Optimal Structural Design of Swing Double-Vane Compressor

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ABSTRACT

In this paper, a novel rotary compressor, namely Swing Double-vane Compressor, was introduced. Based on the working principle of the compressor, thermodynamic model and kinetic model of upper and down vanes, rotor and eccentric shaft were established. Combined with each model, with the working efficiency as the objective function, the primary structural sizes were optimally designed for the compressor by optimization algorithm, after which the working efficiency can be to 0.8757 and its theoretical displacement has improved by 66.94% compared with the single-vane compressor. According to this optimal design, it will provide theoretical foundation to improve the performance of the Swing Double-vane Compressor.

1. INTRODUCTION

In order to solve the problems of working face frictional wear and leakage for the high speed of small rotary compressor, professor Qu, Z. C. (2011) put forward a novel swing vane compressor. Due to lower speed of translation and rotation, the friction loss was reduced and the mechanical efficiency was effectively improved for the hinged structure of the compressor with the better sealing. Li, Q. C. (2009) has carried on the simple analysis of the thermodynamics, leakage and kinetic for the swing vane compressor, compared with the rolling rotor compressor and synchronous rotary compressor, he found that the swing vane compressor has lower speed and smaller acceleration which change slowly, and the friction power is relatively small with a more stable operational condition. Based on the compressor geometry model, thermodynamics model and kinetic model, Li, X. L. (2009) has conducted viable optimizing analysis in structural parameters and verified the feasibility of improved compressor in actual operation, after a relative test of the performance parameters. Teh, Y. (2002) presented a designed improvement to reduce the wear and friction at the vane side effectively by achieving a 30% reduction over its predecessor. Zong, W. B. (2015) established the dynamic model of translational rotary compressor to discuss the natural frequency and vibration type by comparing the critical and actual speed.

In this paper, on the basis of previous studies of swing vane compressor, increase a vane is added into the lower end of the rotor to form double vane structure, its movement mechanism basically consistent with compressor. Based on the analysis of the machine geometry model, thermodynamic model and dynamics model, the mathematical optimization design is established with the EER of compressor defined as the objective function. The state parameters of thermodynamics of compressed gas are obtained using the
optimization algorithm, and the optimal structure parameters can be carried out by analyzing the calculation results.

2. THE ESTABLISHMENT OF THE MATHEMATICAL MODEL

2.1 The Geometric Model

With working principle of new double swing vane compressor shown in Fig. 1, the tip of the main board is embedded into the cylinder by the way of hinged point and the end inserted in the groove of the rotor. Due to eccentric motion of rotor, the vane is swinging around its tip, while the end is dong linear motion in the groove. The down vane which can move randomly in the slot is embedded in the symmetry side of the rotor, and the tip of the down vane is tightly pressed on the inner wall of the cylinder for the centrifugal force and the high pressure of the lubricating oil. Because of the two vanes between rotor and cylinder and simultaneously operation bilaterally of two working chambers, the compressor has completed two cycles when the eccentric shaft rotates a full revolution.

![Figure 1: The working process of double swing vane compressor](image)

Taking the left-side working chamber as an example, The chamber volume of the compressor calculated by the method of Ma, J. J. (2015), can be expressed as shown below:

$$V_i(\theta) = H \cdot \left( \frac{R^2}{2} \theta - \frac{r^2}{2} (\theta + \arcsin \frac{e \sin \theta}{\sqrt{R^2 + e^2 - 2eR \cos \theta}}) - \frac{eR \sin \theta}{2} - (\sqrt{R^2 + e^2 - 2eR \cos \theta} - r) \cdot \frac{b_1}{2} \right)$$

(1)

When $\pi \leq \theta \leq 2\pi$, the relative geometric parameters are given by.
\[ L_0 = \sqrt{e^2 + R^2 - 2 \cdot e \cdot R \cdot \cos \theta} \]  
(2)

\[ \alpha = \arcsin \left[ \frac{e}{L_0} \cdot \sin (2\pi - \theta) \right] \]  
(3)

\[ L_1 = 2R \cdot \cos \alpha - L_0 - r \]  
(4)

Taking the center of the rotor as the origin of the polar coordinates system, its polar coordinates is expressed as \((e, \theta - \pi)\) and the value of radius is \(R\). Then Equation (5) is given by cosine theorem,

\[ R^2 = \rho^2 + e^2 - 2 \cdot \rho \cdot e \cdot \cos[\gamma - (\theta - \pi)] \]  
(5)

Equation (6) is simplified as follow,

\[ \rho = e \cdot \cos[\gamma - (\theta - \pi)] + \sqrt{(R^2 - e^2) + e^2 \cdot \sin^2[\gamma - (\theta - \pi)]} \]  
(6)

According to the equations above, the volume of working chamber is described in the formula,

\[ V_2(\theta) = H \cdot \frac{1}{2} \int_{\alpha}^{\frac{\pi + \alpha}{2}} \left( \rho^2 - r^2 \right) d\gamma - (L_0 - r) \cdot \frac{b_1}{2} - L_1 \cdot \frac{b_2}{2} \]  
(7)

When \(2\pi \leq \theta \leq 3\pi\), as shown in Fig. 2c, the calculation formula of stroke volume is basically similar to that of \(\pi \leq \theta \leq 2\pi\) (Equation (3)\textendash (6)),

\[ \alpha = \arcsin \left[ \frac{e}{L_0} \cdot \sin (\theta - 2\pi) \right] \]  
(8)

\[ \rho = e \cdot \cos[\gamma - (\theta - \pi)] - \sqrt{(R^2 - e^2) + e^2 \cdot \sin^2[\gamma - (\theta - \pi)]} \]  
(9)

The volume of working chamber is represented as follows,

\[ V_1(\theta) = H \cdot \frac{1}{2} \int_{\theta-2\pi}^{\frac{\pi + \alpha}{2}} \left( \rho^2 - r^2 \right) d\gamma - L_1 \cdot \frac{b_1}{2} \]  
(10)

Figure 2: The diagram of working volume of compressor

Due to the symmetry of the structure, the right working volume of compressor can be defined as:
2.2 The Thermodynamic Model

The change of chamber volume will cause the pressure, temperature and mass of fluid to change during the working process, if regarding the chamber volume of compressor as control volume and the medium in the homogeneous state, the equation of energy conservation is established according to the law of thermodynamics as:

\[
\frac{d(mu)}{dt} = \frac{dQ_i}{dt} + \frac{dE_i}{dt} - \frac{dE_{0i}}{dt} + \frac{dW}{dt}
\]  

(12)

Assume that the speed of the compressor is constant, and the angle of the shaft is \( \theta = \omega t \), namely \( dt = d\theta / \omega \), then Equation(13) is given according to the law of thermodynamics \( h = u + pv \), \( v = V_c / m \) as follows:

\[
\frac{d(mu)}{d\theta} = m \frac{dh}{d\theta} + h \frac{dm}{d\theta} - p \frac{dV_c}{d\theta} - V_c \frac{dp}{d\theta}
\]  

(13)

The total derivative of the pressure is obtained:

\[
\frac{dp}{d\theta} = \left( \frac{\partial p}{\partial v} \right) \frac{dv}{d\theta} + \left( \frac{\partial p}{\partial T} \right) \frac{dT}{d\theta}
\]  

(14)

The Equation (15) is the deduction of derivative of specific volume:

\[
\frac{dv}{d\theta} = \frac{d(V_c / m)}{d\theta} = \frac{1}{m} \frac{dV_c}{d\theta} - \frac{V_c}{m^2} \frac{dm}{d\theta}
\]  

(15)

And the expressions of pressure and temperature in the control volume can be written as:

\[
\frac{dp}{d\theta} = \frac{1}{v} \left( \left( \frac{\partial h}{\partial v} \right)_T - \left( \frac{\partial h}{\partial T} \right)_v \left( \frac{\partial p}{\partial v} \right)_T / \left( \frac{\partial p}{\partial T} \right)_v \right) \frac{dv}{d\theta} - \frac{1}{V_c} \left\{ \sum \left[ \frac{dm}{d\theta} (h_f - h_i) \right] + \frac{dQ_i}{d\theta} \right\}
\]  

(16)

\[
\frac{dT}{d\theta} = \frac{1}{v} \left( \left( \frac{\partial h}{\partial v} \right)_T - \left( \frac{\partial h}{\partial T} \right)_v \left( \frac{\partial p}{\partial v} \right)_T / \left( \frac{\partial p}{\partial T} \right)_v \right) \frac{dv}{d\theta} - \frac{1}{V_c} \left\{ \sum \left[ \frac{dm}{d\theta} (h_f - h_i) \right] + \frac{dQ_i}{d\theta} \right\}
\]  

(17)

2.3 The Kinetic Model

The input energy of the swing double-vane compressor mainly consists of two parts, namely the
indicated work and consumption of friction. And on the basis of the analysis of thermodynamics, the indicated work of a whole working cycle is formulated as:

\[ W_i = \frac{n}{60} \int_0^{2\pi} V(\theta) d\theta \]  

(18)

And the indicated power of compressor can be obtained by the equation:

\[ P_i = \frac{n}{60} W_i \]  

(19)

One of the principal factors impacting on the performance of the compressor is the friction losses of motion parts, in the calculation mainly taking the five kinds of losses into consideration, and they are the friction of sliding vanes; the friction between vane-end and cylinder; the friction of the end faces of rotor and cylinder; the friction of the bearing of rotor and shaft; the friction between the rotor and eccentric shaft.

Generally, the calculation method of friction consumption can be summed up as:

\[ L_f = \begin{cases} \mu F_s \\ \omega M \end{cases} \]  

(20)

According to the Ma, J. J. (2015), the sum of five friction losses is respectively the total loss by the definition:

\[ P_f = \frac{n}{60} \sum L_{f,j} \]  

(21)

And the total power consumption can be exactly given as:

\[ P_{tot} = P_i + P_f \]  

(22)

3. RESEARCH OF OPTIMIZATION

In the stage of design and development of the compressor, the method of genetic optimization is adopted to improve the efficiency and reliability of the compressor, combining with the analysis of thermodynamics and kinetics. That is the EER of the compressor is defined as objective function to find out the optimal structural parameters of compressor. Manole, D. (2002).

The structural parameters of the compressor contain the outer radius of rotor, the inner radius of cylinder, the length of vane, the revolving speed, the axial length of cylinder and so on. If considering too many design parameters, the burden of computation can be increasing so that in this paper, three of the structural parameters, namely the radius of rotor \( r \), the axial length of cylinder \( H_c \) and the length of vane \( L_s \), are chosen to be design variables. And the other structural parameters are assigned appropriate constants, for example, the assignment of the inner radius of cylinder \( R_s \) is 50mm, the inner diameter of rotor \( d \) is 40mm, the width of vane is 6mm and the material density of rotor and vane is about 7800kg·m\(^{-3}\).

Thus the design variable of the compressor is expressed as:

\[ X = [r, H_c, L_s]^T \]  

(23)

In this paper, in order to obtain maximum EER of the compressor, the EER of any design variables is defined as the objective function as the following equation:

\[ F_{obj} = EER = \frac{Q_0}{P_{tot}} \]  

(24)
In the simulation, the environmental parameters of the compressor are set: suction pressure 0.1 MPa, suction temperature 293K, discharge pressure 0.6 MPa, revolving speed 2000 r·min⁻¹ and the physical parameters of the working medium is regarded as ideal gas.

In the iterative computations, when the calculation error is less than 10⁻⁴, the optimization design has been considered to be converged. According to the simulation, the optimal structural parameters can be obtained when the number of genetic optimization iteration is 44, they are the rotor radius 42.4 mm, axial length of rotor 46.2 mm and the length of vane 20.3 mm, with the process of iteration shown in Fig. 3. Fig. 4 shows that the average fitness of the generation has a obvious change in the iteration and finally, the corresponding EER reaches to 0.8757.

![Figure 3: The process of parameters optimization](image)

![Figure 4: The process of efficiency optimization](image)

4. SIMULATION ANALYSIS
According to the established thermodynamic model, numerical simulation of the double-vane compressor has completed and subsequently has drawn the analysis and research in detail Ooi, K. T., & Wong, T. N. (1997). In the model, the structural parameters are chosen to be design variable. Table 1 shows the initial structural parameter values. The feasible initial parameter values can be random set between the upper and lower limits of the feasible region.

**Table 1: The structural parameters of the compressor**

<table>
<thead>
<tr>
<th>Main parameters</th>
<th>Values</th>
</tr>
</thead>
<tbody>
<tr>
<td>Cylinder diameter /mm</td>
<td>100</td>
</tr>
<tr>
<td>Outside diameter of rotor /mm</td>
<td>90</td>
</tr>
<tr>
<td>inside diameter of rotor /mm</td>
<td>52</td>
</tr>
<tr>
<td>Eccentric distance /mm</td>
<td>5</td>
</tr>
<tr>
<td>Thickness of vane /mm</td>
<td>6</td>
</tr>
<tr>
<td>Length of vane /mm</td>
<td>20</td>
</tr>
<tr>
<td>Axial length of cylinder /mm</td>
<td>60</td>
</tr>
<tr>
<td>Diameter of eccentric shaft /mm</td>
<td>35</td>
</tr>
</tbody>
</table>

The working chamber of the compressor is surrounded by inner surface of cylinder, external surface of rotor and sides of vane, thus Fig.5 shows how the working chambers of the double-vane compressor and the single-vane compressor change with the angle of eccentric shaft.

![Figure 5: Change of working chamber volume](image)

As indicated in Fig.5, by the time the eccentric shaft completes two revolutions, each chamber in the compressor will have gone through one stroke of suction-exhaust respectively with theoretical displacement being $7.47\times10^{-5}$ m$^3$, while the value of single-vane compressor is $8.95\times10^{-5}$ m$^3$. Thus the double-vane compressor possesses an obvious advantage: its theoretical displacement has improved by 66.94% compared with the single-vane compressor.

Taking the left working chamber as research object, after shaft revolution about 540°, an operation circulation has completed with the temperature and pressure of the chamber changing as shown in Fig.6. Pressure varies with the rotation Angle and pressure fluctuation in the process of suction and...
exhaust is the result of the movement characteristic of valve plate. And the diagram demonstrates that the changing characteristic of temperature is tolerably consistent with the pressure, which corresponds to the law of thermodynamics, it is verified that the thermodynamic model established can be feasible.

![Figure 6: Change of chamber temperature and pressure](image)

6. CONCLUSION

According to the basic structure of the swing double-vane compressor and the relevant theory of thermodynamics and kinetics, the basic optimization parameters of the compressor can be determined. After choosing the appropriate objective function and the method of genetic optimization under the restricted reasonable conditions, mathematical model of optimum design is established. Friction loss of optimized compressor will be reduced dramatically, on the basis of simulation results, we can draw that the optimal structural parameters are the rotor radius -42.4mm, axial length of rotor -46.2mm and the length of vane -20.3mm, and energy efficiency ratio reaches to 0.8757, its theoretical displacement has improved by 66.94% compared with the single-vane compressor, which will provide theoretical foundation to improve the performance of the novel compressor.

NOMENCLATURE

- $A$: valve flow area ($m^2$)
- $b$: thickness of vane ($m$)
- $C$: discharge coefficient
- $e$: eccentric distance ($m$)
- $F$: force ($N$)
- $H$: height of rotor ($m$)
- $L$: length ($m$)
- $L_f$: frictional loss ($W$)
- $m$: mass of fluid ($Kg$)
\[ n \] rotational speed of compressor (rev/min)
\[ p \] pressure (pa)
\[ Q \] heat transfer rate (W)
\[ Q_0 \] cooling capacity (W)
\[ R \] radius (m)
\[ T \] temperature (K)
\[ U \] velocity (m·s\(^{-1}\))
\[ V \] volume (m\(^3\))
\[ v \] specific volume (m\(^3\)/kg)
\[ W_i \] indicate work (J)
\[ \delta \] clearance (m)
\[ \theta \] drive angle (rad)
\[ \varepsilon \] bearing eccentric ratio
\[ \eta \] kinetic friction coefficient
\[ \mu \] dynamic viscosity of lubricant (pa·s)
\[ \omega \] angular velocity (rad·s\(^{-1}\))

**REFERENCE**


**ACKNOWLEDGMENTS**

Many thanks to my supervisor: Professor Qu for his advice and to my teammates: Cang, Rong; Chen, Xiang; Yang, Xu for their work.