A Thermodynamic Chamber Modelling Approach for Oil Free and Oil Injected Twin Screw Compressors

G Ramchandran¹ and J Harrison¹

¹ Gamma Technologies, LLC., Westmont IL 60559, USA

E-mail: g.ramchandran@gtisoft.com

Abstract. As computational modelling becomes an increasingly reliable and key component in accelerating the design process for twin screw machines, the goals for engineers now include developing faster running and physically accurate component models to optimize machine performance and efficiency, minimize internal leakage, reduce unwanted noise and pulsations, and properly size bearing supports in the machine. Accurately capturing these aspects via physical models helps in analyzing operating points that were not tested as well as in understanding how the machine will perform in a surrounding system. Thereafter, engineers can find an optimal design in a timely manner for the fastest speed to market as well as reduce physical testing to keep development costs low. This paper presents the use of a multi-physics modelling platform - GT-SUITE - in conjunction with SCORG – a well-established tool for the design and analysis of twin screw machines – to explore meeting the aforementioned goals. Two case studies are presented for a 3/5 oil free air compressor and a 4/5 oil injected air compressor. Comparisons to the mass flow rates of the gas and oil, temperatures, indicated power and the instantaneous chamber pressure vs rotation angle were made against test data available from the Centre for Compressor Technology at City University. The sensitivity of oil injection timing on the discharge temperature and power is shown and an optimum timing was found. The validated chamber models may be integrated into a system as well as used for further optimization to improve the original compressor performance.

1. Introduction

Rotary twin screw compressors are positive displacement machines that are widely used in light to heavy duty industrial applications to supply compressed air at a desired, continuous rate. Both oil free and oil injected variants are predominant in the market. Modern design of screw compressors involves the use of computational tools to analyze and optimize the performance of existing designs. Many researchers have investigated lumped parameter modelling approaches for oil free and oil injected screw compressors in the years past due to the advantages they offer in terms of computational speed and accuracy compared to more detailed 3D CFD based approaches. While 3D CFD provides more detailed insights into the operation of screw machines, it can be time consuming and not particularly convenient for geometrical optimization. Some of the original numerical models for oil free and oil injected screw compressors [1-10] were based on the approach of solving the equations of conservation of mass and energy applying to a control volume. Geometrical considerations such as the change in the chamber volume with rotation angle and the estimation of various leakage gaps are important aspects of these models to predict the performance of these machines with an acceptable degree of accuracy. Recent advances in
computational capabilities and the demand for quicker designs at low cost has spawned more advanced modelling efforts for screw compressors. Some of these include the works in [11-15].

In this paper, a commercial system modelling tool, GT-SUITE, is used in conjunction with an industry established screw compressor geometry modelling tool, SCORG, to formulate an effective alternative to existing thermodynamic chamber models for screw compressors. GT-SUITE has been used to model swashplate type refrigerant-based compressors [16], scroll compressors [17], and screw expanders [18-22]. This is the first study showcasing the ability to model twin screw air compressors using GT-SUITE. Two case studies are presented comparing GT-SUITE’s model results to SCORG’s thermodynamics solver and test data for oil free and oil injected screw compressors. Also shown is a use case for optimizing screw compressor designs using GT-SUITE.

2. Modelling workflow
A complete model considers all flow paths present within the system. GEM3D (a GT-SUITE pre-processor) was used to import the CAD of the flow path, discretize this into multiple pipes and flowsplits, and finally export the flow system into the GT-ISE modelling environment with the screw machine template. SCORG was then used to generate the chamber volume and port/leakage area profiles. Seamless information transfer from SCORG to GT-SUITE greatly reduces model setup time. Figure 1 illustrates the modelling workflow followed. Details are provided in the subsequent sections.

Figure 1. (a) Geometrical parameters and profiles generated by SCORG and written into text files, (b) GT-SUITE model run with pre-processed inputs.
2.1. Model setup and pre-processing in SCORG
SCORG V5.8.3 was utilized for the first step in the modelling process which involves setting up the rotor profile geometry, defining ports, thermodynamic operating conditions, working fluid, and oil injection (if present). Once geometry calculations are run, a text file containing the chamber volume, inlet and outlet port areas (radial/axial/both) and leakage areas for each rotation angle of the main rotor is generated for use by the GT-SUITE chamber model. An additional text file containing parameters such as boundary pressures, temperatures, compressor speed, geometrical features for the inlet/outlet pipes, port volumes, etc. is generated for the GT model. Optionally, force calculations may be run in SCORG to generate another text file with the axial and radial bearing loads vs angle in order to take advantage of the detailed journal and roller bearing models available in GT-SUITE to evaluate compressor rotordynamics, modal analysis, friction losses and bearing performance. However, these topics are out of the scope of the current study which is focused on the thermodynamic performance.

2.2. Thermodynamic chamber modelling in GT-SUITE
Once the geometry calculations are complete, the GT-SUITE solver may be accessed either from within SCORG itself or via the GT-SUITE user interface. Templates with various combinations of the number of working chambers are available in GT-SUITE. Running the GT-SUITE solver from within the SCORG interface automatically picks the GT template with the expected number of chambers.

The GT templates contain chamber models built from flow components and connections as shown in Figure 2. Pipe templates are used for the inlet and outlet piping. Every working chamber is linked to the inlet and outlet port volumes that are modeled as flowsplits. Individual links are present for the axial and radial ports, as well as for the various leakage paths between chambers. The compressor speed is imposed on the shaft. The number of chambers in the model is evaluated based on the product of the number of main rotor lobes and the integral number of shaft rotations in one working cycle.

![Figure 2. GT-SUITE model map with all the flow paths in an oil injected twin screw compressor](image)

The pre-processor GEM3D is a useful tool where the CAD of the flowsystem (pipe volumes) is imported, cutting planes are used to separate the flowsystem into different flow components, and a simple conversion process is used to convert each shape into a flow component. The 1D flow model
consists of the compressor flow system which is discretized into many volumes comprised of pipes (which are further discretized into subvolumes) and flowsplits (single volumes). A staggered 1D grid approach is adopted (as indicated in Figure 3) 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.

GT-SUITE solves equations for the conservation of continuity, momentum, energy and species in 1D, which means all quantities are uniform within each pipe subvolume. The timestep size used by the explicit flow solver is adaptive and always small enough to satisfy the Courant condition. This allows for the ability to capture phenomena such as unsteady flow, pressure pulsation and high frequency wave dynamics. Although the code is nominally 1D in pipes, flowsplits are specifically designed to account for conservation of momentum in 3D. Scalar terms are solved at the volume’s center while the solution of momentum equation (vector quantities) is carried out separately at each volume opening (boundary). For the momentum equation, the flowsplit geometry is characterized for each boundary by its expansion diameter (diameter into which flow expands after entering flowsplit), characteristic length (distance from boundary plane to opposite side of flowsplit) and boundary orientation (relative angle of each boundary). A characteristic velocity vector is calculated for the flowsplit based on the contributions of momentum flux from all of its boundaries. The momentum flux out of the flowsplit is calculated by using the component of the characteristic velocity in the direction of the boundary. This adequately captures flow through boundaries with arbitrary orientations which is not possible with conventional 1D treatments or typical lumped parameter modelling approaches.

In the current setup, the walls of the flow components are assumed to be adiabatic, so no heat transfer is considered between the fluid and the wall. However, the option to consider heat transfer is available and may be investigated in future studies. Flow through leakage paths and between adjacent subvolumes are modelled through orifices. These are defined via diameters or areas and discharge coefficients as inputs. The momentum equation is solved to calculate mass flow rate through the orifices. For gases, the mass flow rate is calculated based on the isentropic nozzle relationships for subsonic and choked flow regimes (Equations 1 and 2 respectively).

\[
\dot{m} = C_D A_R \rho_o \left( \frac{P_{2,\text{Static}}}{P_{1,\text{Total}}} \right)^{1/\gamma} \sqrt{\frac{R T_o}{\gamma - 1}} \left\{ \frac{2 \gamma}{\gamma - 1} \left[ 1 - \left( \frac{P_{2,\text{Static}}}{P_{1,\text{Total}}} \right)^\gamma \right] \right\}^{1/2}
\]

\[
\dot{m} = C_D A_R \rho_o \left( \frac{2}{\gamma + 1} \right)^{1/\gamma} \sqrt{\frac{R T_o}{\gamma + 1}} \left\{ \frac{2}{\gamma + 1} \right\}^{1/2}
\]

Here, \(\dot{m}\) is the mass flow rate, \(\rho_o\) is the upstream stagnation density, \(C_D\) is the discharge coefficient, \(A_R\) is the reference flow area, \(P_{1,\text{Total}}\) is the total pressure upstream of restriction, \(P_{2,\text{Static}}\) is the static pressure downstream of restriction, \(R\) is the gas constant, \(T_o\) is the upstream stagnation temperature, and \(\gamma\) is the specific heat ratio.

In the case of liquids, the flow rate is based on the incompressible Bernoulli flow equation through an orifice plate (Equation 3). Here, \(D\) is the orifice diameter and \(\rho\) is the liquid’s density.

\[
\dot{m} = C_D \rho \left( \frac{\pi D^2}{4} \right) \sqrt{\frac{2(P_{1,\text{Total}} - P_{2,\text{Static}})}{\rho}}
\]
The transient species mass fraction is solved for within each subvolume and this is important in the case of oil injected compressors where the oil and air are modeled as a homogeneous mixture within each subvolume. Flow is based on the species mass fraction in the upstream subvolume.

In the case studies discussed in the following section, the models built have between 41 to 65 flow volumes depending on the total number of chambers. The average timestep sizes are 1.5e-5 seconds and 6.2e-6 seconds for the oil free and oil injected compressor models, respectively. Most operating conditions converge in 6 to 7 cycles, requiring less than a minute of run time on a modern PC.

3. Case studies

Two screw compressors for which test measurements are available in the public forum were chosen to validate the GT-SUITE model predictions. Measurements at a few operating points were carried out on the air compressor test rig at the Centre for Compressor Technology in City University of London. Details regarding the test setup such as the various sensors (orifice plates, pressure transducers, thermocouples, torque meters) for measuring flow, pressures, the instantaneous chamber pressure in the interlobes, temperatures, and the compressor torque are available for the dry [23] and oil injected air compressors [24]. Figure 4 shows the test schematics for both compressors. Additional steady state measurements for the dry compressor are available in [13]. To highlight GT-SUITE as an effective alternative to one of the predominantly used chamber models in the industry, comparisons were also made to SCORG’s thermodynamic solver in addition to test data.

Table 1 contains the range of operating conditions presented in these comparisons. Table 2 provides information regarding the geometry of the two compressors being modeled. The interlobe, radial, and end axial clearances were reduced by 5 to 20% from the nominal values in the models in order to better correlate to the test measurements. This was deemed acceptable since the operational clearances vary non-uniformly from nominal clearances based on the operating conditions.

**Figure 4.** Test schematic for (a) oil free compressor from [23], (b) oil injected compressor from [24].

Table 1. Operating Conditions Modelled for Comparison to Measurement

| Units          | Oil Free Machine | Oil Injected Machine |
|----------------|------------------|----------------------|
| Compressor Speed | RPM              | 6000 to 9000        | 3000 to 6000         |
| Suction Pressure | bar              | 1.0                  | 1.0                  |
| Suction Temperature | K                | 298.15               | 298.15               |
| Discharge Pressure (Pd) | bar              | 2.0 to 2.5           | 6.0 to 8.0           |
| Oil Injection Pressure | bar              | -                    | Pd - 0.5             |
### Table 2. Screw Compressor Geometry (with ‘N’ rotor profiles)

| Unit                  | Oil Free Type | Oil Injected Type |
|-----------------------|---------------|-------------------|
| **Main Rotor Lobes**  | -             | 3                 |
| **Gate Rotor Lobes**  | -             | 5                 |
| **Axis Center Distance** | mm         | 93                |
| **L/D Ratio**         |               | 1.6               |
| **Main Rotor Diameter** | mm         | 127               |
| **Wrap Angle**        | deg           | 306               |
| **Built-in Volume Index** | cc/rev     | 1.9               |
| **Displacement**      |               | 1787              |

### 3.1. Case study 1 – oil free screw compressor

In this section, some of the GT-SUITE model results and comparisons to test data and the SCORG thermodynamics solver are presented for the oil free twin screw compressor.

#### 3.1.1. Pressure-angle diagram

Figure 5 shows the variation of the chamber pressure with the angle of rotation of the main rotor at two different speeds and for different pressure ratios. While the general trend and GT-SUITE comparisons to test measurement are good, there are some differences noticed close to the opening of the discharge port near the pressure peak. Additionally, the pressure wave dynamics occurring during the compression of the working chamber is captured reasonably well compared to test and is an improvement to SCORG’s prediction where the oscillations appear to be over-damped. The peak pulsation is also captured more accurately as compared to SCORG.

![Figure 5](image)

**Figure 5.** Comparisons of chamber pressure variation for the oil free screw compressor.

#### 3.1.2. Mass flow rates

Figure 6 shows the comparison of the mass flow rate predictions against test data at different compressor speeds and pressure ratios. By adjusting clearances, a good match was found between test and GT-SUITE. This tracks well with what is also predicted by the SCORG model.

![Figure 6](image)

**Figure 6.** Comparisons of average mass flow rates at various speeds for the oil free screw compressor.
3.1.3. Indicated power. Figure 7 shows the comparison of the GT-SUITE indicated power predictions against test data and SCORG at different compressor speeds for a pressure ratio of 2.0. Since the power was measured on the shaft, a constant mechanical efficiency of 95% was assumed for the gearbox driving the main rotor in the test data across all speeds. While probably not the best estimate since the mechanical efficiency of the gearbox typically varies with compressor speed, there is a good agreement in the results with the model predictions within 8% of the test measurements. The GT-SUITE results show a closer prediction to test data compared to SCORG.

![Figure 7. Comparisons of the indicated power at various speeds for the oil free screw compressor.](image)

3.2. Case study 2 – oil injected screw compressor

In the next study, model comparisons to available test data for the case of the oil injected twin screw compressor are discussed.

3.2.1. Mass flow rates. Figure 8 shows the comparison of the normalized mass flow rates predicted by the model compared to available test data. The model trend compares well to the test data with the predicted values being within 7% of the test results.

![Figure 8. Comparison of the normalized average mass flow rates at different steady state speeds to test data for the oil injected screw compressor.](image)

3.2.2. Indicated power. Figure 9 shows the comparison of the normalized indicated power predicted by the model compared to available test data at various compressor speeds for the same operating conditions. A constant mechanical efficiency of 70% was applied to the measured power to estimate the experimental indicated power. For a better comparison against the actual measured power, the built-in GT-SUITE friction model for seals, bearings and rotor drag may be used. Nevertheless, with the current assumption, the model predictions are within 9% of the test data.

![Figure 9. Comparison of the normalized indicated power at different steady state speeds to test data for the oil injected screw compressor.](image)

3.2.3. Additional observations. Figure 10 shows a few other model predictions. It can be seen that the temperature never exceeds 335 K even at 6000 RPM and a pressure ratio of 8.0. From the P-V diagram, it is clear that the amplitude of the pressure pulsations in the chamber increase with both speed and discharge pressure. The compressor appears to be under-compressed at the operating
conditions analysed, i.e., the pressures in the chambers are less than the targeted discharge pressures at the time of the discharge port opening. The peak over-compression can be attributed to throttling and dynamic losses occurring near the outlet port. This restriction is captured by the model with an orifice downstream of the discharge port. A 3D CFD analysis of the same compressor [25] confirms a similar trend. The model also predicts the mass fraction of oil being injected into the working chamber.

Figure 10. (a) Comparison of chamber temperatures, (b) Comparison of the P-V diagrams, (c) Model predicted instantaneous mass fraction of oil in the working chamber

3.2.4. Optimizing oil injection. An exercise was carried out to study the sensitivity of varying oil injection timing and injection port areas on the discharge temperature and indicated power of the compressor. The built-in design optimizer in GT-SUITE was used and a genetic algorithm was chosen to conduct this study. The objective of this design optimization was to find pareto-optimal designs that minimized the discharge temperature of the compressor while maximizing its indicated power. The objective functions for the average discharge temperature and the indicated power were created by equally weighting the two operating conditions: (3000 RPM, PR = 6) and (6000 RPM, PR = 8.0). 150 designs were evaluated by the genetic algorithm. Figure 11 shows the contour plots of the discharge temperature and indicated power plotted for different combinations of timing shift (base design angle shift = 0) and injection port areas (base design injection area multiplier = 1). It is evident that as more oil is injected into the compressor, temperature in the working chamber and subsequently discharge temperature is closer to the oil injection temperature. With more injected oil combined with earlier oil injection (negative angle shift) in the working cycle, a better design is achievable. The pareto plot in Figure 12 indicates all the combinations of optimized designs available to choose from.

Figure 11. Contour plots of (a) average discharge temperature, and (b) average indicated power, (both equally weighted for the two operating conditions considered) for designs varying the injection timing and injection port area magnitude.
4. Conclusions
Thermodynamic chamber models were formulated for oil free and oil injected twin screw air compressors using GT-SUITE. The overall objective of this work has been to provide screw compressor designers with a toolset to find optimal designs in a timely manner for the fastest speed to market while reducing physical testing and keeping development costs low.

- Model comparisons to test measurements of mass flow rates, instantaneous chamber pressures, and indicated powers of both test compressors from available literature show strong agreement.
- The validated model may be used to predict performance at operating conditions difficult to test.
- A design analysis was conducted to evaluate the optimal injection port area as well as the injection timing. A set of pareto-optimal designs were identified, and it was observed that by increasing the oil injection port area and advancing the injection timing, designs offering a lower discharge temperature along with higher indicated powers may be obtained.
- Future work may involve adding a friction model to evaluate operational power accurately and break down the different contributors to friction power loss (bearings, seals, rotor drag, etc.).
- The model may be extended to study the operation of these compressors when present in a system.
- Further studies may also be conducted to minimize leakages, noise and pulsation (muffler design), size bearing supports for sustainable operation, and model shaft rotordynamics in GT-SUITE.

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