Dynamic Modeling Methodology for Near Isothermal Compressor

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Abstract

Compressors are the vital component of the vapor compression systems and account for the majority of energy consumption. Developing appropriate controllers or optimizing compressor design can significantly reduce the carbon emissions. The isothermal compressor combines the compressor chamber and gas cooler, using the liquid piston to compress the working fluid for near-isothermal compression. This methodology can reach up to 30% energy saving compared to the traditional isentropic compression work. This paper leverages the CEEE Modelica Library (CML) to demonstrate a detailed isothermal compressor model that captures the near-isothermal compression process of transcritical carbon dioxide (CO_2) cycle. The model uses the real experimental data as the boundary conditions, and the relevant component-level experimental validation was carried out by using a prototype with 1-ton nominal capacity. The results proved the accuracy of the dynamic model (7.5% relative error for chamber pressure and 0.74 K deviation for chamber temperature), and provide a guideline for designing the isothermal compressor chamber. Finally, the modeling for the isothermal compression cycle is ongoing and the filed is still in its infancy.

Keywords: Isothermal Compressor, Transcritical CO_2 Cycle, Dynamic Modeling

1 Introduction

Vapor compression system (VCS) are extensively utilized in heating, ventilation, and air conditioning (HVAC) area, which together account for more than 30% of electricity generated in the U.S. (EIA, 2022). The compressor, responsible for circulating refrigerant and transferring heat, consumes the majority of this electricity. Therefore, measures to improve the system energy efficiency, especially the innovative design of compressor, can significantly reduce the carbon footprint and boost the resilient energy economy.

The ideal thermodynamic cycle (Carnot cycle) defines the upper limit on the efficiency of refrigeration system in creating a temperature difference through the application of work to the system. In reality, it's not possible to build such thermodynamically reversible engine and the real engines that even operate along the Carnot cycle style (isothermal expansion / isentropic expansion / isothermal compression / isentropic compression) are rare. However, the isothermal compression can bring the system close to the Carnot cycle efficiency, and the related technology have achieved breakthrough progress recently, especially with the widespread adoption of compressed air energy storage (CAES) that driven by the increasing penetration of renewable energy sources (Kim et al., 2022). Given that one of the biggest problems come with CAES is low energy efficiency, i.e., the traditional CAES systems lose energy due to heat generated during the compression, which cannot be fully recovered. A considerable number of studies have explored achieving isothermal compression in CAES applications, including water injection (Patil et al., 2020, Odukomaiya et al., 2016), chamber shape optimization (Zhang et al, 2016) and chamber packing with inserted material (Yan et al, 2015, Saadat et al, 2012).

On the other hand, CAES systems emit greenhouse gases, which pose challenges to the goals of reducing greenhouse gas emissions. In view of long-term environmental safety, one potential substitute refrigerant is carbon dioxide (CO_2), a natural refrigerant that has negligible impact on climate change, which is environmentally benign, non-toxic, and non-explosive. In addition, recent advancements in system design and manufacturing improvements make it possible to achieve high pressure required for CO_2 transcritical operation, and the CO_2 can deliver much higher heat rejection through sensible cooling to regain efficiency when it's compressed beyond critical point (31.1 °C, 7.38 MPa). The CO_2 transcritical cycles have many different system configurations for different applications (Sarkar et al., 2004, Fernandez et al., 2010), with ongoing tightening environmental regulations, further theoretical research is needed to explore the potential for enhancing the energy efficiency of transcritical CO_2 systems.

The CEEE Modelica Library (CML) is a comprehensive Modelica library developed by the Center for Environmental Energy Engineering (CEEE), University of Maryland, College park. It is designed for transient simulation of extensive thermal system configurations and HVAC applications. The CEEE can assist in gaining a deeper understanding of thermodynamic systems. The CML is scalable, allowing users to virtually assess and optimize the VCS's performance. Two features in our implementation are tailored for modeling of isothermal compressor model in CML: (1) Using the liquid piston to compress the working fluid (CO_2) within the heat exchanger-based chamber to enhance the heat transfer. (2) The compression chambers can realize the double-acting mechanism, allowing compression and suction processes to occur simultaneously. The modeling details are elaborated in Section 2, while Section 3 covers the information about our experimental setup and the relevant experimental validation. Finally, the conclusions and future work are summarized in Section 4.

2 Modeling Methodology

In our test unit, a two-chamber isothermal compressor design with shared liquid pump is adopted. Each compression chamber is based on the plate heat exchanger (PHX), with the secondary fluid (water) provided by the air-cooled radiator to cool down the compressed refrigerant. Additionally, both the residual gas cooler and suction line heat exchanger are PHXs as well, utilized to ensure the rated cooling capacity of 1 ton. The piston accumulator is used as the storage component to regulate the system charge level and mitigate pressure fluctuation. The electronic expansion valve (EXV) controls the downstream pressure and mass flow rate, while the electric heater acts as the evaporator to control the refrigerant state at the suction side of the isothermal compressor by regulating its heat load.

The schematic diagram of the test facility with sensors installed is shown in Figure 1. Sensors are in place to measure key operation variables such as pressures, temperatures and mass flow rates, etc. In Figure 1, letters 'T' and 'P' represent the temperature and pressure measurement via thermocouples and pressure transducers, respectively, which can provide us with information about the refrigerant state for different components. The hydraulic directional control valve is installed to facilitates the switching of flow direction when the isothermal chamber completes the compression/ suction stroke, and the oil level reaches the upper/ lower oil level sensor. The check valves were utilized to minimize the dead volume in each compression cycle and the mass flow meter is uilized to measure the suction side mass flow rate of the isothermal compressor.

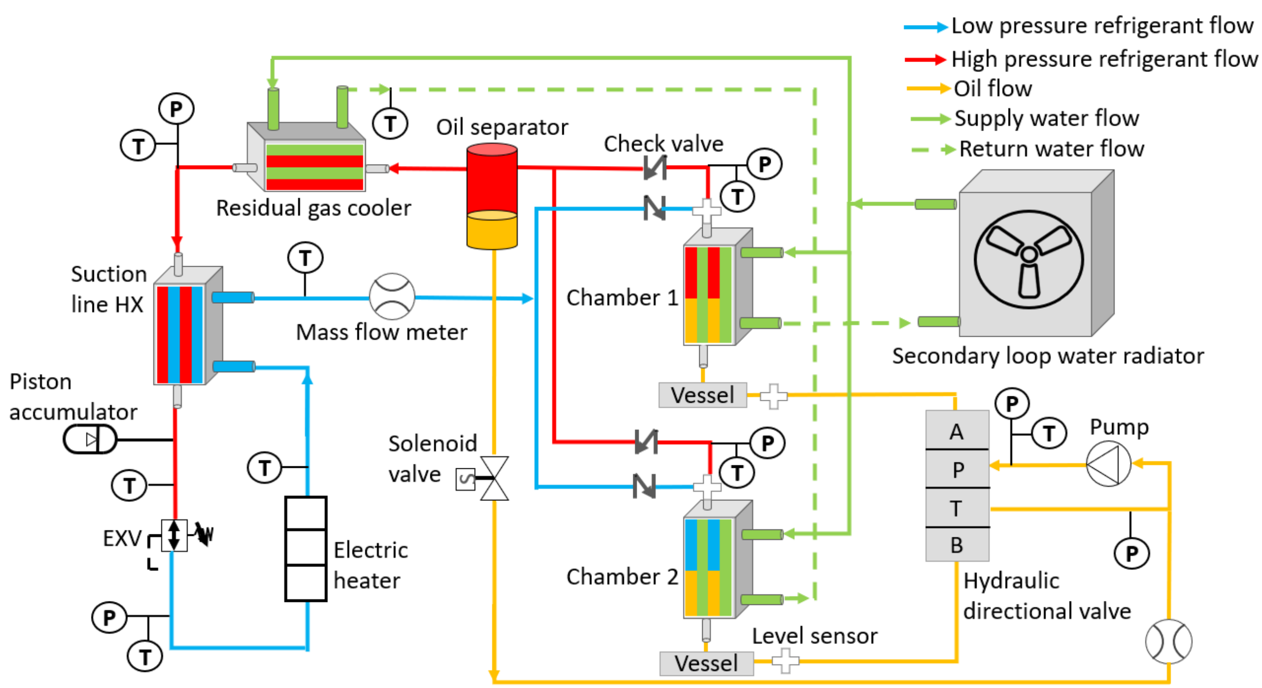

Figure 1. Sensor instrumentation diagram for the test unit.

The Modelica interface of the PHX-based isothermal compressor is depicted in Figure 2 and following assumptions are made for the modeling:

1. The refrigerant side boundary conditions (e.g., pressure and enthalpy) of the isothermal compressor were provided based on the experimental data, while the water-side boundary

10.3384/ECP20715 DOI

conditions were assumed to be constant for simplification.

2. The liquid piston model is simplified as the input to the isothermal compressor, which compresses or suck in the refrigerant within the chamber under the given volumetric flow rate and time period.
3. For each channel of the PHX-based isothermal compressor, the geometric details and the flow conditions (e.g., mass flow rate and temperature) for both primary and secondary fluid were assumed to be the same. Therefore, the individual channel was selected for the modeling, and the corresponding results were multiplied by half the total number of plates to derive the overall component results.
4. The process of solubility / degassing of CO_2 in oil during the compression/ suction is too complex and can affect the charge estimation within the isothermal compressor. Therefore, for simplification, the oil is assumed to be mineral oil with no solubility. Additionally, the model does not consider the heat transfer between the oil and CO_2.

The geometric details of the PHX are shown in Table 1.

Table 1. Geometric details of the PHX.

Parameter	*Value*
Port to port length (mm)	329
Width (mm)	119.5
Corrugation depth (mm)	1.55
Area enlargement factor	1.24
Total number of plate	50
Diamter of port (mm)	23.5

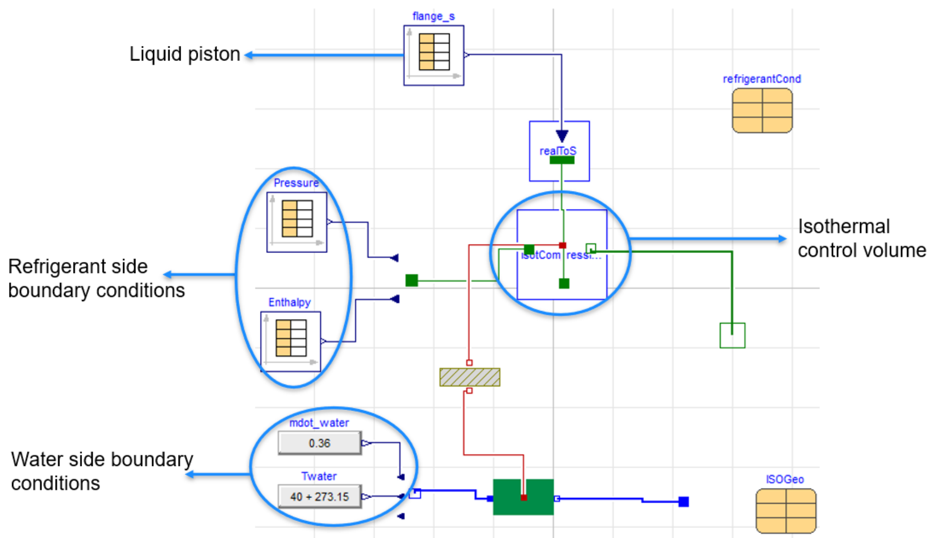

Figure 2. Modelica interface of the isothermal compressor model.

The conservation differential equations for the refrigerant energy, mass and tube wall energy for the isothermal compressor are given in Equation 1 to Equation 3.

$$\dot{U} = \dot{m}_{in}h_{in} - \dot{m}_{out}h_{out} - \alpha_r A\left(T_r - T_w\right) \quad (1)$$

$$\dot{m}_e = \dot{m}_{in} - \dot{m}_{out} \quad (2)$$

$$\dot{E} = C_{th,w}\dot{T}_w = \alpha_r A\left(T_r - T_w\right) - \alpha_{water} A\left(T_w - T_{water}\right) \quad (3)$$

where U is the refrigerant internal energy; $\dot{m}_{in}$ and $\dot{m}_{out}$ represent the inlet and outlet refrigerant mass flow rates of the chamber, respectively; h_{in} and h_{out} represent the inlet and outlet refrigerant enthalpy of the chamber, respectively; T_r, T_w and T_{water} are the temperature of refrigerant, plate wall and water (secondary fluid); $\dot{m}_e$ is the time derivative of refrigerant mass held in the chamber; E is the plate wall energy and $C_{th,w}$ denotes its thermal capacitance; A is the heat transfer area; α_r and α_{water} are the refrigerant side and water side heat transfer coefficient (HTC), respectively;

Based on the relationship between the refrigerant internal energy and enthalpy

$$u = h - \frac{P}{\rho} \quad (4)$$

where u is the refrigerant specific internal energy and ρ is the refrigerant density, the time derivative of the refrigerant internal energy U in Equation 1 can be decomposed into terms of time derivatives of the pressure and enthalpy using the chain rule:

$$\dot{U}=\left(\frac{\partial\rho}{\partial P}Vh-V\right)\dot{P}+\left(\frac{\partial\rho}{\partial h}Vh+\rho V\right)\dot{h}+\left(\rho h-P\right)\dot{V} \quad (5)$$

where V is the chamber volume of isothermal compressor. Similarly, the mass balance equation of Equation 2 can be rewritten as

$$\dot{m}_e=V\left(\frac{\partial\rho}{\partial P}\dot{P}+\frac{\partial\rho}{\partial h}\dot{h}\right)+\rho\dot{V} \quad (6)$$

Equation 5 and 6 allow the reformulation of the governing equations using the pressure P and enthalpy h as the state variables. The resultant state-space governing equations for each channel of PHX-based isothermal compressor are formulated as follows

$$\begin{bmatrix} V_i\frac{\partial\rho_i}{\partial P_i}h_i-V_i & V_i\frac{\partial\rho_i}{\partial h_i}h_i+V_i\rho_i & 0 & \rho_i h_i-P_i \\ V_i\frac{\partial\rho_i}{\partial P_i} & V_i\frac{\partial\rho_i}{\partial h_i} & 0 & \rho_i \\ 0 & 0 & C_{th,w} & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix}\begin{bmatrix}\dot{P}_i\\ \dot{h}_i\\ \dot{T}_{w,i}\\ \dot{V}_i\end{bmatrix}=\begin{bmatrix}\frac{\dot{m}_{in}}{N}h_{in}-\frac{\dot{m}_{out}}{N}h_i-\alpha_{r,i}A_i\left(T_{r,i}-T_{w,i}\right)\\ \frac{\dot{m}_{in}}{N}-\frac{\dot{m}_{out}}{N}\\ \alpha_{r,i}A_i\left(T_{r,i}-T_{w,i}\right)-\alpha_{water,i}A_i\left(T_{w,i}-T_{water,i}\right)\\ -\frac{V_m}{N}\end{bmatrix} \quad (7)$$

where i denotes the i^{th} channel; V_m is volumetric flow rate of the liquid piston and N is half of the total number of plates.

As shown in Figure 1, a flow meter was installed at the hydraulic part to measure the volumetric flow rate of oil, and the experimental results were shown in the left part of Figure 3. For modeling purposes, simplifications were thereby implemented: in each process (compression/suction), the volumetric flow rate was initially kept constant for the first 6 seconds, and then linearly declined throughout the remaining period. Note that the volumetric flow rate profile for each chamber takes mirror relationship.

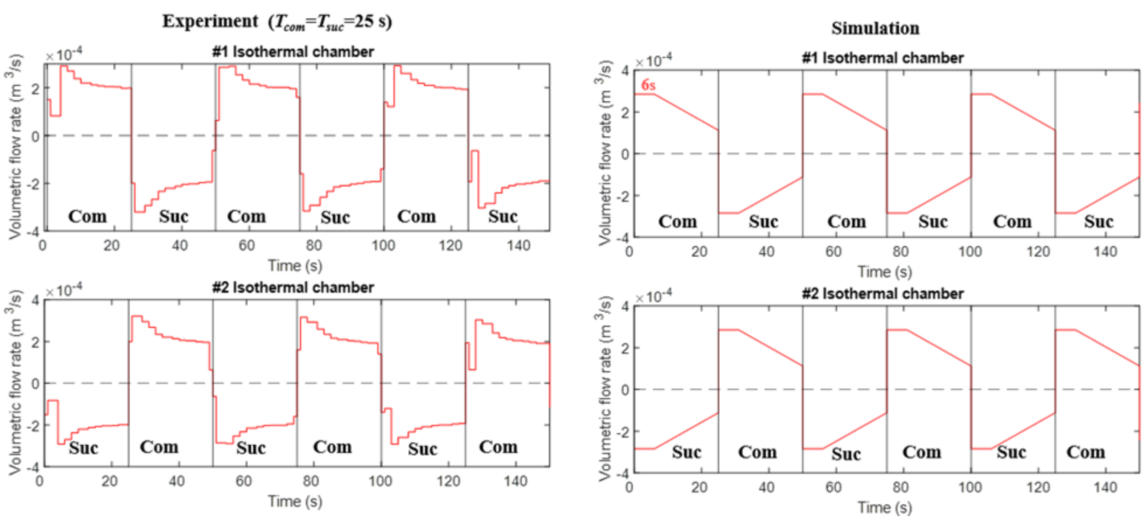

Figure 3. Volumetric flow rate of liquid piston for each compression chamber ("Com": compression, "Suc": suction).

3 Experimental Validation

Figure 4 shows the test rig of isothermal compressor system with the rated cooling capacity of 1 ton. As forementioned, the two PHXs-based compression chambers were utilized to implement the double-acting mechanism. This design ensures each of the compression chamber undergoes either the suction or compression process, with both processes having the same duration by using the level sensor to monitor the oil level. The secondary loop of PHX is managed by the water radiator with fan to release the heat into ambient. The oil separator is introduced to separate the oil dissolved in the refrigerant during the compression process and return it to the liquid pump. This pump provides high-pressure liquid to the compression chamber and is designed to achieve high volumetric efficiency.

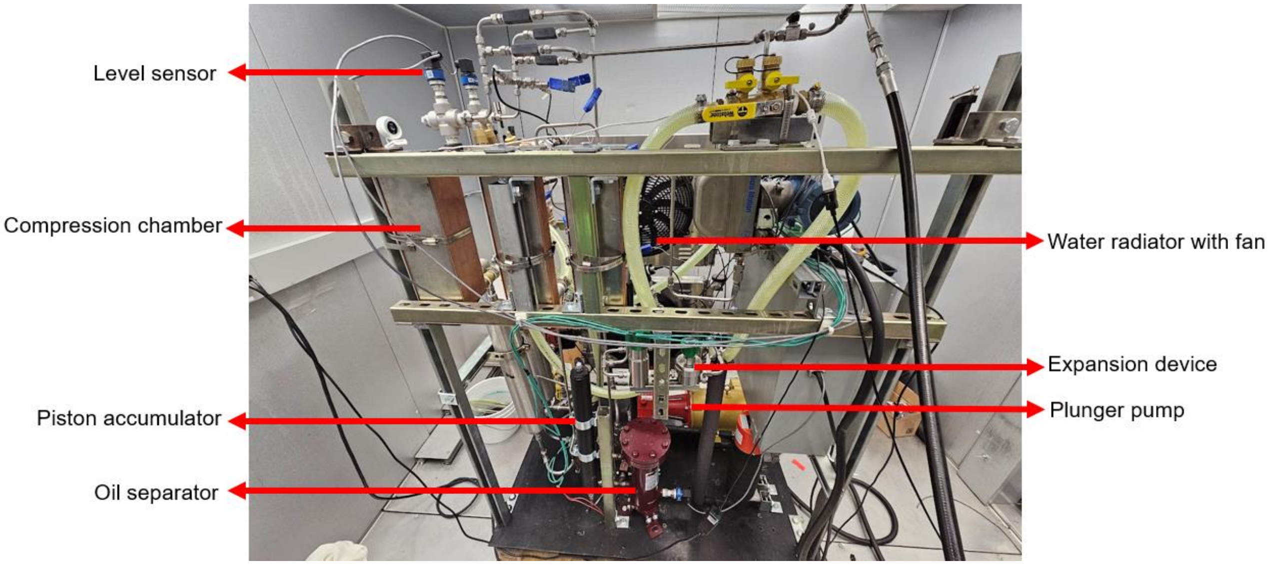

Figure 4. Experimental facility of isothermal compressor cycle.

The experimental validation results for the isothermal compression system model were depicted in Figure 5. Overall, the established model can accurately capture the pressure and temperature behavior of test rig, for both dynamic and steady state characteristics, with 7.5% relative error for the chamber pressure and 0.74 K deviation for the chamber temperature. However, the pressure comparison revealed that the simulation curve appears "steeper" than the experimental counterpart at the end of the compression process (delivery of CO_2), which probably due to the liquid piston model didn't accurately reflect the real volumetric flow rate at that stage. Therefore, one of the future tasks should be the calibration of liquid piston model. Furthermore, during the suction period, there is noticeable deviation between the simulation and experiment. This is mainly because the model didn't account for the degassing process, where quite amount of CO_2 is released from the oil due to the pressure drop.

As for the temperature comparison, the experimental results show a sudden drop at the beginning of the suction process. This drop is due to the expansion of the remaining CO_2 in the isothermal chamber. However, the model did not capture this behavior, likely because the sensor is attached to the surface of the compression chamber, and the model did not account for the thermal mass of the chamber wall.

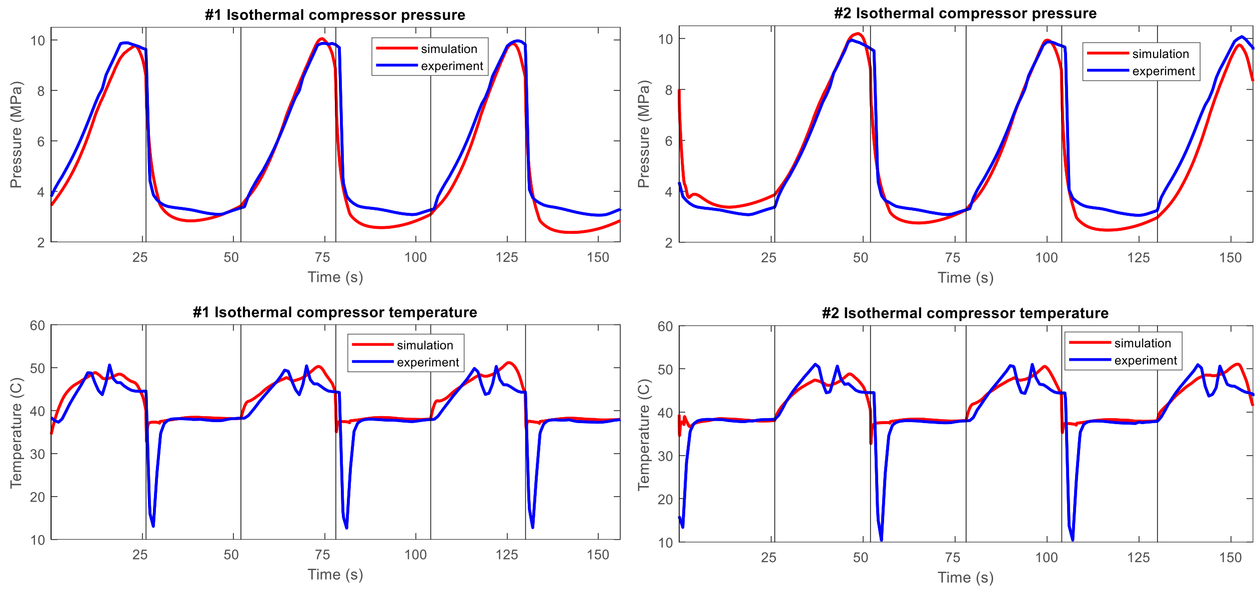

Figure 5. Comparisons of pressure and temperature between experiment and simulation for each compression chamber.

4 Conclusions

In this paper, a dynamic model for the double-acting isothermal compressor based on the CML is established. The isothermal compressor is coupled with liquid piston to serve the dual purpose as a heat exchanger to cool down the refrigerant and achieve the isothermal compression within the chamber. Unlike the conventional compressor model which evolve on much faster time scales than the heat exchanger dynamics and typically established as quasi-steady state model, the compression process for the isothermal compressor model requires much longer time to dissipate the sufficient heat. To demonstrate the accuracy, the experimental tests were carried out based on the 1-ton refrigeration system, where the boundary conditions of experimental data were fed into the model. The results validated that the established model can accurately capture both the steady-state and dynamic behaviors of test rig (7.5% relative error for chamber pressure and 0.74 K deviation for chamber temperature).

To further improve the model accuracy, future work include the calibration of liquid piston model and the incorporation of the CO_2 solubility correlation in the oil. The developed model will be used to guide the future design (e.g., shape optimization of chamber) and prototyping.

Acknowledgements

This material is based upon work supported by the U.S. Department of Energy's Office of Energy Efficiency and Renewable Energy (EERE) under the Building Technologies Office Award Number DE-EE0008674.

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