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A 3-D Transient CFD Model of a Reciprocating Piston Compressor with Dynamic Port Flip Valves

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ABSTRACT

In this paper a full transient 3-D CFD model of a complete single piston compressor system along with dynamic inlet and outlet port flip valves is presented. The model includes the motion of the cam driven piston, and also predicts the motion of the dynamic flip valves. All activities related to model creation, simulation and post-processing were performed within the simulation software Simerics–PD (also known as PumpLinx).

The mesh creation and mesh motion for the piston chamber and the flip valve volumes were performed using meshing algorithms implemented within the simulation software. In addition, a novel mesh motion algorithm was implemented to enable a fully coupled simulation of the inlet port flip valve and the piston surface: which have intersecting swept volumes and may interfere during the operation of the compressor. The entire simulation was performed in a fully coupled manner in which the motion of the valves is calculated within Simerics–PD during the transient CFD simulation. This is achieved by a Fluid Structure Interaction (FSI) methodology where the pressure distribution acting on the valve is used to calculate the valve motion using a robust ordinary differential equation (ODE) solver. Important input parameters of the rotational dynamics (such as torsion constant) of the flip valve were calculated from the beam bending theory. In addition a Full Closure model for flip valves implemented within Simerics–PD ensures that there is no leakage through the valves when they are in the fully closed position, which are essential for accurate compressor simulation.

In the present work, detailed descriptions are provided of the model creation and simulation methodology for an industrial single piston compressor system. The model creation was performed on the real world geometry without simplifications and real-gas fluid properties were used for the simulation. The fully coupled 3-D transient CFD simulation of the piston compressor and the flip valves is shown to have excellent simulation speed and several relevant results are presented. The model set-up and performance of the simulation are demonstrated to be robust and user friendly and the algorithms developed can be readily applied to single piston compressor systems with dynamic port valves.

1. INTRODUCTION

Single piston compressors are popular devices used in refrigerators and air conditioners. While known to be high efficiency machines, international regulations have created a need for the efficiency and performance of such machines to be further improved. Traditional methods of improving the design and efficiency of such machines depend on a trial and error approach in which several iterations of design improvement and experimental testing are performed. However, such a method can be expensive as well as time-consuming and recent effort to minimize the environmental impact of refrigerators and air conditioners as well as an increase in available computing power has led to a focus on computational techniques to improve the design of such machines. Most of the effort in the modeling of reciprocating compressor systems has been using one or zero dimensional models. Soedel (1972) and Ussyk (1984) studied the reciprocating compressor using an integral formulation to represent the cylinder fluid
properties. The valves dynamics were calculated using free modes superposition and the mass flow through the valves using discharge coefficients obtained experimentally. Further improvements in representing the valve movement was conducted by Lopes (1996) and Matos et al. (2000, 2006) modeling a radial diffuser as a one-degree of freedom mass-spring model and the fluid domain with a two-dimensional compressible finite volume methodology also including turbulence effects. The suction reed valve, in particular, has been studied in individual experimental work such as Arantes et al. (2013) as well as computational work like by Bhakta et al. (2012), Suh et al. (2006) and Nagy et al. (2008). The valves have also been studied using an immersed boundary method approach by Lacerda and Gasche (2010), Rodrigues and Gasche (2011) and Barbi et al. (2012) where a virtual physical model was created using a two-dimensional fixed walls radial diffuser, a two-dimensional moving wall radial diffuser and three-dimensional moving reed valve.

However, since the compressor and port valves interact with each other during its operation and the valve dynamics have a large role to the play in the efficiency and performance of the compressor system, to have a useful simulation model the piston compressor and the valves should be simulated in a fully coupled manner. Very few such CFD models of a combined piston compressor valve system exist in literature. In 2006, Birari et al. published a 3D CFD model of a reciprocating compressor, without considering the inlet or the exit valves or their dynamics. Pereira et al. (2010) presented a 2D axisymmetric model of the compressor along with suction and discharge valves, and in 2012 the same group published a 3D CFD model of a simplified compressor with only the suction valve modeled in a simplified form. In 2010, Kinjo et al published a CFD model of a reciprocating compressor where both the suction and discharge valves were modeled dynamically. However, the geometry for this problem was simplified to a 2D approximation of the real geometry.

While the above studies approach a complete high fidelity model of the reciprocating compressor system, all of them have significant approximations which limit their usability to study real-world industrial problems. In addition, the CFD setup for reciprocating compressor simulations can be quite involved using complex mesh motion algorithms as well as variable time step models in order to preserve the stability of the simulation which makes them quite cumbersome for CFD users. Over the past several years the commercial CFD software Simerics-PD (also known as PumpLinx) has been developed keeping in mind positive displacement (PD) fluid machines such as pumps and compressors, and several 3D CFD transient models of PD compressors have been published over the last few years such as scroll compressors (Gao and Jiang, 2014), rolling piston compressors by (Ding and Gao, 2014) and twin screw compressors by (Kovacevic et al., 2014). The present work addresses the two issues discussed above by implementing a template based 3D CFD meshing and solution approach for reciprocating compressors. In order to be useful for industrial compressor design, the present work considers the entire compressor system with full geometric fidelity, and both port valves are modeled dynamically using a fluid structure interaction (FSI) algorithm. The model formulation, meshing technique and simulation results will be discussed in the following sections.

Figure 1: Scheme of a reciprocating compressor
2. RECIPROCATING COMPRESSOR MODEL

In the present section the working principle of the full reciprocating compressor will be described and details of various steps involved in setting up the model such as mesh creation and calculating dynamic parameters for the valves will be discussed.

2.1 Working Principle of the Reciprocating Compressor

The reciprocating compressor typically has three individual moving parts: the piston which has a well-defined prescribed kinematic, and the inlet and the outlet valves which open dynamically based on the pressure forces acting on them. Reciprocating compressor volumetric and energetic efficiency is very dependent on the valve system design. Usually, it uses a reed valve whose dynamics are controlled by the balance between the opening force caused by the pressure difference upstream and downstream the valve and the closing force, like a spring restitution, due to its deflection. A good valve design should have quick response, high mass flow, low pressure drop, avoid backflow and be reliable during extreme operation conditions. Since the flow field affects the structure of the valve and vice-versa, it is considered a fluid-structure interaction problem. Besides the valve system with the suction and discharge valves, the reciprocating compressor is composed by a cylinder, a suction line and a discharge line, as shown in Fig. 1. The cylinder volume variation is due the linear displacement of a piston connected in a crank slider mechanism moved by an electrical motor. The compressor efficiency is the ratio between the cooling capacity and the consumption energy necessary to perform the work.

2.2 Model Creation

The CFD model of the compressor is created starting from the complete solid CAD of the compressor, including all inlet and outlet porting as well as real valve geometry as shown in Fig. 2. The fluid volume was extracted, and the resultant volumes were used for mesh creation.

Figure 2: The CFD model is created based on the solid CAD model of the compressor (top). The extracted fluid volume includes the suction and the discharge valves (shown in green and red).

A view of the entire domain mesh is shown in Fig. 3. For the ports, an unstructured 3D hex-dominant Cartesian mesh was created using Simerics Binary Tree mesh generator. Further details follow about the mesh of the outlet valve, as well as the meshes of the suction valve and the piston chamber.

Figure 3: A view of the mesh for the full piston compressor model.
2.3 Piston Compressor, Suction and Discharge Valve Mesh

The discharge valve mesh was created using the specialized Simerics valve template mesher previously described in Ding and Gao (2014). The template mesher creates a structured mesh considering the possible path of motion of the valve, and during simulation the mesh is deformed appropriately to account for the valve motion as shown in Fig. 4.

![Fig. 4](image1.png)

**Figure 4:** A view of a section through the discharge valve with the valve (in red) at different positions of opening

In reciprocating compressors the swept volumes of the suction valve and the piston head intersect since the suction valve opens into the compression chamber. In order to model this correctly a mesh deformation algorithm was implemented within Simerics-PD. The initial mesh for the valve and the remainder of the chamber volumes were created separately and connected using an implicit mesh interface. During the operation of the compressor the piston motion is well defined, but the valve motion is determined by the dynamics of the machine. In order to ensure that the both motions are captured accurately two mesh deformation zones are defined. These are shown in Fig. 5: Zone1 deformation captures the valve motion, while Zone 2 deformation captures the motion of the piston. The location of separation between the two zones is represented by the yellow line and this separation location is updated automatically during the simulation based on the separation between the suction valve and the piston.

![Fig. 5](image2.png)

**Figure 5:** A section view through the combined mesh of the suction valve and piston compressor chamber. The mesh deformation algorithm is demonstrated.
In Fig. 5a the valve is in the closed position while the piston is close to the bottom dead center. Through Fig. 5b to d the gap between the piston and the valve starts to reduce but the mesh separation into zones ensures that both motions are captured fully. Finally in Fig. 5e the piston is essentially at the top dead center, with the valve fully closed and the gap between the piston and the valve is of the order of microns. Thus, it can be seen that due to the mesh deformation algorithm implemented for reciprocating compressors within Simerics-PD, the intersection of the swept volumes of the suction valve and the piston can be modeled. It must be noted at this point that the initial mesh created is only deformed to simulate the machine and that there is no additional re-meshing step which would add to the simulation time and reduce simulation efficiency.

![Diagram](image.png)

**Figure 6:** A) Suction valve geometry and B) rotation vs bending of the reed valve

### 2.4 Discharge and Suction Valve Dynamics

The suction reed valve is shown in Fig. 6: it is made of a thin sheet of metal with one end fixed and the other end free. The valve will bend under fluid force, and create an opening for the fluid to flow. Due to strong interaction force and small inertia of the reed valve the coupling between the piston compressor and reed valve becomes a very stiff FSI system to solve. A semi-implicit proprietary procedure is applied to eliminate the requirement of extremely small time steps of explicit coupling methods. In the present work both the suction and the discharge valve are modeled as reed valves, where the opening of the valve is captured using an equivalent rigid body rotation for the bending motion.

For the valves the dynamics are solved using a torsional mass and spring system using the following ODE:

$$I \frac{d^2 \theta}{dt^2} + C \frac{d\theta}{dt} + k\theta = \tau(t)$$

Where $\theta$ is the valve opening angle starting from the valve closed position, $I$ is the moment of inertia, $C$ is the rotational friction, $k$ is the coefficient of torsion spring, $\tau$ is the torque from fluid forces, and $t$ is time. For the suction valve the values of $I$ and $k$ need to be chosen based to obtaining a equivalent solution to the analytical solution for cantilever beam bending as in Ding and Gao (2014). This is a reasonable simplification, because for a reed valve that covers a small port opening, we can assume it to be a cantilever beam with a concentrated load at the center of the opening. In this case, the deflection at the opening is given by

$$y = \frac{P a^3}{3 E I}$$

where $P$ is the load, $a$ is the length from fixed end to the center of the opening, $E$ is the modulus of elasticity, and $I$ is the area moment of inertia of the reed valve. From Fig 6b, we can see that $\theta \approx \frac{y}{a}$ and the torque from the load at the opening can be calculated by $T = Pa$, finally leading to

$$k = \frac{3 E I}{a}$$

This value, in addition to the moment of inertia of the valve (can be calculated from any CAD package) can be then used in Eq.1. By using the above method, with carefully chosen dynamic parameters of the flip valve, the port opening center can have the same deflection as a bending valve under the same load.

For the discharge valve a similar procedure is followed, but with some differences since the discharge valve is fixed at two ends due the presence of a retainer, as shown in Fig.7. Therefore, for this valve the values for Eq.1 are chosen carefully to correspond with the analytical solution of a beam supported at both ends, rather than the cantilever beam bending solution.
As for the suction valve, the valve opening area is quite small, and the force on the valve can be approximated as a point load $P$ acting through the center of the valve opening area at a distance $a$ from one end as shown in Fig. 7b. The resultant deformation at the point of load application is

$$y = \frac{Pba}{6EI} (l^2 - a^2 - b^2)$$  \hfill (4)

where $l$ is the total length of the valve, and $b = l - a$. Since the opening is quite small, $\theta \approx \frac{y}{a}$ and the torque from the load at the opening can be calculated by $T = Pa$, and $k$ for Eq.1 can be written as $k = \frac{T}{\theta}$ which leads to,

$$k = \frac{6EIb^2}{a^2(l-a)}$$  \hfill (5)

As before, this calculated value along with the moment of inertia of the valve can be then used in Eq.1 to provide a realistic dynamic simulation of the valve where the opening of the reed valve along the line of load application is same as that of the bending valve.

### 3. CASE STUDY

An industrial reciprocating piston compressor from Tecumseh Products Company with both port valves was simulated to demonstrate the capability of the approach described in the previous sections. The simulation was performed using the real gas properties of the refrigerant R134a. Conservation equations of mass, momentum, and energy of a compressible fluid were solved using a finite volume approach. The standard $k - \varepsilon$ two-equation model (Launder and Spalding, 1974) was used to account for turbulence. In addition all the properties of the R134a: density, viscosity, heat capacity and conductivity are considered as a function of pressure and temperature. The equation for the kinematics of the piston was set as follows:

$$x = -\sqrt{(r1 + r2)^2 - e^2 - r1 \cos(\omega t) - \sqrt{(r2^2 - (r1 \sin(\omega t) - e)^2}}$$  \hfill (6)

Where $r1$ is the crank eccentric length, $r2$ is the con-rod length and $e$ is the eccentric diameter. The rotational speed of the piston $\omega$, is set as 3600 RPM. The boundary conditions of the model were set as shown in Fig.8.
A transient simulation was performed and the pressure inside the compressor at different crank angle is shown in Fig. 9 with the help of two section planes which pass through the suction and the discharge valves. It can be seen that at 0°, the piston is at bottom dead center and both the suction valve is still slightly open (but just about to close) and the discharge valve is shut. At close to top dead center 165° the chamber is at high pressure and the discharge valve is open. As the piston goes through its suction stroke at 270° and 300° the pressure in the chamber falls and the suction valve opens. A similar view of the temperature inside the compressor is shown in Fig. 10. It can be generally seen that as the piston gets closer to the top dead center, the temperature in the chamber rises, while the temperature remains low during the suction stroke.

The pressure-volume (P-V) diagram for the compressor is shown in Fig. 11. The suction and the discharge pressures along with the overpressure and the underpressure are also represented. The underpressure and overpressure areas represent the spent energy in the suction and discharge process respectively, while the P-V diagram can be used to find the efficiency and the power consumption of the compressor. At piston TDC the gap between the piston and the cylinder head is 100 microns, and this small dead volume is reflected on the P-V diagram.
The motion of the suction and the discharge valves as predicted by the model (according to the dynamic force balance in Eq.1) along with the chamber volume are plotted in Fig.12. As can be seen, the results are periodic between the cycles. The trends which were observed in Fig. 9 and 10 can be examined now in more detail and it can be seen that the outlet valve opens for around last 30 degrees of the compression half of the cycle, and is closed by the time the suction stroke starts. During the suction stroke, the suction valve opens about 40 degrees of crank rotation after the top dead center. The suction valve exhibits a valve flutter during the suction stroke, and closes about 30 degrees into the compression stroke. In general it was seen that due to the presence of the retainer the discharge valve shows a motion with much higher stiffness, which makes it open lesser than the suction valve and also closes faster. The suction valve has a much lower stiffness, and therefore can open a much higher degree, and also exhibits a slow valve oscillation.

Finally, the discharge and the suction flow rates are plotted in Fig. 13. As expected the flow rates correspond very closely with the valve opening angles with the discharge flow rates showing up in the form of a sharp spike, whereas the suction flow rates are more evenly distributed along the suction stroke. Another important point to note is that the flow rates go to zero when the valves are closed, since the simulation is able to handle valve full closure with essentially zero gaps between the valves and their seats when they are predicted to be fully closed by the dynamics.
Starting from a properly prepared CAD geometry, the meshing and the setup of the simulation takes less than an hour with the help of the Valve templates within Simerics-PD. Simulation time is about 3 hours for one compressor cycle on a standard 8GB Windows laptop with quad-core 2.7GHz Intel i7 processor. Simulation results were seen to be periodic within 3 cycles of the piston.

4. SUMMARY AND CONCLUSIONS

A novel 3D CFD, transient model of a reciprocating single piston compressor was presented in this work. The compressor simulation included fully coupled FSI simulations of the suction and the discharge flip valves. A methodology for creating the single piston compressor model has been outlined. In particular, the meshing of the discharge valve using the Valve template mesh, and the mesh deformation algorithm used to resolve the suction valve piston swept volume interference has been detailed. The procedure to calculate the parameters for the dynamic FSI simulation of both the valves is also outlined. In the simulation all the geometrical details of the compressor are captured and the physical system is modeled accurately without any simplifications unlike in 0D/1D models. In addition, in the current approach the dynamics of the flip valves take into account the accurate pressure distribution on the valve surfaces which are expected to provide much greater fidelity for the valve motion prediction as well as compressor performance and efficiency prediction. The developed methodology was demonstrated on an industrial Tecumseh reciprocating compressor, and the simulation results and the predicted valve motions were found to be physically realistic. Due to the template based meshing techniques and the robustness of flow solver, the setup and simulation is easy and fast. The ease-of-use, quick simulation turn-around time, and valuable information that can be extracted from such a transient 3D simulation makes this novel model a useful tool for the design and analysis of reciprocating single piston compressor systems.

5. REFERENCES


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