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ABSTRACT

Modeling scroll compressors with 1D fluid dynamics often involves the time-consuming exercise of calculating chamber volume and port areas vs. crank angle either manually from CAD or through scripting. In a novel approach, GT-SUITE simulation software is used starting from the original CAD model of the scroll fixed and orbiting involutes to create a 1-dimensional fluid dynamic model automatically. This allows models to be built in minutes while preserving the true CAD geometry. The process of calculating the chamber volume, port area, leakages, and areas required to convert chamber gas pressure to forces/moments on the orbiting scroll will be presented. Additionally, a comparison between the model prediction to test performance on a production automotive air conditioning scroll compressor will be shown.

1. INTRODUCTION

Scroll compressor performance has historically been maximized through rigorous testing, along with some simulation in several forms including 3D CFD (Picavet et al., Gao, and Gao et al.), 2D chamber modeling (Pietrowicz et al.), and 1D chamber modeling (Chen et al.). Modeling helps to accelerate the development cycle when designing the scroll compressor and reduces the cost of testing.

In order to build a 1D chamber model, the chamber volume, inlet and outlet port areas, and leakage areas as a function of crank angle must be determined in order to predict the detailed performance of the compressor. This often involves analytical calculations with assumed spline curvature for the involutes and using Green's theorem to integrate over the closed curvature representing a chamber to calculate its volume and port area (Yanagisawa et al. and Bell et al.). As an alternative and novel approach, a CAD model can be taken of the fixed and orbiting scrolls to obtain these volume and area profiles in an automated fashion through the use of a commercial multi-physics software called GT-SUITE. The details of this modeling approach are discussed in the next sections.
2. MODELING SCROLL COMPRESSORS IN 1D

A schematic of the scroll compressor geometry examined in this paper is shown in Figure 1 and shows the fixed scroll involute (orange), orbiting scroll involute (green) and outlet port (red). There are 8 total chamber volumes, as outlined in Figure 1, though at this particular snapshot in time, chambers 4A and 4B are non-participating and therefore not visible. As the scroll translates about an orbital path around an eccentric crankshaft, the scroll completes one orbit, 1A becomes the size of 2A, and 2A becomes the size of 3A, 3A becomes the size of 4A that is not participating in this snapshot moment in time, and so on. Though the scroll compressor cycle has periodicity with every orbit, in a 1D sense, the chamber volumes are being tracked from birth until death, which causes the cycle to repeat itself every 4th orbit for this particular scroll design.

![Figure 1: Basic geometry of three CAD solid shapes representing the fixed scroll involute (orange), orbiting scroll involute (green), and outlet port (red).](image1)

In the 1D flow model, the compressor flow system is discretized into many volumes comprised of pipes and flowsplits (volumes), and where pipe passages are further discretized into subvolumes. The 1D Navier-Stokes equations, namely conservation of mass, momentum and energy are solved every timestep in every subvolume. Conservation of species is also considered to solve for species transport throughout the system. In Figure 2, the 1D staggered grid approach used by the solver is shown, where scalar quantities like pressure, internal energy, temperature, density, etc. are solved at the center of each subvolume, and vector quantities like mass flow rate, velocity, species flow rate, etc. are solved at the boundary between subvolumes.

![Figure 2: Staggered grid solution in the 1D flow model solving Navier-Stokes equations and species transport in 1D.](image2)
At each timestep, the Navier-Stokes equations are applied to each subvolume and boundary in the grid; although, the order in which the subvolumes are solved is not important. The solver relies on NIST REFPROP for fluid properties of the refrigerant, which can be in the gas, liquid, two-phase or supercritical region.

The model has 15 total flow volumes. The average timestep is $2e^{-6}$ seconds. Most cases converge in about 50 pumping cycles, requiring about 3 minutes of clock time to run on a modern PC.

### 3. MODEL BUILDING PROCESS FROM CAD

The chamber model can be constructed directly from a CAD file of the scroll compressor based on three CAD solid shapes as shown in Figure 1, including the fixed scroll involute (orange), orbiting scroll involute (green), and outlet port (red). These three shapes are declared by the user in a specialized component inside the 3D pre-processor as shown in Figure 3.

![Figure 3: Definition of scroll compressor recognition of pointing to the three solid shapes inside the pre-processor, as well as definition of the crank angle resolution of the volume and area profiles.](image)

Once the scroll compressor is defined in the pre-processor, the solver will calculate the required volume and area profiles by translating the orbiting scroll shape along its orbital path in fixed increments and recalculate the volume of each chamber as well as the leakage and port areas.

The details behind the calculation of the 3D CAD and transforming into 1D chamber volume and area profiles are described next.

#### 3.1 Converting CAD to a 2D Model

The input CAD model consists of 3D solid-bodies. A local coordinate system may be found by comparing orientations of the input solid-bodies. This coordinate system will have a z-axis aligned with the scrolls from the fixed and orbiting bodies. Bounding boxes aligned with the z-axis are obtained for the fixed and orbiting bodies. A slice-plane is found by taking a mid-point from the intersection of the two bounding boxes in the z-axis direction. (Figure 4).
Figure 4: Fixed (orange) and orbiting (green) solid bodies and slice-plane.

A 2D representation is obtained for each scroll by intersecting the slice-plane with the input fixed and orbiting bodies. An outlet is found by identifying a hole in either the fixed or orbiting body that is closest to the input outlet body and projecting onto the slice-plane. It may be noted that geometry types in the wire-frame model are typically reflective of the input solid-model surface geometry types (e.g., blends become ellipses, planes become lines). Figure 5 illustrates the identified 2D model.

Figure 5: 2D model representation of fixed (orange), orbiting (green) and outlet (red) bodies. An orbit trajectory is illustrated with a circle (black) and initial points of conjugacy are line segment (black) ends.

Scroll tips (inner-most locations) are detected for each 2D scroll representation by a combination of searching for regions of high curvature and checks for proximity with other scroll locations.

Inlets are defined for each scroll by traversing away from the tip-location, identifying (ordered) edges on inner and outer spirals. An inlet is found at a tangent discontinuity or at a curvature inflection point. For both scrolls their corresponding inlets are found on their inner spirals as the algorithm traverses away from the tip locations.
A width (height of each scroll) is found by projecting a tip-location in the slice-plane normal and reverse-normal directions and searching for a scroll base in one direction and the end of the scroll protrusion in the other. The width can be derived from the two searches.

A circular orbit trajectory is obtained by first locating a point of conjugacy (a point where the two scrolls are in closest proximity) and intersecting an orthogonal line with both scrolls. A diameter may be chosen as the largest gap between points of intersection. The orbit trajectory is refined such that the orbiting scroll does not intersect the fixed scroll during the full orbit.

Once an orbit trajectory is found, the orbiting-body is oriented to close the fixed-inlet. This is the starting point for subsequent translations of the orbiting-body along the orbit trajectory. When fixed-inlet and orbiting-inlets are closed at different orientations of the orbiting-scroll along the orbit trajectory, as is the case for this model, the model is considered non-symmetric. Figure 5 illustrates the orbiting-body located in this start-location.

### 3.2 Converting the 2D Model to 1D Data

Once the scroll compressor is defined in the pre-processor as a 2D model with the fixed-inlet closed, the solver will calculate the required volume and area profiles by translating the orbiting-scroll along its orbital path (the orbit trajectory) in fixed angular increments and recalculate the volume of each chamber along with the leakage and port areas.

For the inlet area, this area is calculated for the first crank revolution by calculating the minimum distance between the scroll inlet and the opposite scroll, as shown in Figure 6, and multiplying by the width. This is done for both inlets associated with fixed and orbiting scrolls.

![Inlet port line](image)

**Figure 6:** Inlet port area defined by the distance between scroll inlet location and its projection onto the opposite scroll, as highlighted in blue, multiplied by the width.

Chamber area regions are recognized as lying between inlets, points of conjugacy (minimum distances where the two scroll bodies come close to touching) and a line-segment bounding the inner-most chambers near the scroll tip locations. This line-segment may require adjustment so as to not intersect either scroll. Points of conjugacy and the inner-most chamber dividing line-segment are recalculated for each angle increment. Figure 7 illustrates the points of conjugacy and the inner-most chamber line-segment boundary following the orbiting-inlet having been closed.
Figure 7: Chambers separated by points of conjugacy and a line segment separating scroll tips, following the orbiting-inlet having closed.

For each angle increment, chamber volume may be calculated from chamber areas by multiplying by the width. Other chamber volume measurements include center of gravity, inner and outer wall pressure force vectors, leakage lengths, and overlap with the outlet. The measurement data for each angle increment is gathered in schedules with respect to angle as shown in Figure 8. Chamber volume and areas were compared to the traditional methods in Figure 9. Differences are minimal and where present, are due to a slightly different definition of inlet opening and the calculation method of the overlap area. If the fixed and orbiting scrolls have different involute wrap extents, then separate volume and area schedules will be calculated to capture the asymmetry.

Figure 8: Volume and port area schedules for chambers beginning at chamber inlet opening.
4. MODEL SETUP

A schematic of the scroll compressor model is shown in Figure 10. Since there are 8 working chambers on the physical compressor, there are 8 unique working chamber volumes. The flow path from the inlet to the chamber volumes is highlighted in blue, while the flow path from the chambers to the outlet is highlighted in red. The leakage path between chambers is highlighted in green. The changing volume in each "FixedChamber1", "OrbitingChamber1", etc… part pulls the refrigerant from the inlet into each chamber as its volume expands. Then when the volume in each chamber compresses, the refrigerant is forced to the outlet side of the compressor. There is a reed valve to model the flow dynamics through the valve, along with an orifice restriction at the outlet with a calibrated diameter to target the test back pressure. The internal leakage between each chamber is modeled as an orifice with a leakage area to represent the radial leakage gap. There is an "overlap area" as shown in Figure 8 that is used as a communication area between Chamber1a and Chamber1b, Chamber2a and Chamber2b and so on. This corresponds to the area shown in Figure 7 when the chambers are compressing and a line between fixed and orbiting tip vertices is drawn to divide the chamber into two volumes (1a and 1b, 2a and 2b, and so on).

Figure 9: Chamber volume and area compared to traditional tools using Green’s Theorom

Figure 10: Model map showing all flow paths in the 1D flow model.
A reed valve is used on the outlet to prevent back flow through the chambers. A simple mass-spring-damper model is used to represent the valve opening dynamics based on the pressure differential across the valve and lookup a flow area based on the reed valve lift. Finally, boundary conditions for pressure, temperature, species, and speed are prescribed to the model, and the user can easily change these inputs to explore the design space.

5. TEST SETUP

To validate the model volute discretization, 1D simulations were compared to pocket and rear head test data. Two static pressure measurement locations were in the fixed scroll (Figure 11) and one static pressure measurement was in the rear head to capture dynamic pressure pulsations (not shown). Please note the third hole in Figure 11 is a compressor geometry feature and not used during the data collection process. The instrumented compressor was installed on a calorimeter with data captured after the system had reached thermodynamic steady state conditions.

![Figure 11](image.jpg)

Figure 11: Indication location with respect to scroll wrap.

6. RESULTS

The model results were compared to the test data to determine confidence and accuracy. Static pressure from the model predictions and test data were directly compared in Figure 12 where pressure was normalized to the maximum test measurement at the respective location. From the comparison, the chamber pressure matches well at the beginning of the compression process, but begins to diverge towards the end. In the rear head, the amplitude of the pressure pulsation response was lower for the model prediction; however, the frequency matched the test data.

Accuracy was determined based on the percent difference between the test data and the 1D results. The model and test values were directly compared at various revolution points using local interpolation where required. Figure 13 shows the corresponding percent difference for both the chambers and the rear head. Chamber test data has been combined into one compression curve for easier comparison. Higher confidence is placed near the beginning of the compression curve (revolution < 1.75) where the percent difference is 3.71% on average. After this location and to the end of the compression curve the average percent difference is 21.37%. The cause of decrease in accuracy towards the end of the compression curve has multiple possible reasons. First, the compressor geometry feature shown in Figure 11 acts as a bleed; this feature was not modeled and would add to the error. Also, the discharge reed dynamic response was not fully validated for the scope of this paper. Overall, the accuracy within the scroll with respect to pressure was good in the region of high confidence, and needs to be further vetted with additional testing comparisons at the upper end of the compression curve.
Figure 12: Direct static pressure comparison between model and test data

Figure 13: Pressure difference between model and the maximum test data

The pressure comparison in the rear head is similar to the chamber but more direct due to the simplified physics. The average difference was 0.52\% with a maximum of 3.96\%. The rear head pressure pulsation deviated due to amplitude and this was once again attributed to the reed valve model not being 100\% vetted.

Matching the model to test data required an iterative model development process which varied specific features due to unknowns in the test compressor. Features varied for sensitivity analysis and matching included reed valve stiffness and scroll wall leakage. Both variables had a non-negligible effect on both the scroll compression curve and the rear head pressure pulsation. Figures 14 and 15 show the impact of various leakage values on the chamber compression curve and compressor performance. The slope of the compressor curve increased with higher leakage gaps, while gaps that were smaller than 0.015 mm created artificial overshoots at the termination of the compression curve. Isentropic efficiency decreased with increased leakage gap as expected and was used to properly match the leakage gap to recorded testing efficiency. Using these parameters a realistic leakage gap was selected that met performance expectations.
7. CONCLUSIONS

This paper highlights the novel concept to create a 1D chamber model directly from a CAD model of the scroll compressor, to generate a running model within minutes from known compressor CAD geometry. The process of converting from CAD to a chamber model has been described and the model has shown good agreement to test for transient chamber pressure and outlet pressure pulsation. There were some unknowns in the test that required engineering judgment for input to the model, including details about the reed valve dynamics, as well as leakage gaps measured in the test. As such, the model used typical leakage gap specifications and reed valve characteristics to
produce the comparison to test and meets good agreement. The model also was used to show the trend of chamber pressure pulsation and isentropic efficiency with varying leakage gap, and clearly shows the expected trend of isentropic efficiency decreasing with increasing leakage gaps. Future work will involve additional compressor testing, with the inclusion of additional parameters such as mass flow rate and temperature, and will be used to further refine the leakage gap and reed valve modeling methods.

REFERENCES


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