Fully coupled fluid-structure interaction model of reed valves in a multi-cylinder reciprocating piston compressor

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Abstract. For years compressor researchers have tried to account for the fluid interaction effect of the working fluid on valve motion in displacement compressors. In recent years, the computing capacities and available CFD and FEA programs have allowed fully coupled interaction of fluids and moving structures to be modelled more comprehensively. This paper describes our experience and results from developing a model of a multi-cylinder reciprocating piston compressor with suction and discharge valve systems that are fully coupled with the pressure pulsation in the adjacent plenum. Valve dynamics are captured by the model that affects compressor performance. The results show that higher running speed causes more discharge valve delay on closing due to higher pressure pulsation in discharge plenum. The acoustic property of the discharge plenum as it relates to valve motion is studied by the developed cost-effective standalone model.

1. Introduction
The main components of a reciprocating compressor that affect thermodynamics and flow are the cylinder, piston, connecting rods, valve plate, valves, and cylinder head. Valve behaviour controls the suction and discharge process and may have a substantial impact on coefficient of performance (COP) for refrigerant applications, which is defined by the ratio of refrigerating capacity to power consumption. Consequently, the numerical study of valve behaviour has become an important area in reciprocating compressor design, and has been widely researched in recent years. In this study, we model coupling effect of the reed valve dynamics and gas pulsations in the adjacent plenum for R410a 3 cylinder compressor. Another issue affecting compressor performance is the reed valve leakage, and Rezende [1] and Silva [2] developed numerical tools to predict the leakage gap of the compressor valve. The deformation of the lubricating oil film between the valve and the valve seat causes the stiction effect, which generates the valve delay. Rodrigo [3] developed a theoretical model for solving the dynamic behaviour of a ring-shaped lubricant oil film between a discharge valve and the seat.

Valve dynamics is very important in affecting mass flow rate through the ports. Discharge valve closing delay causes the backflow from discharge plenum to the cylinder, and the suction valve closing delay pushes the compressed gas to flow back to the suction plenum. Delay in discharge valve closing for example effectively increases the compressor clearance volume, as the start of re-expansion process is delayed. Both suction valve delay and discharge valve delay decrease the mass flow rate (capacity), and consequently result in lower COP. The study of valve dynamics is critical to design of a reciprocating compressor, and it involves the complex interaction between fluid and structural dynamics. Unfortunately, the small size and harsh environment inside a compressor, coupled with the material
compatibility of sensors, makes the experimental study [4,5,6] of valve dynamics very difficult, and the numerical study becomes a more viable option. Further, modelling results often provide more descriptive details and understanding of the physical valve and fluid behaviour that is difficult or impossible to obtain from experimental testing. By coupling one-dimensional (1D) lumped structural model with 1D unsteady fluid model, Boswirth [7] developed an efficient tool, KV-DYN, to calculate the valve lift, exit velocity and pressure difference through valves. The late closing of suction valve and pressure pulsation in suction plenum were observed, meanwhile the mechanism of late closing due to pulsations in the plenum was also illustrated by Boswirth [20].

With the advancement of high performance computing and computer clusters, the computational cost of complex models becomes acceptable, and the complicated coupling models needed to study the valve dynamics can be developed. The coupling of the three dimensional CFD solver and 1D lumped structural model [8,9,10] was developed for better prediction of flow field. Mayer [11], Ajay [12] and Gasche [13] constructed a fully coupled three dimensional (3D) CFD/FEA models and compared them with 1D/lumped models. The results showed that the 1D models needed to be calibrated by the fully coupled 3D CFD/FEA models. Barbi [14] utilized an IBM (Immersed Boundary method) to handle the moving valve boundary in his work. David [15] adopted the Cartesian Cut-Cell Method to generate the deforming mesh of the valves. More complicated CFD models by Estruch [16] and Gonzales [17] were applied to the coupling with structural models. In order to shift the resonance frequency of the plenum away from running operation, recently more researchers [18,19] started to evaluate the effect of pressure pulsation in the plenum on compressor vibration and noise, where typically the acoustic property of the three dimensional plenum was simplified as 1D impedance transfer matrix. However to date the authors are not aware of any CFD studies that coupled the pressure pulsation in the plenum with the valve dynamics. Furthermore, the prior studies of valve dynamics were mainly focused on the reciprocating compressor with a single cylinder [9, 16, 17], whereas this paper is focused on 3D simulation of a three-cylinder reciprocating compressor system where pulsation effects in the plenum are considered in conjunction with the fluid and valve dynamics.

In this study, the computational time of considering the fluid-structure interaction of coupling the fluid in all three cylinders with the movement of all reed valves was approximated to be at 15 days which is time prohibitive. Thus a three step sequential modelling approach was developed. First, the model was constructed having only one physical cylinder and one portion of the discharge plenum to analyse the compressor PV diagram. As the next step, to capture the effect of the plenum geometry on pressure pulsation in discharge plenum, the model was extended to include the full discharge plenum with the one physical cylinder. In reality three cylinders work together with a specified crank angle difference. Consequently, a third model was developed with the full discharge plenum connected to one physical cylinder and two virtual cylinders (the virtual cylinders are used to contribute mass flow from those cylinders to the pulsations in the full plenum and not for specifics of valve movement, whereas the one physical cylinder included the fully coupled valve movement to the pulsations in the plenum). The results and analysis in this paper are based on this third model where the fully coupled method of CFD/FEA is employed to study the reed valve dynamics in relation to the pressure pulsations in the plenum. This study concentrated on the behaviour of the discharge valve, as calculated results show a significant effect of pulsations in the compressor discharge plenum on the delay of the discharge valve closure and subsequent reduction in the mass flow rate through the compressor.

2. Methodology
The commercial software CFX and ANSYS Workbench are selected to solve the fluid and solid equations respectively, and the coupling between fluid and solid setting is executed in CFX-Pre by importing the mechanical output file. The flowchart is shown in Figure 1.
2.1. *Modeling of the fluid*

The fluid in the reciprocating compressor system is compressible where the governing equations of compressible flow include mass, momentum and energy balances, and the equation of state closes the system of equations. All equations are discretized and solved at the cell nodes under the finite volume method. High resolution is adopted for the advection scheme, and the K-Omega model is selected to simulate turbulence. A deforming mesh is set up in CFX Pre-processor to account for the changes in the volume of the fluid zones in the compression process. The current study contains 36 fluid zones, and the information communication between adjacent zones is necessary where the different interface conditions are carefully set. To investigate the influence of pressure pulsation in the discharge plenum on the valve behaviour, we assumed non-reflective boundary at the discharge plenum outlet (this is a common and appropriate boundary condition that avoids unwanted downstream system effects).

2.2. *Modeling of the solid*

Virtual work and virtual displacement theorem generates the structural dynamics equation based on nodal displacement. The assembled equation based on the finite element method is shown as follows:

\[ m \ddot{x}(t) + c \dot{x}(t) + k x(t) = f \tag{1} \]

\(x(t)\): nodal displacement matrix, \(m\): mass matrix, \(c\): damping matrix, \(k\): stiffness matrix, \(f\): external force matrix. With the specified boundary conditions, the nodal displacement can be solved, and then the stress and strain calculated. In the current study, the non-linear contact is considered when valves impact the seat or backer.

2.3. *The coupling method*

The mechanical analysis in ANSYS Workbench generates the output file which is imported into CFX-Pre, and all FSI interfaces are shown in the CFX-Pre boundary condition setting cards. The communication property of interfaces between fluid and solid can be set up in CFX-Pre.

3. *Case Definition*

The main components of the reciprocating compressor are the housing, cylinders, connecting rods, pistons, valve plate, valves, and cylinder head, along with bolts and gaskets. The current compressor model is composed of three cylinders. The common cylinder head contains three connected chambers within the suction plenum and three connected chambers within the discharge plenum. The suction valves and discharge valves are reed type valves with one end fixed on the valve plate. The retainer (backer) is designed to limit the maximum displacement of the discharge valves. The properties of the working fluid (refrigerant) are dependent on pressure and temperature, and can be interpolated from the property table. The motion of the piston is pre-defined by the driveline geometry and running speed. Figure 2 shows the structure of the compressor in this study.
3.1. The fully coupled model

The fluid zone starts from the entrance to the suction plenum and ends at the exit of the discharge plenum. The emphasis of this study is on the valve behaviour and pressure pulsation in discharge plenum. The compressor contains three cylinders with identical sets of discharge valves and suction valves. Each cylinder has three individual sets of suction and discharge ports covered by reed valves. The middle cylinder and associated valves are selected for fully coupled physical modelling, while the behaviour of the cylinders and valved ports on either side of the middle cylinder are simplified using the enforced mass flow rate which was calculated by a separate single cylinder model having the same plenum. The fluid zone and solid zone for the fully coupled simulation are shown in Figure 3. One section is sliced from the model in Figure 3, and displayed in Figure 4 where more details of the model are shown.
3.2. The standalone model
In the fully coupled model, both fluid zone and structural zone are discretized and solved. This method has a relatively high computational cost. Since our main interest here is to evaluate the discharge valve behaviour coupled to pressure pulsation response in the discharge plenum, we also developed a standalone model consisting of only the discharge plenum as shown in Figure 5. The mass flow through the discharge ports used by this standalone model is calculated by the fully coupled model, and the discharge plenum is treated as the only fluid zone. Consequently there is no moving solid zone in the standalone discharge plenum model, and investigating plenum geometry changes can be more easily accomplished. The fluid zone of standalone model is shown on the right in Figure 5, and the mass flow rates of all nine ports (3 per cylinder for 3 cylinders) are shown on the left.

![Figure 5. Standalone model of discharge plenum.](image)

4. Results and Discussion

4.1. Results calculated by fully coupled model
We consider several “monitoring” points, where we extract pressure or valve displacement data from the model. One pressure monitoring point is located in the cylinder, and one pressure monitoring point is located in the discharge plenum. Two displacement monitoring points are located on the discharge and suction valves respectively to determine valve movement. All monitoring points are shown in Figure 6 as indicated by probe locations on the left and P1 on the right.

![Figure 6. The location of monitoring points](image)

The cases of fully coupled model were run on one computing node with 28 cores (CPU type: Intel E5-2680v4, 14 cores for fluid dynamics and 14 cores for structure dynamics) of Linux cluster. The high resolution scheme is employed for spatial accuracy. The time step size can capture fluctuations up to 8000 Hz, and the spatial size guarantees there are at least ten cells within the shortest wave length. One operational cycle took 3.5 days. We considered two cases, with the running speed of the compressor at rotating speeds of 29 Hz and 40 Hz, which represents a typical lower and higher end of the operating range for this compressor. For the first case of 29 Hz, Figure 7 shows the cylinder pressure versus cylinder volume (PV diagram).
The Fourier transform of discharge plenum pressure pulsations for this first case is shown in Figure 8 where two significant peaks at 87 Hz and 554 Hz are observed. These two peaks are located at three times and nineteen times of the basic running speed. The first peak is a result of the superposition of the basic excitation frequency from three cylinders (3x29), and the second frequency is the result of fluid acoustic resonance in the discharge plenum.

Figure 9 shows the pressure in cylinder and discharge plenum versus CA (crank angle), along with the discharge valve displacement. Pressure labelled as Pres_Disc, represents pressure pulsations in the plenum. Pressure in the cylinder, labelled as Pres_Cyl, represents pressure during compression, discharge, expansion, and suction processes. For the ideal case there should be no valve closing delay with respect to the piston at the top dead center (TDC) position. However, the current simulation shows phase difference between the TDC and the discharge valve closing. From the data shown in Figure 9 and the details in Figure 10, point A is at the TDC location where the discharge valve displacement is not zero yet but is still nearly 0.2mm from the valve seat. Point B is the location where the discharge valve finally closes and 5 degrees of phase difference (delay) can be calculated. In addition to substantial delay in valve closing due to pulsations in the plenum (discussed in details later in the paper), we can also observe additional delay due to the valve bounce between point A and B, which is initiated when the valve first contacts the seat. At crank angles beyond point B we can also observe a discharge valve deflection into the port caused by increasing pressure differential across the valve during the re-expansion process. It should be noted that a small delay in the discharge valve closing causes much more substantial delay in the suction valve opening (as the re-expansion process would also be delayed). This delay in the suction valve opening results in reduced compressor mass flow as the start of the suction process is delayed. Reduction in the mass flow rate results in the loss of system capacity and COP. As indicated earlier, to save on computational time, we only analysed the discharge valve movement for the middle cylinder (physical cylinder) and the other two cylinders are only considered as sources of mass flow (virtual cylinders) to the plenum. The mass flow rate through the middle discharge port labelled by Cyl2_2 in Figure 5 for the simulated middle physical cylinder is shown in Figure 11 and Figure 12 (showing local details of Figure 11 near TDC). Because of the delayed discharge valve closing we can observe the negative (backward) flow into the cylinder from the plenum.
In order to study the effect of running speed on the valve delay, we also performed a simulation for a second case of running the compressor at higher speed of 40 Hz. Figure 13 and Figure 14 (expanded with local details) compare cylinder pressure versus cylinder volume between running speeds at 29 Hz and 40 Hz.

Figure 15 shows the discharge plenum pressure amplitude versus frequency for the 40 Hz case, where two peaks can be observed at the frequencies of 120 Hz and 560 Hz. Similar to the case of 29 Hz, the first frequency is three times the basic running speed, generated by superimposing the motion of the three cylinders (3x40 Hz). The second frequency is fourteen times the basic running speed, and is the result of fluid resonance in the plenum. It is noted that the second frequency for the 40 Hz running speed case is quite close to the second frequency observed for the 29 Hz running speed case. Both the frequency at 554 Hz for 29 Hz running speed and 560 Hz for 40 Hz running speed are due to the resonant frequency of the discharge plenum (small variations in resonant frequency can be expected due to
changes in fluid properties and changes in the frequency of the forced pulsations). Figure 16 shows the pressure and valve displacement versus CA (crank angle) for 40 Hz operation. The closing delay of the discharge valve at 40 Hz is now 9.5 degree as compared to only 5 degrees for 29 Hz operation, and details for discharge valve closing delay can be observed in Figure 17 for the 40 Hz speed. The increased valve closing delay between the case of 40 Hz vs. 29 Hz operation can be mainly attributed to larger pressure amplitude at the plenum resonant frequency in the discharge plenum (compare amplitude of 2.06 psi in Figure 8 vs. 3.46 psi in Figure 15). And to a smaller extend the delay can be due to a bounce of the valve on its seat due to impact (small peak near point B). If the “bounce” effect is subtracted from the delay, then the effect of pulsations alone on the discharge valve closing delay becomes even more pronounced (roughly 2 degrees for 29 Hz vs. 7 degrees for 40 Hz).

Figure 15. Plenum pressure vs frequency  
Figure 16. Pressure and displacement vs CA  
Figure 17. Pressure and displacement vs CA expanded  
Figure 18. Amplitude versus frequency

Results from 29 Hz and 40 Hz running speed simulation are summarized in Table 1. As stated earlier, in both cases there is a delay in the discharge valve closing. The case with higher speed generates more discharge valve closing delay resulting in more substantial decrease in mass flow rate as compared to no delay case. Reduction in mass flow rate causes subsequent drop in the cooling or heating capacity. One of the reasons for more pronounced discharge valve closing delay at 40 Hz vs 29 Hz operation is the higher pressure pulsation amplitude at plenum acoustic resonance frequency (near 560 Hz).

| running speed | first pressure pulsation peak (psi) | second pressure pulsation peak (psi) | discharge valve delay (degree) | re-expansion in cylinder | frequency peak (Hz) |
|---------------|-----------------------------------|------------------------------------|-------------------------------|-------------------------|--------------------|
| 29Hz          | 2.01                               | 2.06                               | 5                             | low                     | 87/554             |
| 40Hz          | 1.81                               | 3.46                               | 9.5                           | high                    | 120/560            |

Table 1 comparison between different running speeds
4.2. Results calculated by standalone model
To address the effects of pressure pulsations in the discharge plenum and its effect on discharge valve closing delay, a plenum standalone model is developed with the mass flow rate applied to the nine discharge ports. Since the purpose of the plenum standalone model is to only calculate pressure pulsation amplitude in the plenum, which is mainly affected by refrigerant mass flow rate from each port and not by intricacies of the valve movement, we do not need to employ the computationally intensive fully coupled FSI model for this task. The cases with the standalone model were run on the same computing resources as for the fully coupled model, as described above in Section 4.1, and one operation cycle takes only 10 minutes here as compared to the 3.5 days needed with the fully coupled model. In Figure 18 we see good agreement between the two models, which gives us confidence that the standalone model can be used for predicting pressure pulsations while providing a significant reduction in computation time. Figure 19 shows the pressure pulsation amplitude distribution in the discharge plenum at the 560 Hz frequency, which is fourteen times the running speed (14x40 Hz), and the amplitude in the middle chamber of the plenum has the largest amplitude. Figure 20 shows that the side chambers are 180 degrees out of phase with the middle chamber. This indicates that the pulsations in the discharge plenum at this frequency are a result of an acoustic plenum resonance.

![Figure 19. Amplitude distribution at 14X frequency](image1)

![Figure 20. Phase distribution at 14X frequency](image2)

5. Conclusion
This paper described a fully coupled CFD/FEA method to simulate the valve dynamics and pressure pulsations in a multi-cylinder reciprocating compressor. The results show that pressure pulsations in the discharge plenum significantly contribute to the delay in discharge valve closing after the cylinder reaches top dead center. Higher running speed generates higher pressure pulsations in the discharge plenum, causing more pronounced delay in discharge valve closing. This delay can significantly decrease compressor mass flow rate and reduce system cooling or heating capacity. It would also decrease compressor operating efficiency. This occurs because a delay in the discharge valve closing results in the delay in the suction valve opening, which reduces volumetric efficiency of the compressor. Acoustic resonance in the discharge plenum is the main contributor to pressure pulsations in the discharge plenum. To analyze the acoustic property of the discharge plenum, a computationally effective plenum standalone model was developed. The standalone model compares well with the fully coupled FSI model and can be used for efficient mitigation of pressure pulsations in the plenum.

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