elevates the oil level
2. Heat transfer requirement from 7,000 – 10,000 kPa is ten times higher than that of 4,000 - 7,000 kPa

Results and Discussion - Test #2 and #3





- The operation speed was reduced to 270 RPM for higher isothermal efficiency. In Test #3, evaporative cooling was applied on the chamber's surface.
- The temperature chart displayed a notable decrease in temperature, while the pressure chart demonstrated the effectiveness of using evaporative cooling to reduce pressure.

Results - Performance Comparison

Parameter	Unit	900 RPM	270 RPM	270 RPM with Evaporative Cooling
Energy per Cycle	kJ	6.12	3.91	3.67
Initial CO_2 Density	Kg/m^3	84	83	81
CO_2 in Chamber	g	107	106	13.5
$W_{isothermal}$ / W_{actual}	%	59.7	80.0	93.8
Average CO_2 Mass Flow Rate	g/s	7.4	2.9	3.2
Cooling Capacity	W	1,719	674	743

Conclusions

- A compressor with near-isothermal properties was constructed by combining a liquid piston pump with a gas cooler. During testing, the prototype exhibited the capability of continuously compressing CO_2 beyond the critical point.

- An isothermal efficiency of 60% was achieved for 900 RPM operation, 80% for 270 RPM operation, and 94% for evaporative cooling enhancement.

Future Work

- Develop a Highly Efficient Isothermal Compressor Chamber Design

Potential Isothermal Chamber Design

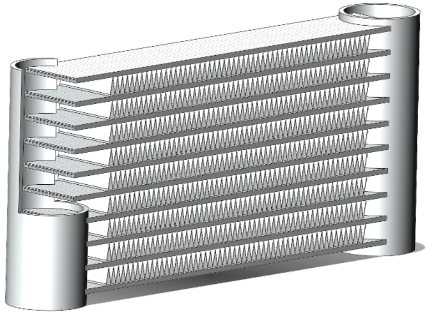


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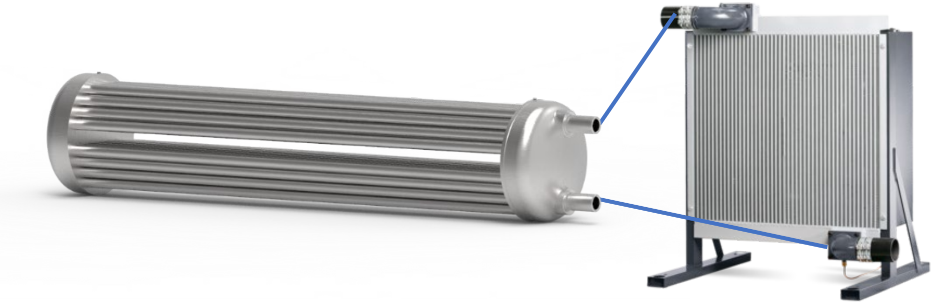
Plate Fin HX (5 mm OD)



Shell and Tube HX
Water as Secondary Loop

Finned Tube HX (4.8 mm OD)

Small Diameter Round Tube HX
(0.1 – 0.9 mm OD)

Acknowledgments

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