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A non-linear reciprocating compressor model representing the interaction between thermodynamic process and unsteady flow

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Abstract

Accurate comprehension of the thermodynamic demeanor and pressure pulsation propagation is of great attractive in a reciprocating compressor plant. A numerical model has been built to consider the reciprocal interaction between compressor and pipelines. Simulation of the thermodynamic process is performed based on mass balance and first law of thermodynamics in the suction chamber, cylinder volume and discharge chamber, together with thermodynamic relationships. Pressure pulsations in the pipelines are predicted by solving one dimensional unsteady flow equations. Comparison between numerical results and previous experimental results encounters good agreement and the mutual influence between thermodynamic process and pressure pulsation is then highlighted by extending a further analysis of the compressor and the pipelines.

Keywords: Reciprocating compressor; pressure pulsation; thermodynamic cycle; valve dynamics;

1. Introduction

Reciprocating compressors are an important component in many gas industries. This specific type of machinery consuming large amount of energy must be designed with increasing efficiency and reliability to meet the higher worldwide request for energy conservation and emission reduction [1]. Availability of well-tuned models in the preliminary design process is critical to prevent compressor matching mistakes. Unfortunately, the coupling between compressor and the piping system brings a challenge for accurately predict the thermodynamics.

The compressor model based on three-dimensional computational fluid dynamics (3D CFD) can give the most detailed characterization of the machinery performance. Pereira et al. [2] presented a CFD analysis of reciprocating refrigeration compressor with the resolution domain including suction and discharge mufflers, cylinder and valves.

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Aigner [3] investigated the internal flow inside the cylinder based on the established quasi-2D and 3D numerical models by employing the simplified effective flow area within the valve flow and force. The 3D CFD model, however, cannot be suitable for specific goals due to the unacceptable computational time.

The lumped parameter models of reciprocating compressor are extensively applied for simulating the thermodynamic cycle [4], which suit most of the design demands for giving results efficiently and describing the performance globally. However, they poorly consider the interaction between the compressor and the connected pipelines. The necessity of examining pulsation effect on the thermodynamic process was proposed by Elson and Soedel [5]. Thus, the hybrid models were established by combining the lumped parameter models with the acoustic description of the pipeline system by four-pole method [6] for basic acoustic elements, or acoustic finite element method (FEM) characterization [7] for complex geometries. However， the hybrid models are limited in the event of acoustic resonant response because they neglect the second-order terms in the governing equations.

A promising compressor model is potentially obtained by simultaneously simulating the thermodynamic cycle of compressor and unsteady flow in the pipelines. Benson et al. [8] applied the Euler method to solve the equations of the first thermodynamic law and mass conservation for cylinder volume and the Method of Characteristics (MOC) to calculate the equations of unsteady flow in the pipelines. Liu et al. [9] developed a transient gas dynamic model for the simulation of pressure pulsation and the compressor performance. In these models, one-dimensional flow equations representing the flow through compressor valves were considered as boundary conditions for the flow from the suction pipe to cylinder or from the cylinder to discharge pipe. Thus, the suction and discharge chambers must be modelled as pipe elements. The more proper approximation of the suction/discharge chambers is promisingly treated as volumes since the chambers may be geometrically complex or with multi-port.

The objective of this work is to develop a model of reciprocating air compressor to examine the interaction between the compressor and its piping system. The LW2 scheme is employed to predict the unsteady flow. An integrated suction chamber-cylinder volume-discharge chamber (SCD) model is implemented based on conservation of mass and first law of thermodynamics. At each time step, the previous fluid properties in the node connecting suction/discharge chamber and pipe are applied as boundary conditions for the solution of the SCD model by performing the Trapezoidal version of the MOC (TMOC) developed by authors [9]. The compressor performance was assessed respect to valve displacement, in-cylinder pressure oscillations and p-θ diagrams.

2. The non-linear numerical model

2.1. SCD model

Fig. 1 shows a schematic diagram of a single-acting reciprocating compressor with suction and discharge pipes. The first law of thermodynamics and mass conservation are the base to examine thermodynamic properties. Both pressure and temperature are assumed as uniform in the control volumes. The momentum equation is neglected and these lumped open systems undergo a quasi-state process. Leakage is not yet considered and the ideal air is used.

The equation of mass conversation for the control volumes of Fig.1 can be written as follows:

$$\frac{dm_{cv}}{d\theta}=\frac{1}{\omega}\left(\frac{dm_{in}}{dt}-\frac{dm_{out}}{dt}\right) \tag{1}$$

where ω is angular speed, θ is crank angle, t is time, and dm/dt could be computed for compressor cylinder [4]:

$$\begin{aligned}
\frac{dm_{in}}{dt}=\frac{dm_s}{dt}&=\begin{cases}\varphi_{sv}A_{sv}\sqrt{2\left(p_s-p_{cv}\right)\rho_s} & \text{for } p_s>p_{cv}, y_s>0\\ -\varphi_{sv}A_{sv}\sqrt{2\left(p_{cv}-p_s\right)\rho_{cv}} & \text{for } p_s<p_{cv}, y_s>0\end{cases}\\
\frac{dm_{out}}{dt}=\frac{dm_d}{dt}&=\begin{cases}\varphi_{dv}A_{dv}\sqrt{2\left(p_{cv}-p_d\right)\rho_{cv}} & \text{for } p_{cv}>p_d, y_d>0\\ -\varphi_{dv}A_{dv}\sqrt{2\left(p_d-p_{cv}\right)\rho_d} & \text{for } p_{cv}<p_d, y_d>0\end{cases}
\end{aligned} \tag{2}$$

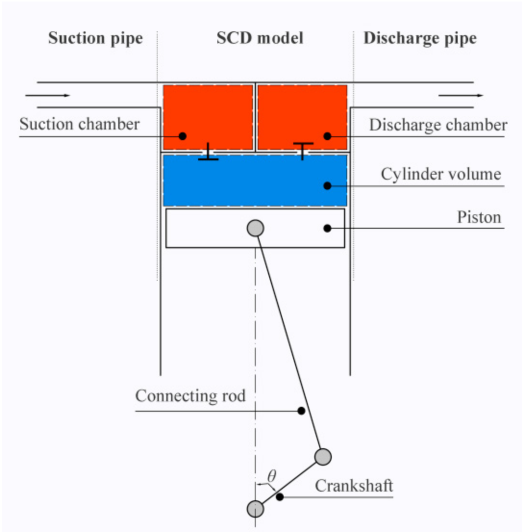

Fig. 1. Schematic of reciprocating compressor.

where A_{sv} and A_{dv} are the flow areas, and φ is the semi-empirical coefficient that considers the non-idealities of valves, and, y_s and y_d are the displacement from the orientation of the closed valve plate. For the suction chamber, subscript '*out*' is for suction valve and $dm_{out}/dt = dm_s/dt$, and '*in*' is for suction pipe:

$$\frac{\mathrm{d}m_{in}}{\mathrm{d}t} = \rho_{sp} u_{sp} A_{sp} \tag{3}$$

where subscript '*sp*' refers to the suction line. For the discharge chamber, subscript '*in*' is considered for the discharge valve and $dm_{in}/dt = dm_d/dt$, and '*out*' stands for discharge pipe in which mass flow rate is expressed as:

$$\frac{\mathrm{d}m_{out}}{\mathrm{d}t} = \rho_{dp} u_{dp} A_{dp} \tag{4}$$

where subscript '*dp*' refers to the discharge line.

The first law of thermodynamics for the energy balance over a control volume can be written as follow:

$$\frac{dT_{cv}}{d\theta} = \frac{1}{m_{cv}\left(c_{pcv} - R_g\right)}\left[\frac{dQ_{cv}}{d\theta} - \frac{dW_{cv}}{d\theta} + c_{pin}T_{in}\frac{dm_{in}}{d\theta} - c_{pout}T_{out}\frac{dm_{out}}{d\theta} - \left(c_{pcv} - R_g\right)T_{cv}\frac{dm_{cv}}{d\theta}\right] \tag{5}$$

in which, for compressor cylinder volume, the heat transfer rate $dQ/d\theta$ is computed in accordance with the method described by Liu et al. [9] and the work variation is $dW_{cv}/d\theta = p_{cv}\,dV_{cv}/d\theta$. For suction and discharge chambers, no heat transfer is considered $dQ/d\theta = 0$ and $dW_{cv}/d\theta = 0$ due to the fixed control volume.

The instantaneous volume of the cylinder from top dead centre is presented by:

$$V_c = V_{cl} + \frac{\pi D_c}{4} r_1 \left[1 - \cos\theta + \frac{r_2}{r_1}\left(1 - \sqrt{1 - \left(r_1/r_2\right)^2 \sin^2\theta}\right)\right] \tag{6}$$

where V_{cl}, r_1 and r_2 are clearance volume, crank radius and length of connecting rod respectively.

The valve dynamics is presented in a second-order ordinary differential form of the mass-spring-damper as below:

$$m_{eq}\omega^2 \frac{d^2 y}{d\theta^2} + c_{eq}\omega \frac{dy}{d\theta} + k_{eq} y = C_D A_D \Delta p + G_{init} \tag{7}$$

where m_{eq} is the equivalent mass of the valve, k_{eq} is the spring stiffness, $c_{eq} = 2\xi\sqrt{k_{eq} m_{eq}}$ is the damping coefficient which is often ignored due to its low magnitude and hardness to obtain, G_{init} is the pre-load force, C_D is the drag coefficient that can be obtained from previous subject and A_D is the valve plate area [3]. The pressure difference Δp is $p_s - p_{cv}$ for the suction valve and $p_{cv} - p_d$ for the discharge valve. A rebound coefficient of 0.3 is introduced to account for the impact effect of the collision between valve plate and valve limiter/seat.

2.2. Governing equations in the pipes

The gas dynamic models solve the one-dimensional non-homentropic unsteady flow accounting for the cross-sectional area change, friction and heat transfer processes. Thus, a non-homogeneous hyperbolic system formed by a conservative arrangement of the continuity, momentum and energy equations is shown as follow [10]:

$$\frac{\partial}{\partial t}\begin{bmatrix} \rho A \\ \rho u A \\ \rho e_0 A \end{bmatrix} + \frac{\partial}{\partial x}\begin{bmatrix} \rho u A \\ \left(\rho u^2 + p\right) A \\ \rho u h_0 A \end{bmatrix} + \begin{bmatrix} 0 \\ p\dfrac{dA}{dx} \\ 0 \end{bmatrix} + \begin{bmatrix} 0 \\ \rho G A \\ -\rho q A \end{bmatrix} = 0 \tag{8}$$

where G represents the friction term, q refers to the term of heat transfer and subscript '0' stands for the stagnation state. The closure of the conversation system is performed by the equation of perfect gas properties.

2.3. Numerical solution

The resolution of the numerical model follows a two-step numerical computation. The first step is solution of the pulsating flow field in the pipes. The flow properties of internal points are solved by employing the LW2 method and the flow properties at pipe end-points, except for the boundary conditions of the SCD model, are updated by application of the TMOC developed by authors. Then depending on the previous fluid properties of the boundary points connecting suction/discharge chamber and pipe, the thermodynamic conditions of the SCD model are solved by following a Trapezoidal integral procedure [9], which is in essence an implicit algorithm that must be calculated iteratively. Temperature and density in the control volumes are the two independent thermodynamic properties that are enough to pick out other thermodynamic properties. For each time step, these two properties are updated until the convergence condition. At every iteration, the valve dynamics computation are carried out by means of the 4th order Runge-Kutta method and the boundary conditions of the SCD model are also computed repeatedly by using the TMOC. The whole procedure is iterative with time step until the convergence condition is reached.

3. Results and discussion

To assess the correctness of the developed model, experimental validation was carried out by comparing the predictions with the available measured data [11]. In Fig. 2, the in-cylinder pressure variation of the SCD model and experimental results is shown, together with the data from the compressor model with constant boundary pressures. During the expansion and compression process, pressure changes of both the models are in good coincident with the measured values. However, the SCD model shows the merits of predicting the thermodynamic cycle by analysing in detail the pressure changes in the suction/discharge phase. The SCD model can predict pressure oscillation much better than the model with constant pressures, especially for the comparison in discharge process.

The verified model was employed to perform a further evaluation of the compressor-pipelines coupling. A well-

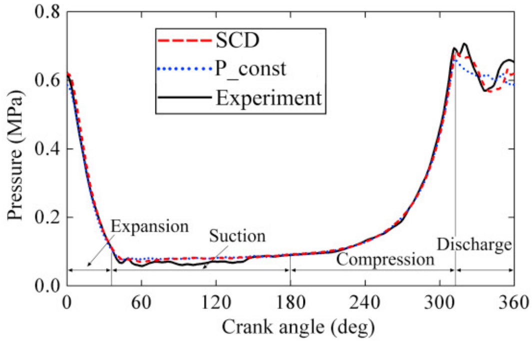

Fig. 2. In-cylinder pressure comparison between numerical and experimental data.

designed compressor with simple configurations is considered. The main specifications are listed in Table 1.

Fig. 3 shows the variation of (a) pressure in the suction chamber, (b) in-cylinder pressure with no piping configurations, suction pipe length of 1.45 m and of 2.08 m (second harmonic resonance in the suction pipelines). The length of discharge pipe is set as 0.1 m with the aim of restricting the effect of discharge system. It is clear that there is a certain influence of the pressure pulsation inside the pipelines on the compressor thermodynamic cycle and during the suction process, especially in the state of resonance. As can be seen in the suction phase, the in-cylinder pressure oscillation corresponds directly to the pressure pulsation in the suction system. The similar conclusion could be concluded from Fig. 4 when there is no discharge pipe, discharge pipe length of 1.27 m and of 1.9 m (second harmonic resonance in the discharge pipelines).

4. Conclusions

In this paper, a fully non-linear numerical model was developed for the analysis the thermodynamic process and pressure pulsation in the reciprocating compression system. The main advantage of the model is coupling the time-domain simulation of the compressor thermodynamic cycle with the time-domain computation of the transient flow in the pipeline system. The former allows the modelling of suction chamber, cylinder volume and discharge chamber with mass and energy conservation equations, while the latter is performed by using the Two-step Lax-Wendroff method. The validity of the model had been confirmed by giving satisfactory predictions against available experimental results. Afterwards, sensitivity analysis was carried out to highlight the mutual effect of pressure pulsation in the piping system and compressor thermodynamic performance.

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Table 1. Main specifications of the examined air compressor

Cylinder diameter	158 mm	Gas composition	air
Piston stroke	90 mm	Diameter of suction pipe	69 mm
Connecting rod length	250 mm	Diameter of discharge pipe	50 mm
Rotational speed	760 rpm	Volume of suction cavity	0.006 m^3
Number of cylinders	2	Volume of discharge cavity	0.006 m^3
Maximum valve lift	2.5 mm	Suction temperature	308.15 K
Spring stiffness of suction valve	1239 N/m	Suction pressure	0.3 MPa
Spring stiffness of discharge valve	2178 N/m	Discharge pressure	1.071 MPa

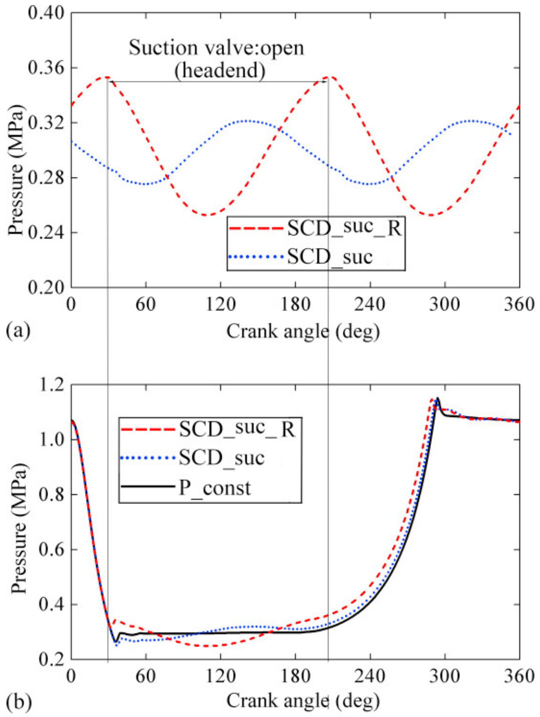

Fig. 3. Interaction between compressor and suction system with or without resonance: (a) suction pressure; (b) in-cylinder pressure.

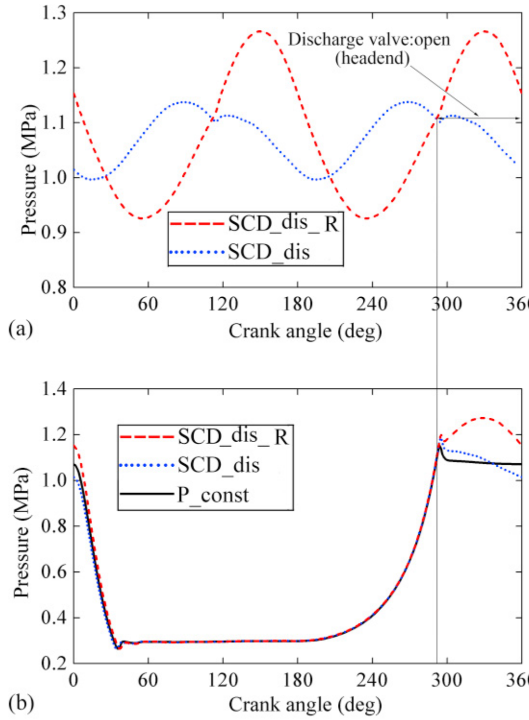

Fig. 4. Interaction between compressor and discharge system with or without resonance: (a) discharge pressure; (b) in-cylinder pressure.

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