Simulation study of multi-chamber rotary compressor

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Abstract – A new multi-chamber rotary compressor (MCR C) has been designed and the simulation result is presented in this paper. MCR C utilizes the space within the cylinder and it has the advantage of being more compact. Mathematical models which include geometrical, thermodynamics, mass flow and discharge valve have been formulated to evaluate the performance of MCR C. Parametric studies have also been carried out to determine the effects of design parameters such as suction and discharge ports size, valve length, valve thickness and valve width on compressor performance. In this paper, mathematical models will be presented and the predictions of the model will be shown and discussed.

Keywords – multi-chamber; simulation; mathematical model; rotary; MCR C

1. Introduction

In order to reduce the energy consumption by AC&R [1-6], a new multi-chamber rotary compressor (MCR C) has been designed. This new machine improves the compactness of conventional rolling piston compressor by introducing more working space at a given overall volumetric size of the compressor. In its basic form, MCR C comprises an outer cylinder, a hollow cylinder, an inner cylinder and a vane as shown in Figure 1 and 2.

The working principle of MCR C is illustrated in Figure 3. As the shaft rotates, the volumes of the working chambers vary and result in suction, compression and discharge process. During the operation, the working fluid is being drawn into both outer and inner suction chambers. Meanwhile the working fluid in both outer and inner compression chambers are being compressed and discharged when the pressure reaches the desired discharged pressure.
In this paper, mathematical models which include geometrical, thermodynamics, mass flow and discharge valve are formulated to evaluate the performance of MCRC. The effect of various design parameters on the compressor performance will also be presented and discussed.

2. Mathematical model

2.1. Geometrical Model
In a compressor, the thermodynamics properties of the working fluid are driven by the variation of working chamber volumes. The mathematical expression of outer compressor suction chamber volume, $V_{suc1}$, compression chamber volume, $V_{com1}$, inner compressor suction chamber volume, $V_{suc2}$, and compression chamber volume, $V_{com2}$, are derived as following.

\[ V_{suc1} = \ldots \]
\[ V_{com1} = \ldots \]
\[ V_{suc2} = \ldots \]
\[ V_{com2} = \ldots \]
potential energy and kinetic energy which can be obtained within the properties of the working fluid. During operation, the change in chamber volume gives

\[ V_{vulc} = \frac{1}{2} \left[ r_{oc}^2 \theta_1 - r_{oc}^2 \theta_3 - e (l_v + r_{oc}) \sin \theta_3 - l_v w_v + \frac{1}{4} w_v^2 \tan \theta_1 \right] l_{com1} \]

\[ V_{com1} = \pi (r_{oc}^2 - r_{oc}) l_{com1} - \frac{1}{2} \left[ r_{oc}^2 \theta_1 - r_{oc}^2 \theta_3 - e (l_v + r_{oc}) \sin \theta_3 + l_v w_v + \frac{1}{4} w_v^2 \tan \theta_1 \right] l_{com1} \]

\[ V_{vulc2} = \frac{1}{2} \left[ r_{ih}^2 \theta_2 - r_{ih}^2 \theta_1 - e \left( l_v + r_{ih} \frac{1}{2} w_v \right) \sin \theta_1 - w_v \left( l_v - \frac{1}{2} w_v^2 \right) - \frac{1}{4} w_v^2 \left( \frac{2}{2} \theta_2 - \gamma_2 \right) \right] l_{com2} \]

\[ V_{com2} = \pi (r_{ih}^2 - r_{ih}^2) l_{com2} - \frac{1}{2} \left[ r_{ih}^2 \theta_2 - r_{ih}^2 \theta_1 - e \left( l_v + r_{ih} \frac{1}{2} w_v \right) \sin \theta_1 + w_v \left( l_v - \frac{1}{2} w_v^2 \right) + \frac{1}{4} w_v^2 \left( \frac{2}{2} \theta_2 + \gamma_2 \right) \right] l_{com2} \]

where,

\[ l_v = -r_{ih} - t_h + (e^2 + r_{oc}^2 - 2r_{oc} \cos \theta_4) \]

\[ \theta_3 = \cos^{-1} \left( \frac{r_{oc}^2 - (r_{oh} + l_v) \cos \theta_4}{2e r_{oc} \sin \theta_4} \right) \]

\[ l_v = \frac{1}{2} w_v - r_{ic} - ecos \theta_1 + \left( r_{ih} - \frac{1}{2} w_v \right)^2 - e^2 \sin^2 \theta_1 \]

\[ \theta_2 = \cos^{-1} \left( \frac{e^2 + r_{ih} \frac{1}{2} w_v^2 - r_{ih} l_v - \frac{1}{2} w_v^2 \theta_2^2}{2e r_{ih} \frac{1}{2} w_v} \right) \]

\[ \gamma_1 = \begin{cases} \cos^{-1} \left( \frac{r_{oc}^2 + (r_{oh} + l_v)^2 - e^2}{2e (r_{oc} \sin \theta_4)} \right); & 0 \leq \theta_1 \leq \pi \\ -\cos^{-1} \left( \frac{r_{oc}^2 + (r_{oh} + l_v)^2 - e^2}{2e (r_{oc} \sin \theta_4)} \right); & \pi \leq \theta_1 \leq 2\pi \end{cases} \]

\[ \gamma_2 = \begin{cases} \cos^{-1} \left( \frac{(r_{ih} - \frac{1}{2} w_v)^2 + (r_{ih} + l_v - \frac{1}{2} w_v)^2 - e^2}{2(r_{ih} - \frac{1}{2} w_v)(r_{ih} + l_v - \frac{1}{2} w_v)} \right); & 0 \leq \theta_1 \leq \pi \\ -\cos^{-1} \left( \frac{(r_{ih} - \frac{1}{2} w_v)^2 + (r_{ih} + l_v - \frac{1}{2} w_v)^2 - e^2}{2(r_{ih} - \frac{1}{2} w_v)(r_{ih} + l_v - \frac{1}{2} w_v)} \right); & \pi \leq \theta_1 \leq 2\pi \end{cases} \]

2.2. Thermodynamic Model

During operation, the change of the volume of each working chamber results in the variation of properties of the working fluid. From the First Law of Thermodynamics \[7\], an energy rate balance of a control volume with the assumption that the fluid properties are uniform and have no spatial variation within the working chamber, gives

\[ \frac{dE_{cv}}{dt} = \frac{dQ_{cv}}{dt} - \frac{dW_{cv}}{dt} + \sum \frac{dm_i}{dt} \left( h_i + \frac{v_{i}^2}{2} + gz_i \right) - \sum \frac{dm_o}{dt} \left( h_o + \frac{v_{o}^2}{2} + gz_o \right) \]

The rate of work done due to change in chamber volume is given as

\[ \frac{dW_{cv}}{dt} = p_{cv} \frac{dV_{cv}}{dt} \]

The total energy of the control volume \( E_{cv} \) consists of internal energy, kinetic energy and gravitational potential energy which can be presented by

\[ E_{cv} = m_{cv} \left( u_{cv} + \frac{v_{cv}^2}{2} + gz_{cv} \right) \]

Substituting equation (12) and equation (13) into equation (11), assuming adiabatic and neglect all the potential energy and kinetic energy terms the following equation is obtained

\[ m_{cv} \frac{du_{cv}}{dt} + u_{cv} \frac{dm_{cv}}{dt} = -p_{cv} \frac{dV_{cv}}{dt} + \sum \frac{dm_i}{dt} h_i - \sum \frac{dm_o}{dt} h_o \]

solving this equation for internal energy together with the derivative of density in equation (15), respective thermodynamic properties can be obtained
The rate of change of mass of the control volume can be expressed as [8]

\[
\frac{dm_{cv}}{dt} = \sum \frac{dm_i}{dt} - \sum \frac{dm_o}{dt}
\]  

(15)

2.3. Mass Flow Model

The mass flow rate through the suction port and discharge port of both outer and inner compressor is formulated as following

\[
\frac{dm}{dt} = C_d \rho_{2s} A_{flow} \sqrt{2(h_1 - h_{2s})}
\]  

(16)

where \(C_d\) is given a value of 0.6 [9] and \(A_{flow}\) is defined as the port area.

2.4. Discharge Valve Response Model

The valve used in the discharge is an automatic reed valve. In MCRC, there are two discharge valves for both outer and inner compression chambers.

Assuming that the reed behaves like a beam, the free body diagram of an infinitesimal small element of reed is shown in Figure 4.

![Free body diagram of an infinitesimal small element of valve](image)

**Figure 4.** Free body diagram of an infinitesimal small element of valve

From elementary flexural theory [10], the following equations can be obtained

\[
E_v I_v \frac{\partial^4 y}{\partial x^4} + \rho A_c \frac{\partial^2 y}{\partial t^2} + c \frac{\partial y}{\partial t} = f(x, t)
\]  

(17)

Equation (17) is a linear, non-homogeneous equation with second order in time and fourth order in space. It is influenced by pressure excitation and the body force as shown at the right-hand side of the equation. Since it is a distributed system, the valve motion is analysed with free vibration and forced vibration approaches.

2.4.1. Free Vibration Analysis

For free vibration, equation (17) is simplified as following

\[
E_v I_v \frac{\partial^4 y}{\partial x^4} + \rho A_c \frac{\partial^2 y}{\partial t^2} + c \frac{\partial y}{\partial t} = 0
\]  

(18)

At certain time, the valve motion displays a unique profile and this profile does not change with time but the amplitude does. Hence, a separable solution is introduced, which is expressed as

\[
y(x, t) = f(x) \cdot g(t)
\]  

(19)

Substituting equation (19) into equation (18), applying method of separation of variables and dividing the whole equation by \(\rho A_c f g\), the following equation is obtained
\[ \frac{\rho A_c}{1} \frac{d^2 f}{dx^2} = - \frac{1}{\rho A_c} \frac{d g}{dt} + \frac{c}{\rho A_c} \frac{d g}{dt} = p^2 \]

(20)

where \( p^2 \) is the constant of separation and equation (20) is actually two ordinary differential equations which can be solved with the boundary conditions. By introducing \( a^2 = \frac{\rho A_c}{\rho A_c} \) and \( \beta^2 = \frac{p^2}{a^2} \) at \( k^{th} \) mode of vibration, the solution is as follows

\[ y_k(x, t) = A_k e^{i(\omega_k t)} \left[ \sin(\beta_k x) - \sin h(\beta_k x) \right] - \frac{\sin(\beta_k L) + \sin h(\beta_k L) + \cosh(\beta_k L)}{\cos(\beta_k L) + \cosh(\beta_k L)} (\cos(\beta_k x) - \cosh(\beta_k x)) \]

\[ \left[ \frac{\rho A_c g_k(0)}{q_{d,k}} + g_k(0) \cos(q_{d,k}) \right] \sin(q_{d,k} x) + g_k(0) \cos(q_{d,k} x) \]

(21)

The damped frequency \( q_{d,k} \) is defined as \( q_{d,k} = \frac{p_k}{\sqrt{1 - \xi^2}} \), where \( p_k \) is the natural frequency at \( k^{th} \) mode, which is represented by

\[ p_k = \beta_k^2 \left( \frac{\rho A_c}{\rho A_c} \right) \]

(22)

where \( \beta_k \) can be solve from the characteristic equation as below

\[ \cos(\beta_k L) \cosh(\beta_k L) = -1 \]

(23)

2.4.2 Forced Vibration Analysis

For forced vibration analysis, under the assumption of variable separable form and taking the principle of superposition into account, the solution can be expressed as

\[ y(x, t) = \sum_{k=1}^{N} f_k(x) g_k(t) \]

(24)

The orthogonal natural modes \( f_k(x) \) are normalized to satisfy orthonormality condition

\[ \int_0^L f_j(x) f_k(x) dx = \delta_{j,k} \quad \text{and} \quad \int_0^L f_j(x) \frac{d^2 f_k(x)}{dx^2} dx = \left( \frac{\rho A_c}{\rho A_c} \right)^2 \delta_{j,k} \]

where \( \delta_{j,k} \) is the kronecker delta. Substituting equation (24) into equation (17), divide the whole equation by \( \rho A_c \) and multiply the equation by \( f_k(x) \), integrate and apply the orthonormality condition, the following equation is obtained

\[ \frac{d^2 g_k}{dt^2} + \frac{c}{\rho A_c} \frac{d g_k}{dt} + \frac{p_k^2}{2} g_k = \int_0^L f_k(x) \frac{(f(x,t))}{\rho A_c} dx \]

(25)

The right-hand side of the equation is known as modal forces. In the case of MCR, the force term \( f(x, t) \) refers to the force per unit length, which is valid for the part where the valve is exposed to working fluid pressure only. As the natural modes \( f_k(x) \) is normalized, a relationship is found for the amplitude \( A_k \)

\[ A_k^2 \int_0^L \left[ \sin(\beta_k x) - \sin h(\beta_k x) \right] - \frac{\sin(\beta_k L) + \sin h(\beta_k L) + \cosh(\beta_k L)}{\cos(\beta_k L) + \cosh(\beta_k L)} (\cos(\beta_k x) - \cosh(\beta_k x))^2 dx = 1 \]

(26)

Considering that the discharge port diameter is small as compared to the valve length, and let \( f(x, t) = \frac{|P(t) - P_{disc}| A_f(Y)}{A_x} \), equation (25) becomes

\[ \frac{d^2 g_k}{dt^2} + \frac{c}{\rho A_c} \frac{d g_k}{dt} + \frac{p_k^2}{2} g_k = f_k \left( x + \frac{\Delta x}{2} \right) \frac{d P(Y)}{\rho A_c} \]

(27)

A general second order ODE solution cannot be applied in equation (27) as the chamber pressure is coupled with the valve response. By rewriting equation (27) into two first order ODEs as shown in equation (28) and equation (29), combining with thermodynamics model and solve both equations simultaneously, the modal coordinate as well as the displacement of the valve can be obtained.

\[ \frac{d \phi_1}{dt} = \phi_1 \]

(28)

\[ \frac{d \phi_1}{dt} = N_k - 2 \zeta \rho \phi_1 - p_k^2 \phi_2 \]

(29)
where $\zeta$ is given a value of 0.004 [11], $\phi_1 = \frac{d g_k}{dt}$, $\phi_2 = g_k$ and $N_k = f_k \left( x + \frac{\Delta y}{2} \right) \frac{\Delta P(v) A_r(y)}{\rho A_c}$.

3. Operating specifications
The geometrical, kinematics, thermodynamics, the mass flow and the discharge valve response models are solved numerically using the 4th order of Runge-Kutta method [12]. These models are integrated concurrently with FORTRAN and the instant thermodynamics properties of the working fluid are obtained from REFPROP database provided by NIST [13]. The following assumptions are made in this numerical simulation

- Adiabatic condition throughout the process.
- Frictional losses are not considered.
- Perfect sealing and no internal leakage.
- Constant shaft angular velocity.

The operating specifications is summarised in Table 1.

| Table 1. Operating specifications |
|----------------------------------|
| Working fluid                | R-32 (Difluoromethane) [14] |
| Operating speed ($\omega_1$) | 3000 rpm                     |
| Saturated evaporating temperature | 7.2 ºC [15]                |
| Saturated condensing temperature  | 54.4 ºC [15]               |

4. Results and Discussion
The normalized variation of volumes, pressure-volume (PV) diagram and variation of mass in MCRC are shown in Figure 5, Figure 6 and Figure 7 respectively. The pressures in the working chambers remain reasonably constant during the suction process and the mass increases gradually. Compression process begins at the end of suction process, where the working fluid pressure increases and the mass remains constant since there is no more working fluid flowing into the compression chamber under the assumption of perfect sealing. After the pressure of the working fluid reaches the nominal discharge pressure, it does not discharge immediately. Additional pressure builds up is required to overcome the valve stiffness and discharge only when the valve is uncovered. It is observed that both outer and inner compressors behave similarly therefore only the results of outer compressor will be shown and discussed.

4.1. Effect of Suction Port Size
The suction ports of MCRC are without a suction valve and the working fluid in suction chamber is always in contact with the upstream working fluid through the suction port. Figure 8 and Figure 9 show the effects of suction port size on the compressor performance. When the suction port diameter is 6.0 mm, the suction pressure is far below the nominal suction pressure and the mass intake is also lower as compared with larger suction port size. This implies a higher suction loss when the suction port is smaller. Hence, the size of the suction port must be appropriately selected.
Figure 5. Normalized variation of volumes in MCRC

Figure 6. Pressure-volume diagram for MCRC

Figure 7. Variation of mass of working fluid in working chambers

Figure 8. Effect of suction port size on outer compressor performance (pressure-volume)

Figure 9. Effect of suction port size on outer compressor performance (mass)
4.2. Effect of Discharge Port Size
A discharge port is always required to be sufficiently large to allow for ease discharge of the working fluid. Also, a larger port size avoids unnecessary over-pressure which may cause discharge valve failure and higher discharge loss. Figure 10 shows the variation of pressure under different discharge port size. The pressure “hump” occurs before the valve opening gets lower when the port size increases, and hence results in a lower discharge loss.

4.3. Effect of Valve Thickness
It is found that as the valve thickness increases, valve stiffness increases. Figure 11 shows the effect of valve thickness on compressor performance. Thicker valve reed also caused a higher discharge loss.

5. Conclusion
The initial simulation study of MCRC has been carried out which includes geometrical model, kinematics model, thermodynamics model, mass flow model and valve response model. Parametric study shows that larger suction and discharge port sizes are preferred in order to reduce the suction and discharge losses. Lastly, a thicker valve reed results in stiffer valve, which has incurred larger discharge loss and vice-versa.

In this paper, the embodiment design of MCRC has made use of the volume which is occupied by the roller in a rolling piston compressor, which is expected to be smaller in size as compared with the conventional rotary compressors. As MCRC is still under early stage development, the compressor performance as well as the corresponding efficiencies are not considered. In fact, more internal leakage flows and frictional losses will certainly play a role by having multi working chambers. For example, the frictional loss between the drive shaft and the hollow cylinder due to the surface in contact. More mathematical models such as frictional losses model, internal leakage model, heat transfer model and oil lubrication network model will be implemented in the future in order to make a more comprehensive theoretical model. With such a complete theoretical model, the comparison between MCRC and existing compressors will be more reasonable. The accuracy of the model will then be checked with the experimental result in the future.

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### Nomenclature

| Symbol | Description |
|--------|-------------|
| \( A \) | Area \([m^2]\) |
| \( A_c \) | Cross sectional area of valve \([m^2]\) |
| \( A_f \) | Effective force area on discharge valve \([m^2]\) |
| \( A_{flow} \) | Fluid flow area \([m^2]\) |
| \( A_k \) | Amplitude of \( k \)th mode of vibration \([-]\) |
| \( c \) | Damping coefficient \([Ns/m^2]\) |
| \( C_d \) | Coefficient of discharge \([-]\) |
| \( e \) | Distance between outer/inner cylinder centers and hollow cylinder centre \([m]\) |
| \( E \) | Summation of kinetic energy, potential energy and internal energy \([J]\) |
| \( E_v \) | Modulus of elastic of valve material \([Pa]\) |
| \( f_k(x) \) | Valve response profile at \( k \)th mode \([m]\) |
| \( f(x,t) \) | Valve response profile at \( k \)th mode \([m]\) |
| \( g_k(t) \) | Variation of valve response profile amplitude at \( k \)th mode \([-]\) |
| \( h \) | Specific enthalpy of working fluid \([J/kg]\) |
| \( h_{2s} \) | Specific enthalpy of working fluid at downstream under isentropic condition \([J/kg]\) |
| \( I_v \) | Moment of inertia of valve \([m^4]\) |
| \( K \) | Flow coefficient \([-]\) |
| \( l \) | Length \([m]\) |
| \( L_v \) | Length of valve \([m]\) |
| \( m \) | Mass \([kg]\) |
| \( M \) | Bending moment \([Nm]\) |
| \( N_k \) | Modal force \([N]\) |
| \( p_{cv} \) | Pressure in control volume \([Pa]\) |
| \( p_k \) | Natural frequency of the valve at \( k \)th mode \([rad/s]\) |
| \( P \) | Pressure \([Pa]\) |
| \( q_{d,k} \) | Damped response frequency of valve at \( k \)th mode |
| \( Q \) | Heat transfer \([J]\) |
| \( r \) | Radius \([m]\) |
| \( s \) | Entropy \([J/kg-K]\) |
| \( t \) | Time \([s]\) |
| \( T \) | Temperature \([K]\) |
| \( u \) | Internal energy \([J/kg]\) |
| \( v \) | Velocity \([m/s]\) |
| \( V \) | Volume \([m^3]\) / Shear force \([N]\) |
| \( w \) | Width \([m]\) |
| \( W \) | Work done \([J]\) |
| \( x \) | Location along the valve longitudinal axis \([m]\) |
| \( y \) | Valve displacement \([m]\) |
| \( z \) | Elevation \([m]\) |
| \( \alpha \) | Angular acceleration \([rad/s^2]\) |
| \( \delta_{ks} \) | Kronecker delta \([-]\) |
| \( \eta_s \) | Isentropic efficiency \([-]\) |
| \( \zeta \) | Damping coefficient \([-]\) |
| \( \rho \) | Density \([kg/m^3]\) |
| \( \omega \) | Angular velocity \([rad/s]\) |
| \( \gamma \) | Subtended angle \([rad]\) |
| \( \Delta P \) | Pressure difference \([Pa]\) |