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Simulation of 1D Flow Coupled with 3D Multi-Body Dynamics Model of a Double-Acting Swashplate Compressor

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ABSTRACT

This paper describes an integrated 1D/3D modeling approach for a multi-cylinder double-acting automotive A/C compressor using a multi-physics CAE system simulation software, GT-SUITE. This tool is used to build a 1D flow model of a compressor using the original CAD, along with a 3D multi-body dynamics model to study overall compressor performance. Various flow and mechanical-related built-in libraries, supported with pertinent physics, are utilized to study fluid flow and related forces on the pistons, shaft and the swashplate. There exists an easy approach in the software to schedule the suction and compression cycles in the cylinders. A full 3D modeling approach becomes computationally expensive especially when the compressor performance is to be evaluated during speed ramps, while the proposed 1D flow modeling approach yields results quickly.

Analysis results will include gas pressure forces, effects of piston and swashplate inertia, as well as the generated mass flow rates. Also included will be the results of transient pressure pulsations and forces. The model also is able to predict any flow-related resonances that exist. Comparison with measured data is shown, where possible.

1. INTRODUCTION

The use of automotive air conditioning systems is on the rise. These systems consist of a variety of compressors ranging from low-cost fixed displacement compressors to high-end variable displacement compressors of the piston type. Other types of commonly used compressors include belt-driven, as well as electrically-driven scroll compressors. This paper discusses the modeling of a belt-driven fixed displacement piston compressor. Single-acting and double-acting compressors are used in automotive A/C systems depending on the size of vehicle. For discussion purposes, a double-acting compressor is chosen as the subject of this paper.

There is a demand for compressors that are designed for low noise and vibration along with enhanced performance capabilities. These compressors must withstand harsh underhood conditions and also, belt-driven compressors must undergo varying speeds without inducing passenger discomfort. These factors impose a considerable challenge to compressor manufacturers in the development of design strategies. In recent years, this has been addressed by adopting various numerical modeling approaches. These consist of 1D and/or 3D modeling schemes involving flow, structural, thermal and acoustic fields. Xie and Bansal (2000) took a control volume approach to model the performance of a reciprocating compressor without modeling the pressure dynamics. Liu and Hui (2012) modeled

the kinematics and dynamic motion of a wobble plate compressor with analytical relations used for the pressure and friction forces. Srivastava et al. (2016) proposed a simplified mathematical model to predict the suction and compression effects of a fixed displacement swash plate compressor as well as the shaft torques. Gimelli et al. (2012) compare the mass flow rates predicted by a dynamic 1D thermo-fluid dynamic model accounting for the pressure wave dynamics with experimental results. Kacani (2016) presented a parametric design process to predict the strength of the different components of the piston unit within a reciprocating compressor by adopting a 2D/3D FEA approach.

In the past, flow and structural analyses were performed separately with manual transfer of boundary conditions between analyses. Recently, fluid-structure analyses have been combined with limited applications (Sun and Ren (1995), Xie et al. (2017)). Lemke et al. (2016) compared a 3D Fluid-Structure Interaction (FSI) dynamic model of a flapper valve to a corresponding 1D model as well as experimental measurements. Dhar et al. proposed a 3D transient CFD model as well as an FSI model for port flip valves for a single piston compressor.

In this paper, a novel approach combining a 1D flow model with a 3D multi-body dynamics structural model is presented. This methodology allows for a fast running model that can be used in various stages of the design process for swashplate compressors, while ensuring accurate performance predictions. The commercial multi-physics software, GT-SUITE, is used in this study.

2. MODELING SWASHPLATE COMPRESSORS IN 1D

Figure 1 shows a cross-sectional view of the simplified double-acting swashplate geometry studied in this paper. Compared to a more common swashplate compressor where each piston only has one chamber, the pistons on a double-acting compressor each have two chambers opposite to each other. This particular swashplate compressor is a wobble plate type compressor where the piston is constrained to move linearly in the direction of the rotation axis and the angled swashplate rotates about the same axis, thereby creating a wobble motion that displaces the pistons. As the pistons move, the volume of each chamber changes. Suction occurs when the inlet port of a particular chamber overlaps with the suction side volume of the compressor while the volume in that chamber is increasing. This volume is then displaced during the compression cycle where the outlet port area is activated by the opening of a reed valve to allow the compressed fluid to be pushed out at a high pressure. Each reed valve opens when the pressure within the chamber overcomes the cracking pressure of the valve. Shoe joints are present between each piston and the swashplate that allow for the swashplate to slide at these interfaces. The friction at these joints also are studied in a later section. Note the fixed swashplate angle, indicating a positive displacement compressor.

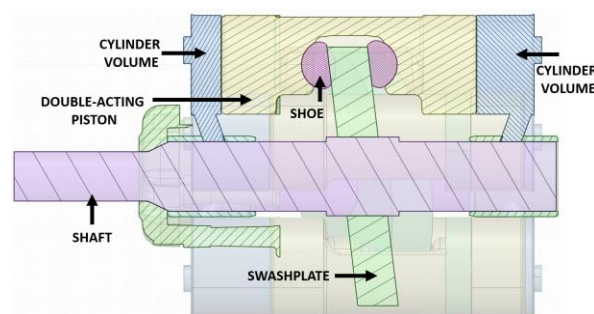


Figure 1: Basic cross-sectional view of a double-acting swashplate compressor.

The simulation tool used for this study, GT-SUITE, solves the Navier-Stokes equations in 1D where the equations of conservation of mass, momentum and energy are solved at every timestep in every subvolume. The 1D flow modeling is performed by discretizing the entire compressor flow system into many volumes consisting of pipes and flowsplits. The pipe passages are further discretized into smaller subvolumes along their lengths. Conservation of species also is 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.

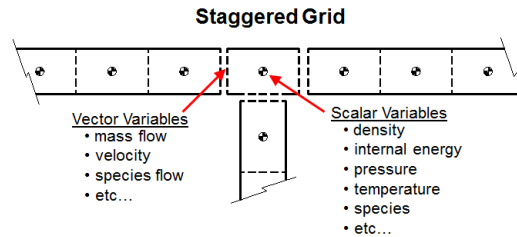


Figure 2: Staggered grid solution utilized in GT-SUITE for solving Navier-Stokes equations and species transport in 1D.

At every timestep, the Navier-Stokes equations are applied to each subvolume and boundary in the grid, though the order in which the subvolumes are solved is irrelevant. 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 126 total flow volumes, 116 of which are pipe and flowsplit subvolumes and 10 of which are the cylinder chamber volumes. The average timestep is 6e-6 seconds. Most cases converge in 10 to 15 cycles, requiring about 2 to 3 minutes of real time to run on a modern PC.

3. MODEL BUILDING PROCESS AND MODEL INPUTS

Building a complete swashplate compressor model can be a time-consuming process since such a model requires critical information that includes:

1. Discretizing the entire flow path of the refrigerant into multiple subvolumes
2. Defining the chamber volume variation profile, and the inlet and outlet port area profiles
3. Analyzing the effects of the various components of forces (gas pressure, inertial forces due to the pistons and swashplate, contact friction forces) on the moving components as well as to estimate the torque requirement using an optional mechanical system model of the compressor

Each of these model building steps will be described in detail in the following sections.

2.1 Modeling the Flow Path

The pre-processing tool called GEM3D in GT-SUITE enables an easy building process for the flow model. Figure 3 illustrates this process. The CAD of the inlet and outlet flow volumes of the compressor is imported into GEM3D and they are discretized and converted into flow components. The 1D flow model is automatically generated once complete. This process allows for a fast generation of the flow model including defining the initial and boundary conditions of the flow system, such as the targeted suction and discharge pressures for the compressor, the refrigerant properties, as well as components such as bearings, valves and other flow restrictors that are present in the system.

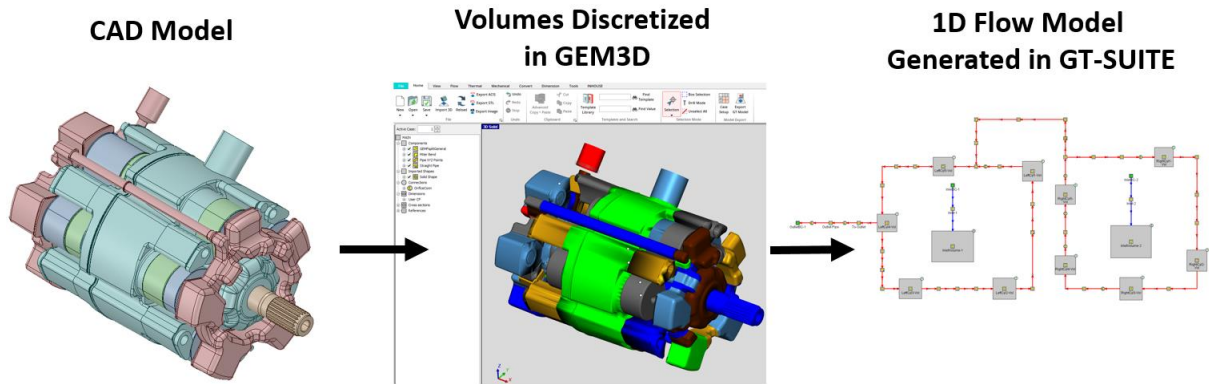


Figure 3: Process of importing the CAD model into the GEM3D pre-processor and generating the flow model.

2.2 Building the 1D Compressor Model

To reduce the overhead in building the 1D model of the compressor, there are pre-defined templates available within GT-SUITE's flow library. The variation in volume within each chamber – over one rotational cycle of the shaft – is automatically computed within the pre-defined template "SwashPlateShaft," which accepts easily defined inputs representing the geometrical attributes of the swashplate and chambers. This includes a starting angle for the volume within "Chamber 1" where the piston is assumed to be at the top dead center (TDC) of the chamber. The dead volume at this position is required to estimate the minimum volume within the chamber over the cycle. Other easily defined geometrical attributes – such as the cylinder bore diameter, number of cylinders, the radial position of the cylinders on the swashplate and the swashplate (incline) angle – are used to evaluate the instantaneous chamber volume that the refrigerant can occupy (within Chamber 1) as a function of the shaft angle. Depending on the number of cylinders, the volume profile is automatically phased for each of the other chambers influenced by the swashplate's motion. All of the above described inputs are easily evaluated from the CAD model of the compressor and are illustrated in Figure 4. The "SwashPlateShaft" template is driven by a "SpeedBoundaryRot" template that is used to specify the rotational speed of the shaft, which either can be constant (at steady state) or assigned to a speed profile to study a transient ramp.

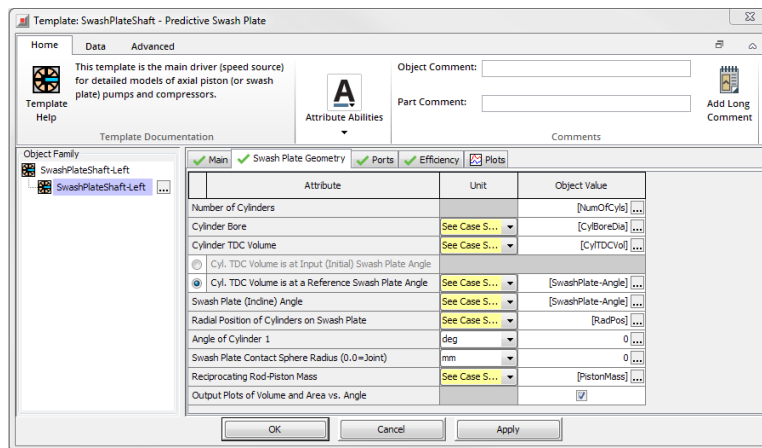


Figure 4: Geometrical attributes input within a pre-defined "SwashPlateShaft" template to automatically evaluate and phase the volume variation within each cylinder as a function of the shaft rotation angle.

The port area schedule (overlap) between Chamber 1 and the inlet port is computed as a function of the shaft rotation angle based on a variable overlap area from the CAD model. Similarly, the outlet port area variation is

computed based on the reed valve lift as pressure builds up within each chamber. The phasing of these area profiles happens automatically based on the number of cylinders.

The volume and area profiles for two chambers present opposite each other across one piston are as shown in Figure 5. As seen from this figure, the profiles are shifted by exactly 180 degrees as expected.

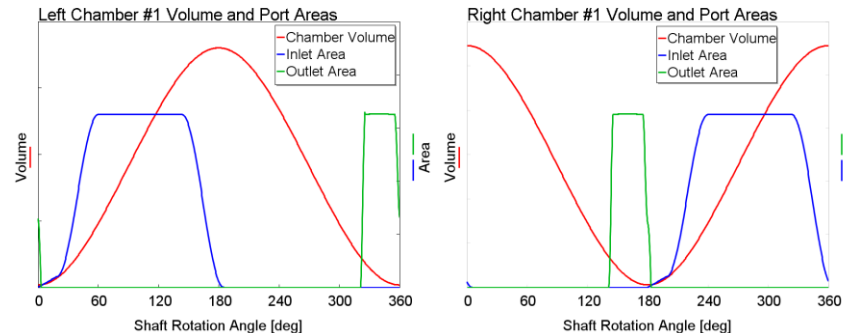


Figure 5: Volume and port area profiles for the chambers on either side of Piston 1 as a function of shaft rotation angle.

2.3 Integrating the 1D Flow Model with a 3D Mechanical Model

In addition to the 1D flow model, a 3D multi-body dynamics mechanical model of the swashplate assembly was built by representing the various parts with easy-to-define templates from GT-SUITE's mechanical library. The primary reason behind building this model was to analyze the various forces and moments acting on the swashplate, as well as the friction at the contact interfaces, and the overall torque requirement at various operating conditions.

Figure 6 shows the schematic view of the mechanical structure of the double-acting swashplate compressor.

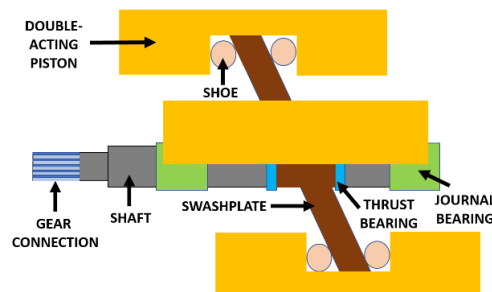


Figure 6: Schematic view of the mechanical components in the swashplate assembly.

The pistons and the swashplate are modeled as rigid bodies. The swashplate (at a fixed angle of inclination) is constrained to rotate about the shaft axis (creating a wobble motion), while the pistons are constrained to translate along the same axis, thus displacing the fluid within the cylinders. These components are modeled by considering their center of mass locations, their masses, and the moments of inertia in order to compute the inertial forces associated with their motion.

The pressure force exerted by the gas on each piston face is transmitted to the surface of the swashplate via the shoes present between each piston and the swashplate. Additionally, the surfaces of the swashplate that come in contact with the pistons are modeled as flat discs, and their locations, orientation (based on the swashplate angle), and radii are specified according to the CAD model. The shoes are modeled as spheres, and the contact between the shoes and the swashplate surfaces are modeled using Hertzian elastic contact theory and enable the evaluation of the contact friction between the two geometries, the Hertzian stresses, elastic deformations and the friction power loss at the contact interfaces. The material properties of the piston and swashplate surfaces – such as density, Young's modulus, Poisson's ratio and thermal expansion coefficients – are taken into account for the contact model. Additionally, the

model also considers the surface roughness of both the piston and swashplate surfaces and has a built-in Greenwood-Tripp asperity contact model to calculate the asperity loads in a mixed-lubrication contact.

The shaft of the compressor is modeled using beam elements with multiple nodes that can each have 6 degrees of freedom. These beam elements are subjected to gravity, external forces/torques and other constraints such as rotation about the shaft axis as in this case. The shaft is modeled with beam elements along with its material properties so the deflection of the shaft can be considered, especially at high shaft speeds.

Finally, the compressed refrigerant gas exerts pressure on the faces of the pistons and this pressure variation – as a function of the shaft angle – is obtained from the 1D flow model, thereby completing the integration between the flow and mechanical models.

4. MODEL SETUP

A schematic of the flow model of the swashplate compressor is shown in Figure 7-A. Since there are 10 working chambers where the refrigerant undergoes suction and compression, there are 10 unique chamber parts on the model map. The flow path from the inlet into 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 the outlet of each chamber to the inlet is highlighted in green. Each chamber part sucks in the refrigerant from the inlet the volume expands. Once the volume in each chamber is compressed, the refrigerant is forced to the outlet side of the compressor. There is a reed valve present between the chamber volume and the outlet 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 the outlet and inlet of the chambers – through the radial gap between the piston and cylinder – is modeled as laminar leakage flow by accounting for the pressure differential across the leakage interface as well as the effect of the motion of the piston with respect to the fixed cylinder (Poiseuille and Couette flow). The changing leakage piston length due to piston motion also is taken into account. A reed valve is modeled on each outlet port to prevent back flow through the chambers. It also models the flow dynamics of the valve based on a simple mass-spring-damper model and is used to represent the valve opening dynamics based on the pressure differential across the valve and looks up an outlet flow area based on the reed valve lift. Finally, the boundary conditions for pressure, temperature, species (refrigerant) and speed are added to the model; whereas, all these attributes, as well as the geometrical attributes of the model, can be parameterized to explore the design space and perform optimization studies.

A schematic of the mechanical 3D multi-body dynamics model is shown in Figure 7-B. The pistons and swashplate are represented by 3D Inertia components. The shaft is composed of 3D beam elements. The shoe-swashplate contacts are modeled as connections present between each piston (sphere modeled at the piston face nodes to represent the shoes) and the swashplate (circular flat plates modeled at the swashplate surface nodes). The chamber pressures are obtained by sensing the pressure in each chamber and applying it to either of the pistons' faces.

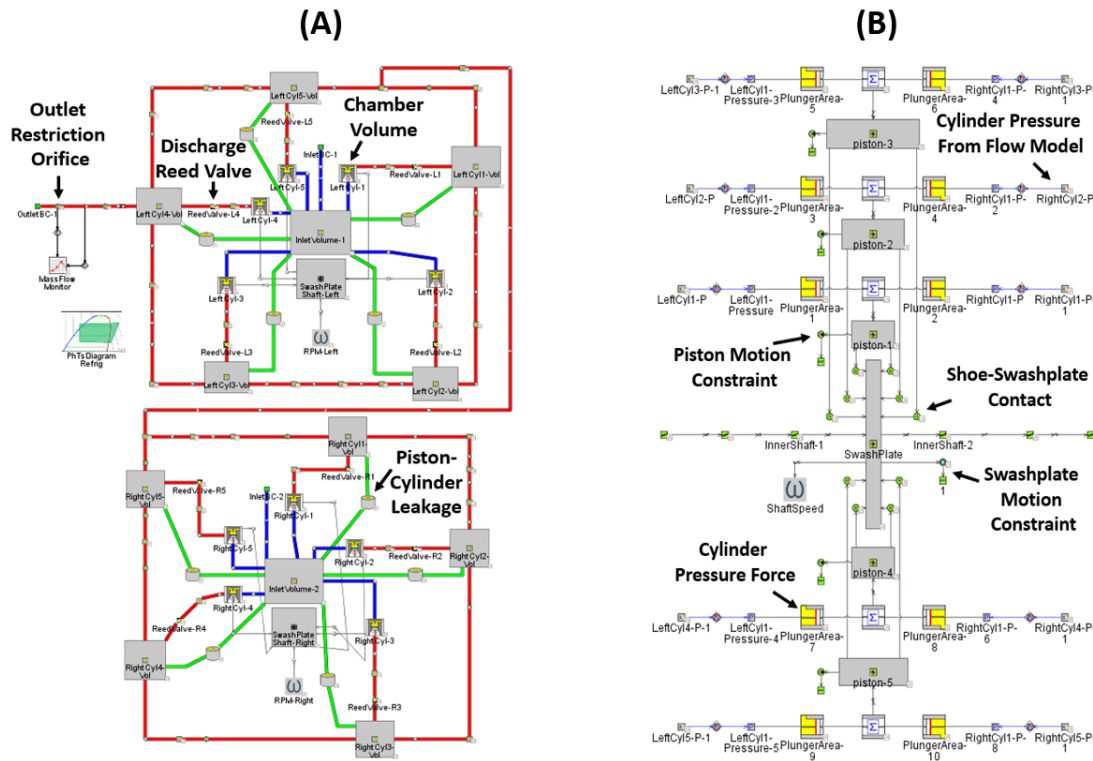


Figure 7: (A) Model map showing all flow paths in the 1D flow model; (B) Model map showing the 3D multi-body dynamics components.

In the next sections, the test setup will be discussed as well as some of the results obtained from these models.

5. TEST SETUP

Figure 8 shows the compressor studied in this paper mounted on a test stand in the laboratory. This compressor is equipped with accelerometers to capture the body vibrations and a dynamic pressure transducer mounted on the outlet pipe downstream of the compressor to capture the pressure pulsations.

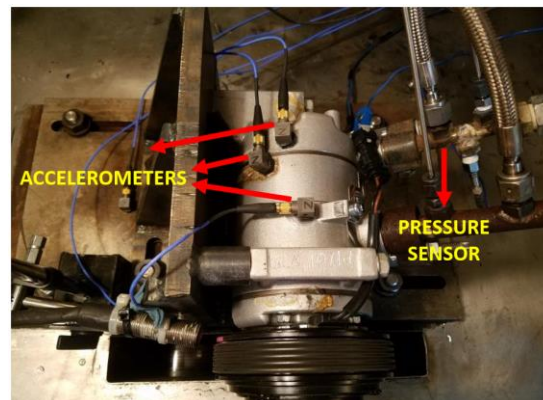


Figure 8: Experimental setup showing the locations of dynamic pressure measurement and accelerometers.

The test rig is capable of capturing the instantaneous pulsations for steady state as well as speed ramp up and ramp down operation. The instrumented compressor was installed on a calorimeter for testing at specific operating

conditions. Data was captured after the system had reached thermodynamic steady state conditions for 10 minutes. The tests were repeated multiple times and the average values for the quantities have been used in validating the model. For the purposes of comparison of test data to the model predictions, the data at steady state for specific constant shaft speeds were chosen.

6. RESULTS

Figure 9 illustrates the comparison of the mass flow rate of the refrigerant at the outlet of the compressor at various shaft speeds. The mass flow rates predicted by the GT-SUITE model have been normalized with respect to the test measurements. Four constant shaft speeds were chosen, ranging from a low (1) to high (4). The diameter of the restriction orifice at the outlet of the pipe in the model was calibrated to target the test back pressure at each of these shaft speeds. As seen from the figure, the model is able to predict the mass flow rates to within 3 - 5 % of the test results.

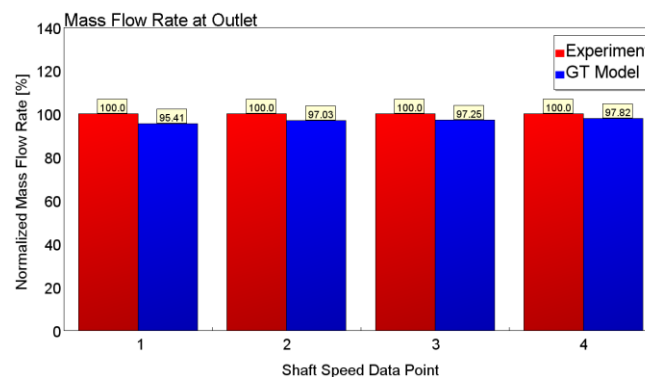


Figure 9: Comparison of predicted compressor mass flow rates against test data at various shaft speeds.

The model also captures the crank-angle resolved instantaneous pressure pulsations at the outlet of the compressor. This is illustrated in Figure 10 for two shaft speeds (low and high). An interesting finding from this comparison was that at low shaft speeds (1000 RPM shown), five pressure pulsations were observed (corresponding to the number of chambers on one side of the swashplate), whereas at higher speeds (9000 RPM shown), there were only four pulsations seen.

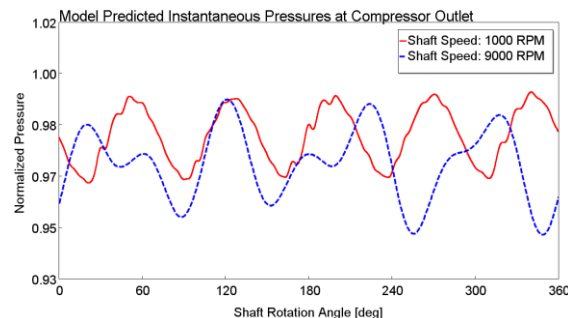


Figure 10: Comparison of normalized pressure peaks at compressor outlet over one cycle.

This trend was verified by comparing the FFT of the pressure trace at the outlet between the test data and the model for the two shaft speeds. Figure 11-A shows the FFT of the discharge pressure amplitude at the compressor outlet at 1000 RPM comparing the test data and the model prediction. The first peak (in both test and model) is observed at 83 Hz which is the 5th order and this corresponds to the number of pistons on one side of the swashplate. Figure 11-B shows a similar plot comparing the test and model predicted FFT's at 9000 RPM. Here, the first peak is observed at 600 Hz, which is the 4th order. These comparisons illustrate the capabilities of the model in capturing the fluid-

borne (refrigerant) noise. While the trend of these pulsations has been captured satisfactorily, the discrepancies in the magnitude of the pulsations, as well as the amplitudes at higher frequencies, can be attributed to the discharge reed valve dynamic response which was not fully validated for the scope of this paper.

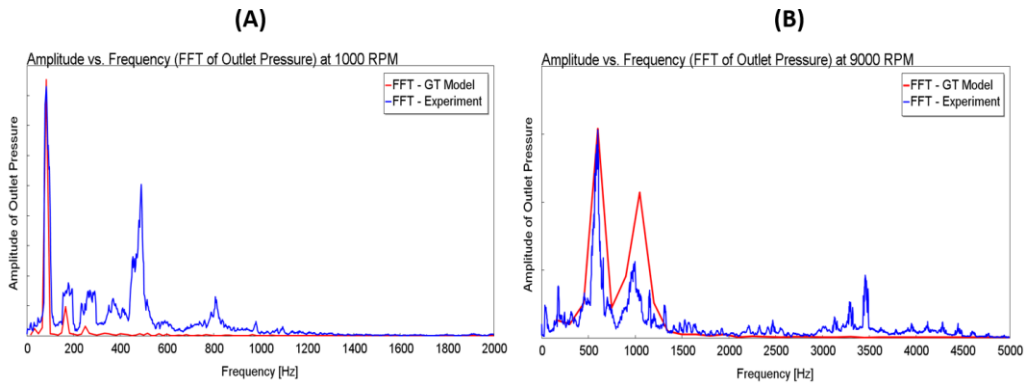


Figure 11: (A) FFT of the discharge pressure pulsation at compressor outlet (1000 RPM): primary harmonic corresponds to 5th order (83 Hz); (B) FFT of the discharge pressure pulsation at compressor outlet (9000 RPM): primary harmonic corresponds to 4th order (600 Hz).

The model also captures the instantaneous pressure build-up within each chamber of the compressor. Shown in Figure 12 are the chamber pressures in the two chambers opposite to one of the double-acting pistons. As expected, the pressure trace is identical but phased by 180 degrees, as one chamber undergoes suction while the other undergoes compression.

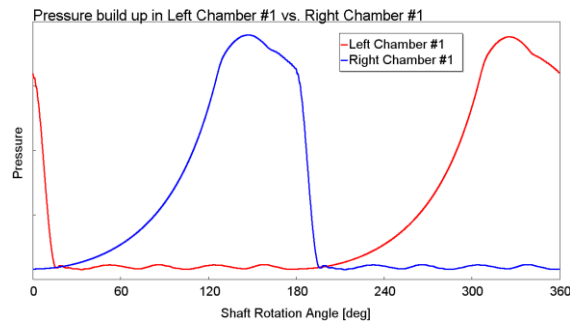


Figure 12: Pressure comparisons between the two chambers on either side of Piston 1 over one cycle.

In reality, compressors are tested under varying speed conditions. The model can be used to study the compressor performance under speed ramp-up and ramp-down conditions as well. Figure 13 demonstrates such a case where the mass flow rate predicted at the outlet of the compressor increases as the compressor speed increases. This correlates well in comparison to the expected relationship between mass flow rate and the compressor speed.



Figure 13: Compressor refrigerant mass flow rate predictions under transient speed ramp operation.

The results discussed thus far require only the 1D flow model. However, this provides only the fluid torque component in the compressor torque breakdown. To predict the required compressor torque accurately, the frictional power losses at the shoe-swashplate contacts must be found as these contribute toward the bulk of the friction torque. The 3D multi-body dynamics model predicts this contact friction loss at the spherical-plane Hertzian contact. Figure 14-A shows the Hertzian contact load at one of these contacts between Piston 1 and the swashplate. As expected, this force follows the same trend as the corresponding chamber pressure profile. The combination of all contact forces on one side of the swashplate result in five pulsations as expected (Figure 14-B).

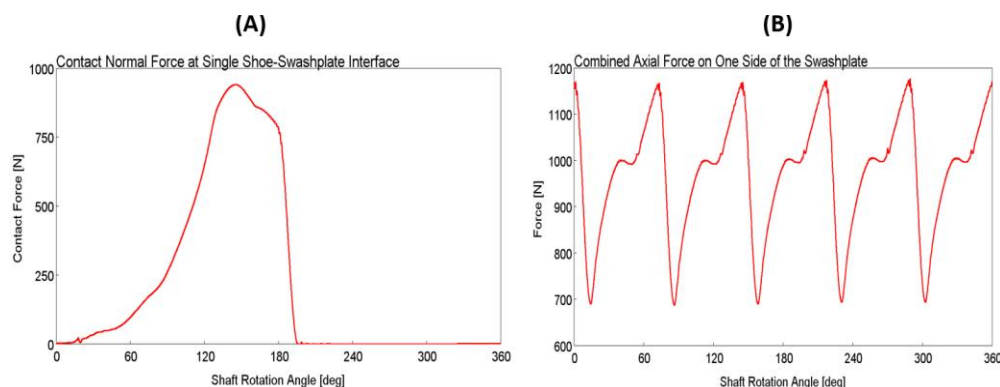


Figure 14: Axial Contact Force acting on one side of the swashplate showing (A) the influence of a single chamber and (B) the influence of 5 chambers, at 2000 RPM.

The Hertzian normal contact loads at the shoe-swashplate interfaces now result in contact friction. These friction forces and corresponding power losses are shown in Figure 15. These are illustrated for the shoe-swashplate contact interfaces on either side of Piston 1.

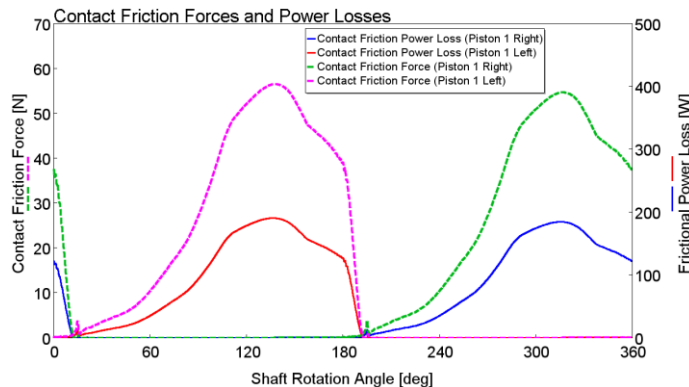


Figure 15: Contact Friction Forces and Power Losses at the Shoe-Swashplate contact interfaces for Piston 1.

The resulting compressor torque can now be evaluated and is compared against test data in Figure 16 for two speeds. The model predicts the torque requirement to within 5-7% of the test data. The difference can be attributed to the viscous power losses due to the operation of the journal bearings and thrust bearings present in the actual compressor which are not modeled in this study.

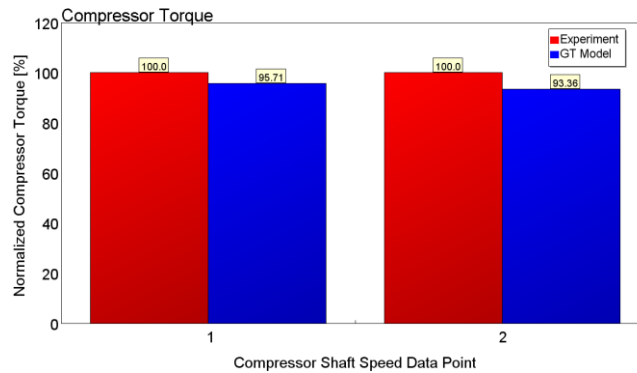


Figure 16: Comparison of compressor torque requirements between test and model at two speeds.

7. CONCLUSIONS

This paper describes a novel one-way coupled FSI approach to model a double-acting fixed displacement swashplate compressor by combining a predictive 1D flow model with a 3D multi-body dynamics model of the compressor. The model building process has been described and the predicted results such as refrigerant flow rates and pressure pulsation behavior have been validated against historical test data. Some interesting results shown included the change in the order of the primary harmonic of the discharge pressure with the compressor shaft speed, as well as the possibility of studying the compressor performance in transient runs vis-à-vis real automotive driving cycles. The multi-body dynamics model was able to predict the friction losses at the contact interfaces of the compressor as well as provide a reasonable prediction of the compressor torque. The model can now be used to perform design studies for optimizing compressor performance. Future work will involve modeling the reed valve dynamics at the compressor discharge ports in greater detail and modeling the power losses due to the operation of the bearings.

REFERENCES

- Dhar, S., Ding, H., and Lacerda, J. (2016). A 3-D Transient CFD Model of a Reciprocating Piston Compressor with Dynamic Port Flip Valves. *23rd International Compressor Engineering Conference at Purdue University, July 11-14*
- Gimelli, A., Rapicano, A., Barba, F., and Pennacchia, O. (2012). Reciprocating Compressor 1D Thermofluid Dynamic Simulation: Problems and Comparison with Experimental Data. *International Journal of Rotating Machinery, 2012*
- Kacani, V. (2016). Systematic Calculation of the Piston Rod Unit. *23rd International Compressor Engineering Conference at Purdue University, July 11-14*
- Lemke, N. C., Konig, M., Hennig, J., Forsterling, S., and Kohler, J. (2016). Transient Experimental and 3D-FSI Investigation of Flapper Valve Dynamics for Refrigerant Compressors. *23rd International Compressor Engineering Conference at Purdue University, July 11-14*
- Liu, Y. G., and Hui, G. (2012). Motion Simulation and Dynamic Analysis of a Four-Cylinder Wobble-Plate Compressor. *Proceedings of 2012 International Conference on Mechanical Engineering and Material Science at Yangzhou, China, December 16-18*
- Srivastava, P., Sharma, K., and Jha, R. (2016). Modeling of Fixed Displacement Swash Plate Compressor: A Mathematical Approach towards Calculation of Shaft Torque and Volumetric Efficiency. *SAE Technical Paper 2016-28-0172*

Sun, S. Y., and Ren, T. R. (1995). New Method of Thermodynamic Computation For A Reciprocating Compressor: Computer Simulation of Working Process. *International Journal of Mechanical Science*, 37(4), 343-353.

Xie, F., Nieter, J., Lifson, A., Reba, R., and Sishtla, V. (2017). Fully coupled fluid-structure interaction model of reed valves in a multi-cylinder reciprocating piston compressor. *10th International Conference on Compressors and their Systems at City University London, IOP Conf. Series: Material Science & Engineering*, 232, September 11-13

Xie, G., and Bansal, P. K. (2000). Dynamic Simulation Model of a Reciprocating Compressor in a Refrigerator. *15th International Compressor Engineering Conference at Purdue University*, July 25-28

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