ORNL/TM- 2023/3042

Building Technologies Office 03.02.02.38 Milestone Report—Compressor Technology Development for Refrigerants with <150 Global Warming Potential

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June 2023

ORNL/TM-2023/3042

Buildings and Transportation Science Division

BUILDING TECHNOLOGIES OFFICE 03.02.02.38 MILESTONE REPORT—COMPRESSOR TECHNOLOGY DEVELOPMENT FOR REFRIGERANTS WITH <150 GLOBAL WARMING POTENTIAL

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managed by
UT-BATTELLE, LLC
for the
US DEPARTMENT OF ENERGY
under contract DE-AC05-00OR22725

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1. PURPOSE

The major purpose of this project is to enable the development of compressors specifically designed for compatibility with low–global warming potential (GWP) refrigerants (with GWP values below 150). These compressors are intended for applications in residential and light commercial air-conditioning and heat pump systems.

This research consists of several discrete stages. Initially, the team undertook an exhaustive review of the existing literature with the intention of discerning the most efficacious methodologies currently in use for compressor modeling. This fundamental stage was designed to foster a comprehensive understanding of the intricate dynamics and mechanics of compressors.

After this review, the project transitioned to the complex modeling of a 3 ton air-conditioning system, followed by preliminary modeling of compressors, using the knowledge gleaned from the literature review. Future stages of this research include comprehensive modeling of compressors and robust performance validation. This validation process involves rigorous laboratory testing aimed at affirming the accuracy of these models.

So far, the team has concluded an extensive literature review and established the groundwork for compressor modeling. This report demonstrates the findings from the literature review, a detailed description of the model, and the preliminary validation results.

2. LITERATURE REVIEW

Compressor modeling is commonly classified into three primary categories by researchers, each distinguished by the degree of detail or knowledge required for model development [1]. *Black-box* models require the least amount of detailed knowledge, whereas *white-box* models necessitate the highest level of understanding. All models that occupy the spectrum between these two extremes are referred to as *gray-box* models.

2.1 BLACK-BOX MODELS

Black-box models are empirical models relying on correlations derived from statistical data. Often employed as submodels in global thermodynamic system simulations, these macromodels use their empirical underpinnings to accurately predict several parameters, including capacity, mass flow rate, compressor power, discharge temperature, isentropic efficiency, and system coefficient of performance.

A classic example of a black-box model is the statistical model for the 10-coefficient compressor map as defined by the Air-Conditioning, Heating, and Refrigeration Institute 540 standard. Renowned for their reliability in interpolation, these models are frequently provided by compressor manufacturers with their products.

Despite their robustness, black-box models have limitations, particularly when extrapolating data. The uncertainty in extrapolation is high because of the model's inherent design, which lacks explicit information about the internal processes and their interrelations. This absence of specific knowledge introduces potential inaccuracies, thereby emphasizing the need for caution when using black-box models beyond their tested and validated ranges.

2.2 WHITE BOX MODELS

The compression process within a compressor is transient; it is marked by substantial fluctuations in volume, pressure, and temperature within a single compression cycle. To accurately represent these detailed changes throughout this transient process, fully distributed models must be used.

White-box models are fully distributed models that depict physical processes through the solution of partial differential equations. These models include computational fluid dynamics (CFD) models and finite element models, which are based on rigorous mathematical and physical principles. Notably, Stosic et al. [2] pioneered the development of a CFD model for screw and scroll compressors. This model, primarily focused on analyzing the pressure history within the compression chamber, employs unsteady, turbulent, and compressible fluid flow dynamics. However, this model does not consider heat transfer, thereby restricting its scope to the compression process alone.

A 2D unsteady CFD model was developed by Ooi et al. [3] in 2004, which studied the flow field and heat transfer. This paper also compared the numerical results with several heat transfer empirical correlations. However, the effect of leakage flow and lubrication on temperature was not considered.

Cui [4] developed a 3D CFD model for the entire refrigerant path for scroll compressors, including the suction port, discharge port, and scrolls. An analysis oriented to the evaluation of the pressure distribution in the pockets and in the leakages through the flank gap was performed through their model.

Zhang et al.[5] presented a 3D transient CFD model with dynamic meshing that considered radial clearance and axial clearance for scroll hydrogen pumps. The presented modeling and simulation methods were validated experimentally by using a scroll air compressor. The results showed that axial clearance has a more significant influence than radial clearance on the performance of scroll-type hydrogen pumps.

Sun et al.[6] established a 3D unsteady CFD model to numerically investigate the flow field and the performance of six modification schemes of scroll refrigeration compressors.

Kin and Shim[7] investigated the hydrodynamic characteristics of thrust bearing, which included lubrication film, temperature distribution, and thermal stress. The maximum deviation between experimental data and simulation results was 5%, which demonstrated the effectiveness of the simulation model.

Song et al.[8] concluded that the application of CFD for scroll compressors cannot be widely adopted owing to following reasons: (1) 2D models cannot reflect the accurate geometry and variation of the flow field, (2) the quasisteady simulation may not be reliable, and (3) the single-phase flow model does not reflect the effect of the lubricated oil. This conclusion is also applicable to modeling other positive displacement compressors because the modeling process is very similar between different compressors.

2.3 GRAY-BOX MODELS

Mechanistic is the most common type of gray-box models for compressor modeling. This type is based on the thermodynamic process of compression. It simulates the compression process using mass and energy conservation equations inside the compression chambers.

Different physical processes occur in the operation of compressor, including vapor compression, leaking through gaps, heat transfer in the suction path and compression chamber, friction on bearings and rolling parts, and loss through the discharge valve.

To study the performance of a compressor through numerical methods, the mechanistic model usually consists of different submodels, such as the geometrical model, mass flow and leak model, discharge valve model, and overall energy balance model. The pressures, temperatures, and volume flow rate in the compressor chambers can be derived by combining these submodels.

The earliest concept of a scroll compressor was presented by Creux in 1905 [9]. This compressor is defined as an involute of a circle. However, the manufacturing technology at that time prevented the development of scroll compressors that could compete with reciprocating compressors. In the 1970s, the scroll compressor came back to people’s focus because of rising energy costs and much more advanced manufacturing technology compared with that of the 1900s. Many new scroll compressor products were released in the 1980s.

In 1984, Morishina et al. [10] adopted the concept of the involute of a circle in their model. The displacement volume and pressure ration, radial, and tangential forces were calculated in the analytical model. Morishina et al. also developed the motion equation of the orbiting scroll and Oldham coupling. However, the chamber volume is only a function of the orbiting angle up to 2π, which is insufficient to depict the compression from suction to discharge. The authors also presented the earliest model for mechanical loss. This model is calculated by summing the losses generated by each point of contact between the Oldham ring and orbiting scroll. This concept was also adopted by some other researchers [11-13]. Ishii suggested the mechanical efficiency value in different papers: 80% in [14] and 92.5% in [11]. Part et al. [15] fitted the empirical formulas of mechanical and motor efficiencies as a function of compressor motor frequency.

Based on Morishina’s work [10], Yanagisawa [16] developed a full scroll compressor geometrical model in 1990, which expresses every chamber volume as a function of orbiting angle by a general formula. This improvement makes the development of a thermodynamic model much easier. The team also investigated the optimum operating pressure ratio that maximizes the compression efficiency by changing clearances and speed.

In 1992, Margolis et al. [17] developed the typical baseline model for leakage in scroll compressors, which is compressible gas through an isentropic nozzle. Because friction is present in the flow paths, the friction is considered by introducing empirical factors to compensate for the pressure losses. Many other papers adopted the same method to describe the leakages, such as [18, 19].

Cho et al. [20] found that a correction factor of 0.1 for the discharge flow can fit the flow data well, which means that the isentropic flow model is far from accurate. The major problem of this model is that friction loss is not considered. Ishii et al. [21] suggested using incompressible flow through leakage gaps in the model. Compared with the compressible flow modeled in the same paper, the incompressible flow modeling result can match well with experiments. In both papers, a constant value of gap sizes was used for the radial and flank gaps. Most models neglect the oil effect for simplification. However, for oil-injected scroll compressors, the oil effect should be considered in the model. Li and Wang [22] developed a separated oil–gas leakage flow model assuming that the flow through the gaps is laminar; a critical clearance gap was determined using the oil and gas flow rates. However, this model is not suitable for turbulence flow. Bell et al. [23] treated the flow as homogenous oil–gas leakage flow with a corrected isentropic nozzle.

Halm introduced the overall energy modeling of scroll compressors in 1997 [24]. He developed energy conservation equations for the refrigerant in each of the compressor chambers and the overall energy balance equations for the various compressor elements using a lumped-element method. By solving the energy conservation equations and the coupled overall energy balance equations, mass flow rate, power consumption, and discharge temperature were predicted.

Yanagisawa's model was further improved by Chen et al. [19], who developed the mathematical models using different initial angles of the involute. For modeling leakage, they adopted the same concept as Margolis [17]. The gap size was defined as a function of the speed and compression ratio. An experimental study was also conducted to support the modeling result. In the second part of this study [25], Chen et al. presented the overall energy balance model inside the compressor. The model used the same concept as Halm's thesis [24]. The components of the compressor were split up into different *lumped capacitance* elements, which are associated with a *lump temperature*. The energy balance was analyzed in an equivalent electrical circuit for thermal masses and their corresponding resistances. From this model, Chen et al. found that the scroll wrap temperature is almost linear with the involute angle. A discharge bypass load test stand was built to validate the accuracy of the models. The experimental results indicated that their model could predict mass flow rate, power input, and discharge temperature very well.

In 2005, Baolong Wang et al. [26] noted that one of the major shortcomings of Chen's geometric model [19] is that it is based on a particular initial involute angle, which is not the case in real applications. Then, Chen presented a generalized approach of a geometric model for scroll compressors. This model can take an arbitrary initial involute angle. Therefore, the model resolved the difficulty of practical application. Furthermore, Chen used the geometric model and empirical data of motor and mechanical efficiency to calculate the temperature, pressure, and mass flow in the compressor. The simulation results based on this model were compared with the experimental result done by Winandy [27]. The relative error of the mass flow rate was within 2.5%, and the error for the motor power was within 5.0%. Wang further developed this model to incorporate vapor/liquid injection in the scroll compressors [28]. The governing equations for vapor and liquid injections were established. This paper also noted that the flow rate coefficient, C_d, used in the flow model can vary a lot from suction to discharge. So, C_d is usually set based on experiments, and different researchers have different results. The modeling result was validated in another paper [29]. The relative errors for mass flow rate, injected mass flow rate, and discharge temperature are all within 2%–4%.

In 2011, Bell [30] developed a comprehensive geometric model based Baolong Wang et al.'s [26] improvement that is fully adaptable to arbitrary initial angles. In this dissertation, he extended the model of Chen and Halm [19] by incorporating oil injection in the compression process. The major modeling result of this dissertation is published in a journal paper [31]. This approach allows the optimization of oil flooding scroll compressors without costly and time-consuming experiments.

Beside the framework developed by Morishina [10], Yanagisawa [16], Chen [19], and Wang [26], another framework for scroll wrap analysis was developed by Bush et al. [32], which derives the curves by enforcing conjugacy in the most general way. This approach was used by Gravesen [33] in the study of variable wall thickness scroll wraps. Furthermore, Blunier et al. [34] derived an analytical solution for constant wall thickness scroll based on Gravesen's model.

All the studies discussed in this section are about scrolls with an involute of circle profile. Other than the involute of a circle, other profiles were also studied, such as the involute of squares [35] and spiral curves [36]. However, because of the complexity of manufacturing, only involutes of circles are widely used in real applications.

With continuous efforts on compressor modeling [19] [24] [25] [30], PDSim, which is a novel generalized compressor simulation package, was developed by the Ray W. Herrick Lab at Purdue University [37]. PDSim is the first open-source generalized positive displacement compressor and expander simulation package. It can be used for modeling most positive displacement compressors, including reciprocating, scroll, rotary, linear, Z, and spool compressors. Tanveer et al. [38] compared PDSim with MATLAB and Modelica by modeling the same compressor using all of these tools. The results indicate that PDSim is potentially a better platform for compressor optimization on a qualitive basis.

In summary, Table 1 presents a comprehensive comparison of various types of models. Based on the information provided, it can be inferred that the mechanistic model, which falls under the category of gray-box models, is the optimal approach for this research project. This choice is justified by its ability to strike a balance between model complexity, accuracy, computational efficiency, and overall adaptability.

Table 1. Comparison of different models for compressor study

Type	White-box	Gray-box (mechanistic)	Black-box (performance)
Definition	Physics-based Fully distributed model (partial differential equations)	Construction-oriented Qualitative knowledge (rules) Simplified relationships	Extrapolation based on experimental data
Technique	CFD simulation Finite element analysis simulation	Mechanistic modeling Lumped parameter model	Compressor maps using curve-fitting Semiempirical based on isentropic and volumetric efficiency curves
Features	Good description of geometry Detailed physical process Good accuracy after calibration	Covers a wide range of compressors 95% faster than white-box Good accuracy after calibration	Short development time Little domain expertise required
Drawbacks	Time consuming development (meshing and computation)	In-depth knowledge of compressor physics required	Not reliable for extrapolation out of experimental range
Application areas	Optimizing individual components by investigating dynamic behavior	Design based on variables: speed, capacity, heat transfer, internal leaks, fluid properties	Performance prediction within the range of test data System design

3. MODEL DESCRIPTION AND VALIDATION

The Herrick Lab at Purdue University, who is a partner of this project, has been developing a generalized simulation framework for positive displacement machines over the last few years. An open-source modeling framework named PDSim was created for generalized positive displacement machine simulation. In the project, the PDSim was validated for modeling scroll compressors, rolling piston compressors, and linear compressors. The PDSim model setup with key assumptions, features, and equations are shown in Section 3.1. The applications for using PDSim in a scroll compressor, rolling piston compressor, and linear compressor, as well as the model validation, are described in Sections 3.2, 3.3, and 3.4, respectively.

3.1 MODEL DESCRIPTION

The motivation for the development of PDSim was that positive displacement compressors and expanders have significant overlap in their construction. Therefore, there is utility in a tool that models the common features shared amongst all types. These uses include the following:

- Control volumes (CVs), which form the heart of the device and the model. These CVs (working chambers) have volume profiles governed by the geometry. The change in the volume of the working chambers results in compression (with power input) or expansion (with power recovery). The CVs exchange heat and mass with the elements with which they communicate, as shown in Figure 1.

- Flow models, used to connect tubes, chambers, and other components in the machines. As the complexity of the machine increases, the permutations of the chambers that may communicate through flow increase significantly.

- Electric motor losses and heat transfer. The machine itself is usually driven with an electric motor (alternatively, the machine drives a generator in the case of an expander), which may in turn be cooled by suction or discharge gas.

- Mechanical losses, and the commensurate heat transfer, that arise. These losses include friction generated by bearings, seals, and other relative motion between parts.

- The numerical challenges that typically arise in simulating these devices (e.g., numerical stiffness, step size selection) are common to all types of positive displacement machines, as are methods to deal with these challenges.

Following the compression process in a compressor, the mechanistic model of a generic positive displacement (PD) machine can be divided into a certain number of blocks, for which the following specific submodels are required:

- Definitions of CVs (static and dynamic) and their geometries in terms of $V(\theta)$ and $dV(\theta)$ or $V(t)$ and $dV(t)$

- Leakage paths and their flow models

- In-chamber heat transfer model

- Motor losses in semihermetic and hermetic machines

- Shell thermal network and overall energy balance

- Thermophysical properties and multilayer solution scheme

Compression processes in compressors are transient thermodynamic transformations. Therefore, to be studied, a system of transient conservation equations is necessary. Each working chamber is described thermodynamically by means of open CVs. In each CV, as shown in Figure 1, uniformity is assumed in the thermodynamic properties, which also implies quasiequilibrium conditions. Such an approach has been proven to be accurate enough to predict the performance of PD machines. Nonuniformity of the thermodynamic properties within the chamber volume can be accounted for by means of CFD methods. The following assumptions are applied to a general CV:

- Uniform temperature and pressure throughout the CV
- Negligible kinetic energy of the CV
- Negligible gravitational effects
- Thermal interaction possible through heat transfer
- Mass flow into or out of the CV

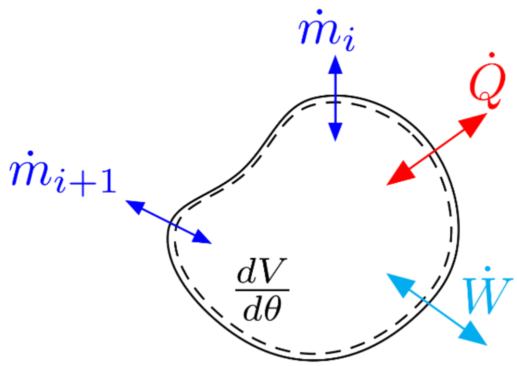

Figure 1. Schematic of a compressor CV at crank angle of θ [39].

The conservation of mass for a generic CV can be written as Equation (1):

$$\frac{dm_{\text{CV}}}{dt} = \sum_i \dot{m}_i \, , \tag{1}$$

where m_{CV} is the mass of the working fluid within the chamber (or CV), and $\dot{m}_i$ represents the mass flow rate of the *i*th flow path entering or exiting the CV (positive if it flows into the CV). The flow interactions are usually referred to as leakage flows.

The conservation of total energy reduces to the conservation of internal energy because of the assumptions made, and it is given by Equation (2):

$$\frac{d(mu)_{\text{CV}}}{dt} = \dot{W}_{\text{CV}} + \dot{Q}_{\text{CV}} \sum_i (\dot{m}h)_i \, , \tag{2}$$

where the first two terms on the right hand of Equation (2) refer to the boundary work rate of the CV and heat transfer rate of the CV, respectively. The term $(\dot{m}h)_i$ is the enthalpy flow term for the *i*th flow path. In the case of PD machines, the boundary work rate term is usually expressed as Equation (3):

$$\dot{W}_{\text{CV}} = -p_{\text{CV}} \frac{dV}{dt} , \tag{3}$$

The system of equations developed in this section describes a dynamic model of a PD machine that needs to be integrated over time. Such a formulation is general enough that can be applied to PD machines that are crank-motion driven (e.g., scroll and rolling piston) and dynamic linear compressors [40] in which the stroke is determined by electromagnetic and gas pressure forces. As the integration over time progresses, the system will reach a steady periodic solution. In the case of PD machines having a crank motion, the steady periodic solution is usually of particular interest (unless transients such as start-up or shutdown are considered). The steady periodic solution corresponds to a complete working cycle of the PD machine that is typically identified as a function of the rotation angle. For this reason, the system of differential equations is conveniently expressed as function of the crank angle by recalling that $\theta = \omega t$. The conservation of mass for the CV yields the following form in Equation (4) for the derivative of the mass with respect to the crank angle in the CV:

$$\frac{dm_{\mathrm{CV}}}{d\theta} = \frac{1}{\omega}\sum_{i} \dot{m}_i \tag{4}$$

The thermodynamic state of the CV can be calculated by knowing two independent thermodynamic properties for pure fluids and fluid mixtures that are treated like pure fluids (see Lemmon [41]). The CoolProp property library [42] is used to retrieve the necessary thermodynamic and transport properties. The Reference Fluid Thermodynamic and Transport Properties Database (REFPROP) thermophysical property library [30] can be used as the thermophysical property back end through CoolProp. Because the equations of state used in CoolProp and REFPROP are all of the Helmholtz energy explicit formulation and have as independent variables temperature and density, the two independent state variables used to derive the differential equations are chosen to be temperature and density.

The conservation of energy in the CV is expressed in terms of the time derivative of the temperature as Equation (5):

$$\frac{dT}{dt} = \frac{-T\left(\frac{\partial p}{\partial T}\right)_{\rho}\left[\frac{dV}{dt} - \frac{1}{\rho}\frac{dm}{dt}\right] - h\frac{dm}{dt} + \frac{\dot{Q}}{\omega} + \frac{1}{\omega}\sum_i (\dot{m}h)_i}{mc_v} \tag{5}$$

If the time is associated with the crank angle in rotating machines, such as scroll and rolling piston compressors, the conservation of energy in the CV can then be expressed in terms of the crank angle derivative of the temperature, as shown in Equation (6):

$$\frac{dT}{d\theta} = \frac{-T\left(\frac{\partial p}{\partial T}\right)_{\rho}\left[\frac{dV}{d\theta} - \frac{1}{\rho}\frac{dm}{d\theta}\right] - h\frac{dm}{d\theta} + \frac{\dot{Q}}{\omega} + \frac{1}{\omega}\sum_i (\dot{m}h)_i}{m_{\mathrm{CV}}}. \tag{6}$$

The conversion from dU/dt to dT/dt is carried out because all the terms in Equation (6) are explicit in temperature and density, which are the state variables upon which the equation of state is based.

By introducing the geometry parameters and other particular modeling aspects (e.g., leakage flows, heat transfer correlations, friction losses), the same structure can be applied to any PD machine. In particular, the block diagram in Figure 2 illustrates the general structure of a PD machine model with the required submodels. Such a workflow was applied in this research to implement scroll and rolling piston compressors. The linear compressor workflow is introduced separately in Section 4.

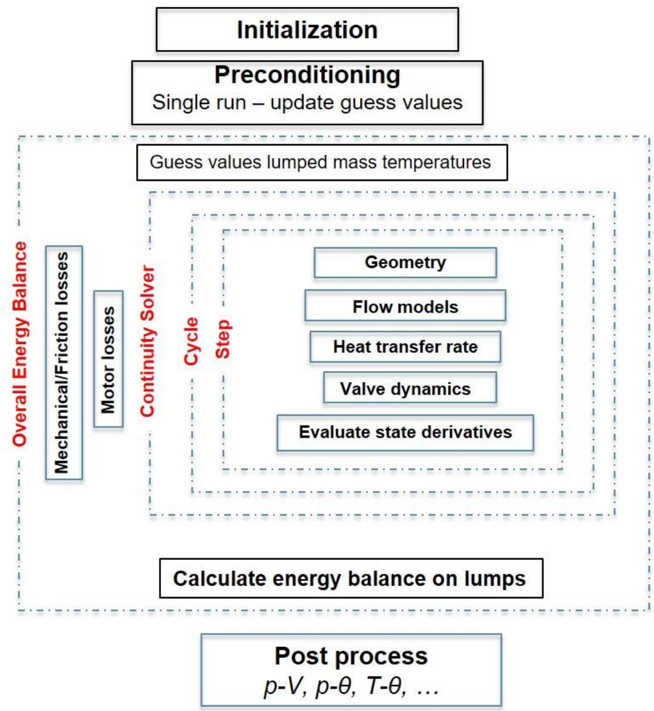

Figure 2. General workflow of a compressor/expander model.

The PD compressor simulation model, called PDSim, has been developed in the Python programming language with support for both 2.x and 3.x distributions. The adoption of Python is justified by the open-source and cross-platform nature of the programming language, the numerous packages available, and the active community. PDSim takes advantage of the high-level features of Python, as well as the low-level code through *Cython*, to achieve higher computational efficiency. PDSim is coded in an object-oriented fashion that ensures a plug-and-play structure that simplifies the construction of a model and facilitates the extension of the existing libraries.

Each family of PD machines represents a class in PDSim that can be used to actually build a specific compressor or expander model. The general structure of a class of a simulation model in PDSim is shown in Figure 3. It follows the typical work process of building a mechanistic model of a PD machine. For a given machine type, the corresponding core class is derived from the main *PDSimCore* class (located *PDSim/core/core.py*) and inherits a number of methods that allow building the actual model to be run with minimal coding required. These methods include the following:

- *add_CV( )*: add the control volumes
- *add_flow( )*: add the flow paths (inlet and outlet flows, suction and discharge flows, leakage flows)
- *add_tube( )*: add inlet and outlet tubes of the compressor/expander, as well as possible injection lines
- *add_valves( )*: add valves, if any

The pseudocode of a simple compressor model in PDSim is reported in Figure 4. PDSim has been used to model the performance of a variety of positive displacement compressors. Examples are shown in the next couple of sections.

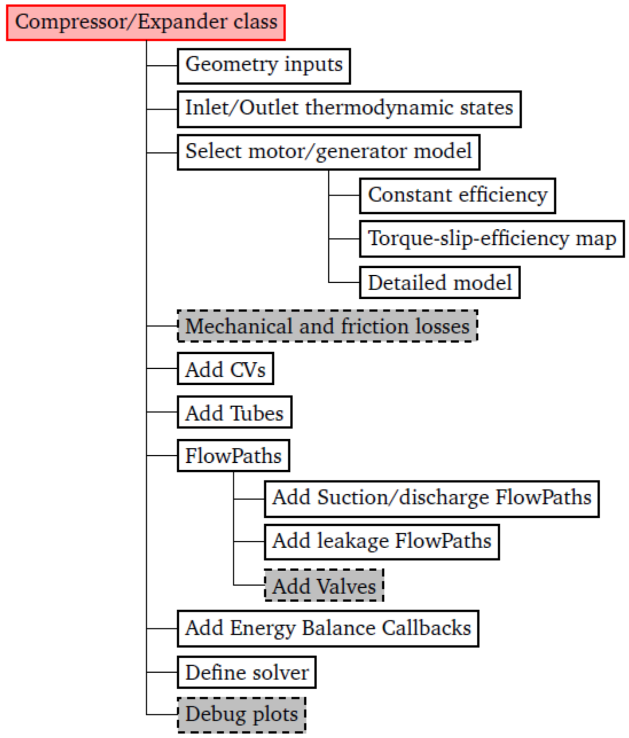

Figure 3. General class of a compressor/expander model in PDSim. Dashed boxes represent optional elements of a compressor/expander simulation model.

Algorithm 1 Compressor pseudo-code

```
# Declare Python and PDSim imports
# Define compressor function
function Compressor(**kwargs):
    # Instantiate the model
    comp = CompressorClass()
    # Define the inputs
    # For example: comp.Vdead = 0.5e-5
    # Define boundary conditions
    Ref='Air'
    inletState=State.State(Ref,'T' : 298.15,' P' : 101.325)
    outletState=State.State(Ref,)
    mdot_guess =
    inletState.rho*comp.Vdisp*comp.omega/(2*pi)
    # Add control volume(s)
    comp.add_CV(ControlVolume(key,inletState
    VdVFcn, becomes))
    # Add inlet tube
    comp.add_tube(Tube(key1='inlet.1',key2='inlet.2',
    L,D,mdot=mdot_guess,State1=inletState.copy()
    fixed=1,TubeFcn=comp.TubeCode))
    # Add outlet tube
    comp.add_tube(Tube(key1='outlet.1',key2='outlet.2',
    L,D, mdot=mdot_guess,State2=outletState.copy()
    fixed=2,TubeFcn=comp.TubeCode))
    # Add flow paths to link flow nodes
    comp.add_flow(FlowPath(key1,key2,MdotFcn))
    comp.add_flow(FlowPath(key1,key2,MdotFcn))
    # Connect all the callbacks
    comp.connect_callbacks(
    endcyle_callback=comp.endcycle_callback,
    heat_transfer_callback=comp.heat_transfer_callback,
    lump_energy_callback=comp.lump_energy_callback
    )
    # Time the simulation
    t1=clock()
    # Define the solver
    comp.solve(key_inlet=='inlet.1',
    key_inlet=='outlet.2',
    solver_method==kwargs.get('solver_method','Euler'),
    OneCycle = False,
    UseNR = False
    )
    print('time taken',clock()-t1,'s')
    from PDSim.plot.plots import debug_plots
    debug_plots(comp)
if __name__ == '__main__': then
    # Run the model
    Compressor (solver_method = 'Euler')
```

Figure 4. Pseudocode of a simple compressor model in PDSim.

3.2 MODEL VALIDATION

3.2.1 Scroll Compressor

Scroll compressors consist of a fixed scroll, orbiting scroll, Oldham's ring, frame, and crankshaft. In the fixed scroll, the crankshaft drives the orbiting scroll to compress the refrigerant through an orbital motion. Figure 5 shows a structure schematic of a scroll compressor from [43]. Scroll compressors are characterized by a fixed wrap and an orbiting wrap with an involute whose geometry is based on the form of an involute unwrapping from a circle. Because of the orbiting movement, the definitions of the compressor CVs change continuously throughout the compression process. The mathematical formulation of the scroll geometry is quite involved and requires the calculation of the following quantities [39]:

- Volumes of all chambers as a function of the crank angle
- Derivatives of the volumes of all chambers with respect to the crank angle
- Centroid of each chamber to evaluate the overturning moments owing to gas forces
- Force components on the orbiting scroll owing to pressure forces
- Radial leakage areas between chambers

Previous work on scroll compressor geometry with a displacement of 83 cm^3/revolution was readily available from [2]. The geometry of the compressor and the evolution of the working chamber volumes with respect to the crank angle are shown in Figure 6.

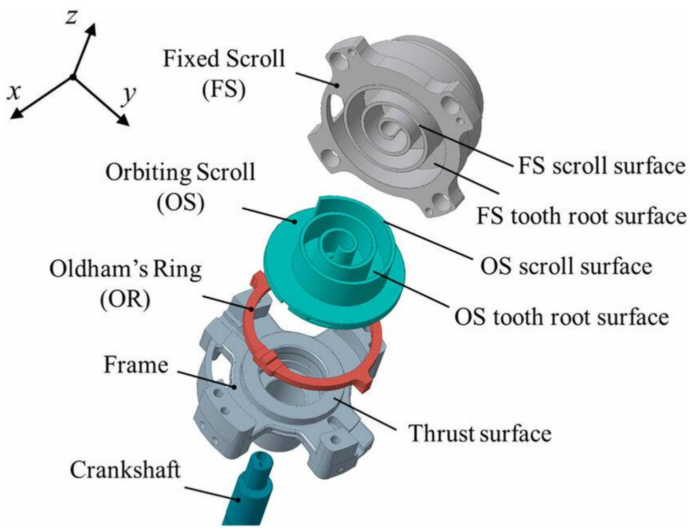

Figure 5. Structure schematic of scroll compressor.

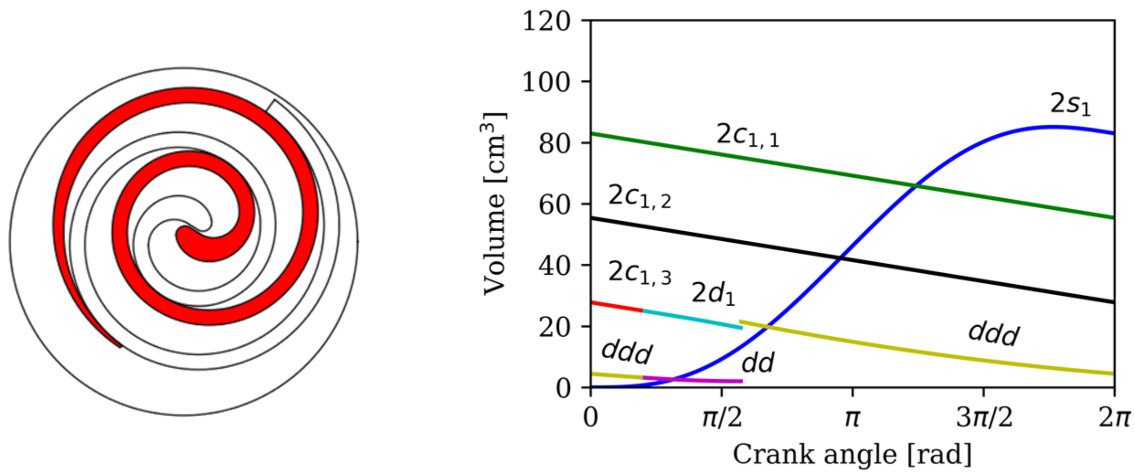

Figure 6. Heat treatment cycle in 75°F ambient.

Because the model approach has been identified, the model was validated using the available experimental data for a scroll compressor using R134a as refrigerants. The geometry model was updated to match the provided compressor displacement. The validation results are shown in Figure 7.

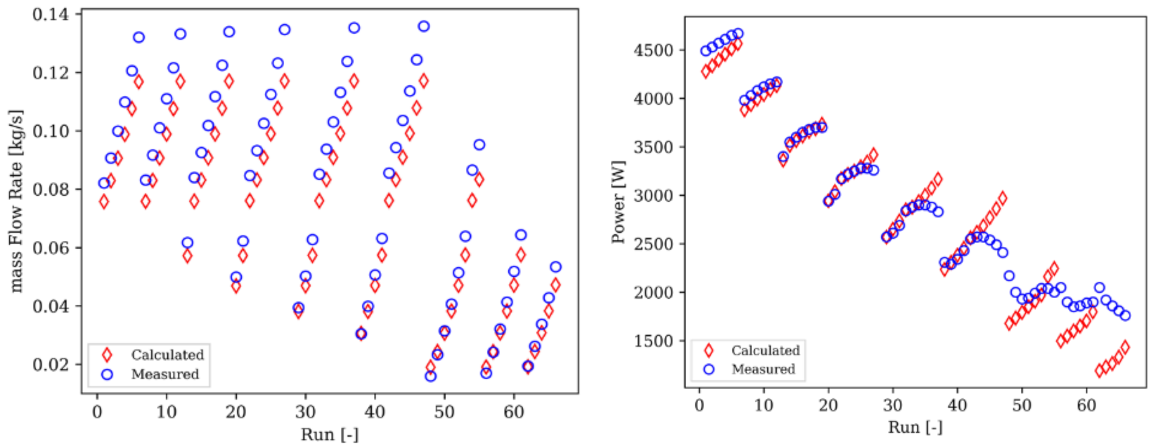

Figure 7. Scroll compressor model validation for (left) mass flow rate and (right) input power without tuning [8].

3.2.2 Rolling Piston Compressor

The rolling piston is used to compress low-pressure and low-temperature refrigerant vapor to high-pressure and high-temperature vapor. The shaft and the piston (i.e., the sliding parts) are lubricated using a lubricant that is pumped into the cylinder. Figure 8 shows a schematic of a rolling piston compressor.

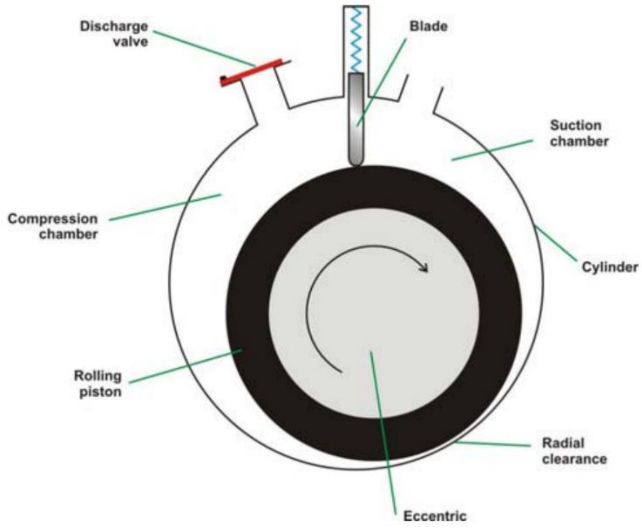

Figure 8. Schematic figure of a rolling piston compressor [45].

Using the PDSim structure, a rolling piston compressor model was developed and validated against the experimental data in Engel and Deschamps [46], where experimental data and geometric information were provided. The data validation is shown in Figure 9, where six different operating conditions were predicted and compared with experimental measurements. The model predicted performance within 5% error for most of the quantities, with a slightly higher error for the mass flow rate.

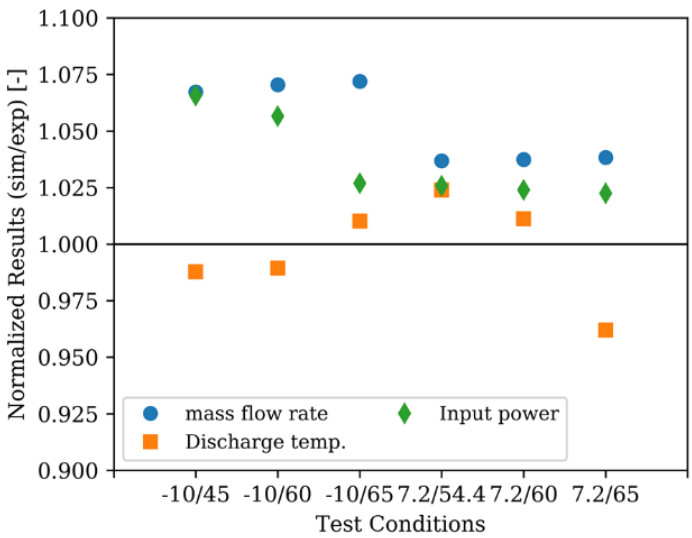

Figure 9. Rolling piston compressor modeling validation with R134a with normalized data in six different testing conditions shown in the *x*-axis [44].

3.2.3 Linear Compressor

Linear compressors have higher efficiency, oil-free operation, a lower cost, and a smaller size. In linear compressors, the piston is driven by a linear motor instead of a crank shaft, which allows oil-free operations that increase the choices of refrigerants and their operating temperature ranges. Figure 10 shows a schematic of a linear compressor.

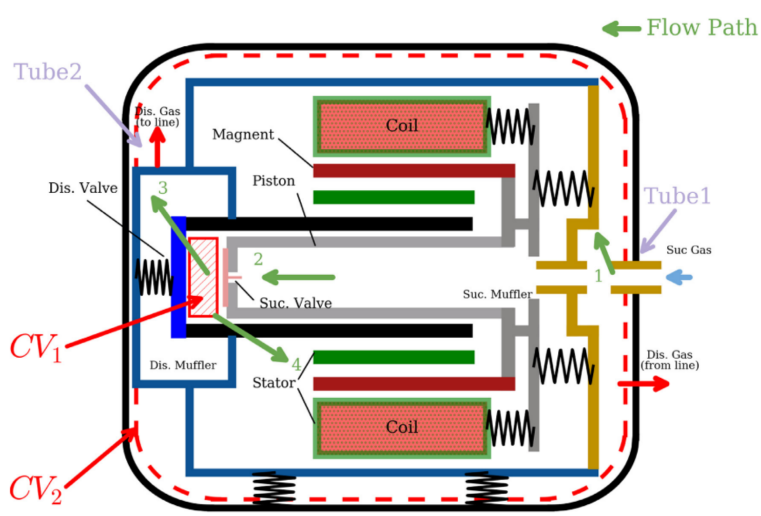

Figure 10. Schematic figure of a commercial linear compressor [40].

In contrast with the scroll and rolling piston compressors that are crank angle–based, a linear compressor has a linear motion without a crank angle involved. Thus, the linear compressor model developed using PDSim was modified to be solved with respect to time. The updated flow chart to simulate a linear compressor using time as a variable is shown in Figure 11.

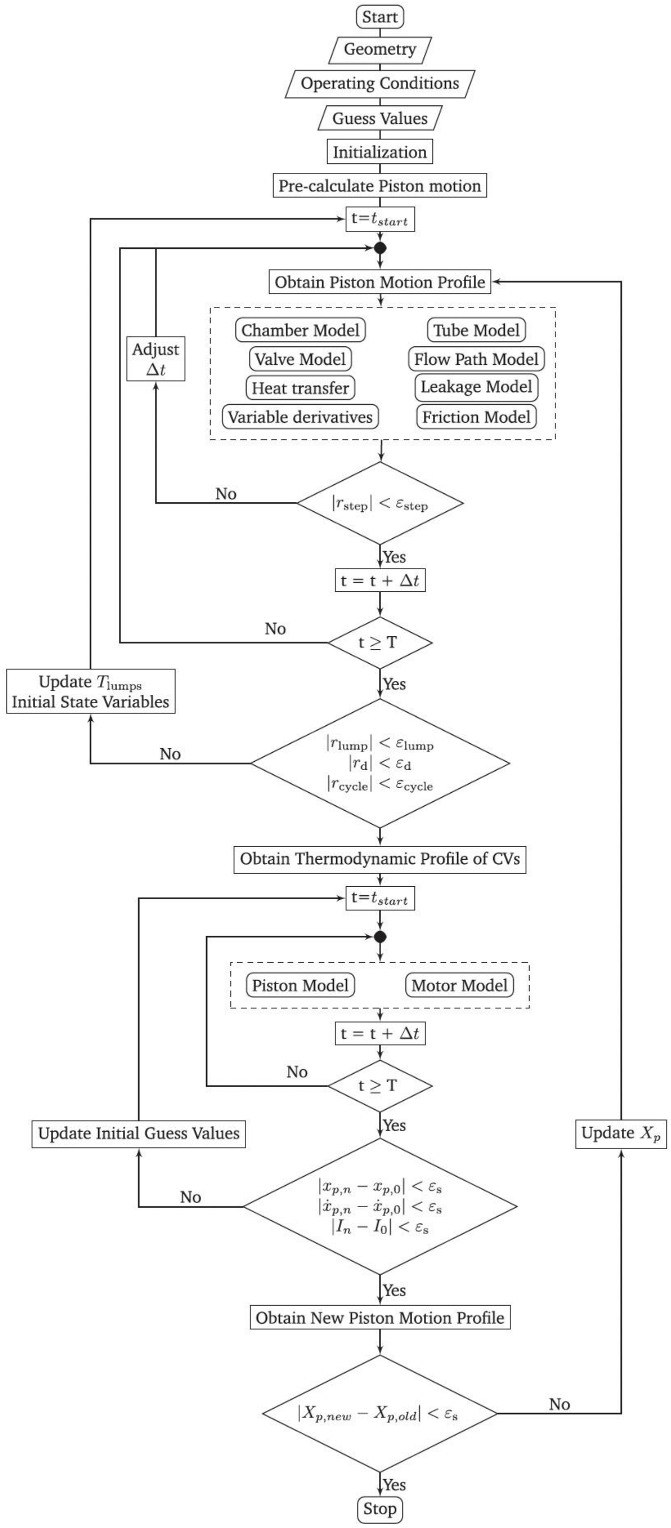

Figure 11. Flow chart of the periodical steady-state linear compressor simulation model [40].

Following the identified modeling approach, the linear compressor model was developed and validated against experimental measurements, as presented in this report. The geometric information, experimental test setup, and the testing conditions are detailed in Zhang et al. [47]. The validation results are shown in Figure 12 with two different compressors: A and B.

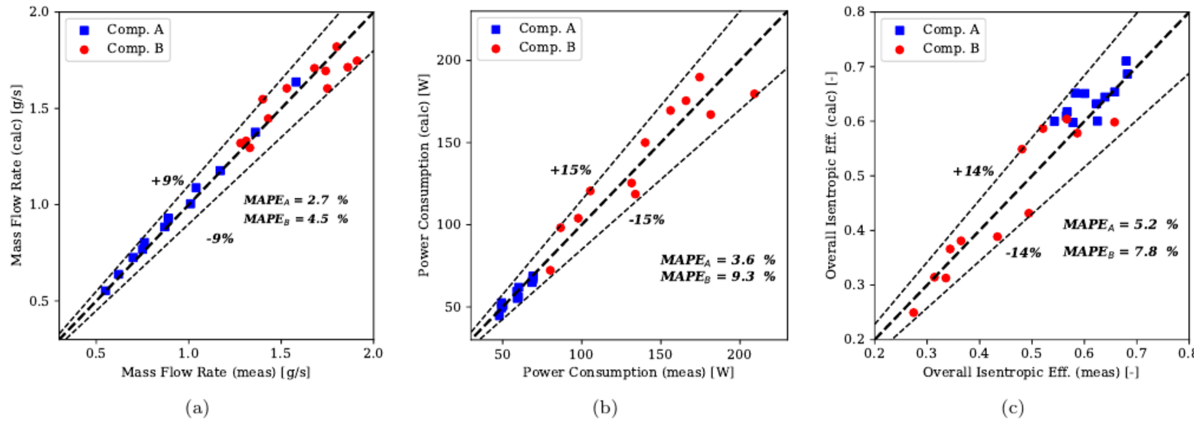

Figure 12. Flow chart of the periodical steady-state linear compressor simulation model [47].

4. SUMMARY

In a comprehensive review of relevant literature, several methodologies for compressor modeling were evaluated, including black-box, white-box, and gray-box models. A subtype of gray-box models—mechanistic models—was determined to be the most suitable approach for this research project because of its balance between model complexity, accuracy, computational efficiency, and overall adaptability.

PDSim, a sophisticated modeling tool designed for PD compressors, was developed by Purdue University using the mechanistic modeling method. The versatility of PDSim is evidenced by its successful application in modeling diverse compressor types such as scroll, linear, and rolling piston compressors. Importantly, the accuracy of this modeling approach has been thoroughly validated through experimental studies conducted for each type of compressor.

As part of future research endeavors, PDSim will be leveraged to design compressors specifically for low-GWP refrigerants. The primary objective of this initiative is to optimize the performance of compressors for each type of these environmentally friendly refrigerants, with the dual intent of boosting operational efficiency and mitigating environmental impacts.

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