Mechanical and deformation characteristics of rotary compressor discharge valve

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Abstract. The discharge valve plays an important role in the rotary compressor. For better understanding the mechanical characteristics of discharge valve in the process of movement, a mechanical model considering the coupling effect with limiter is established. Some major parameters, i.e., displacement, stress, stiffness and elastic force, are obtained to characterize the mechanical properties of the discharge valve. Two cases of valve component are calculated by the mechanical model, and the results indicated that the profile of limiter has an important influence on the movement of discharge valve. In order to verify the accuracy of the model, finite element method and high-speed photography are used. The comparison results show the calculation error of valve stiffness obtained by the model is within 3.9% compared with the finite element method results, and the valve motion is consistent with the high-speed photography results. Based on the above analysis, the profile of limiter in one valve component is optimized. The initial winding displacement of the valve is advanced from 2.115 mm to 1.453 mm after the optimization of limiter profile, which exhibits better limiting effect.

1. Introduction
As the heart of air conditioning, compressor plays an important role in refrigeration cycle, and the discharge valve is the core component to achieve the normal working of the compressor. A large number of research results show that the discharge valve component is important to the reliability, performance and noise level of compressor [1-5]. When the discharge valve slaps the limiter and the valve seat repeatedly, the bending fracture and the impact fracture often occur. In order to solve this problem, the bending stress and impact stress of the discharge valve are usually reduced by adjusting the lift and stiffness of the discharge valve, respectively. However, this method often leads to the sacrifice of compressor performance [6].

For better improving the reliability and performance of compressor, the motion characteristics of discharge valve under the coupling action with the limiter should be better understood. The limiter is main to limit the lift of discharge valve, avoiding fracture of discharge valve due to excessive bending stress. Kim et al. [7] provided a novel method for structural stability of limiter by calculating the maximum contact stress of limiter using Adina simulation. Zhang et al. [8] found the impact speed of valve mainly depends on the lift of limiter, and gave the reasonable range of lift and impact speed. Kwon et al. [9] found that the stability of compressor is better with the decrease of limiter length, and the performance of compressor is better with the increase of limiter length.
The above researches are mostly aim at the reliability and performance of compressor, but not consider the effect of limiter profile. The limiter profile mainly includes linear, double linear, single curvature, single curvature linear, double curvature and so on, and the limiter profile has a great influence on the valve movement. Zheng et al. \cite{10} analyzed the stress distribution of three limiter profiles, and indicated that valve has better motion characteristics with single curvature type. Wang et al. \cite{11} found that single curvature and single curvature linear limiter have no effect on the design of small compressor, but single curvature linear limiter has larger effective flow area and smaller characteristic lift. Ho et al. \cite{13} verified the maximum stress of valve under two kinds of limiter, the results shown the maximum stress of valve under limiter A is higher 26% than it under limiter B. Mu et al. \cite{12} proposed a valve model considering the coupling effect with limiter, and established the relationship between the effective working length of valve and the elastic force of valve with lift. Nevertheless, the above researches have not carried out a detailed study on the mechanical deformation characteristics of valve under the restriction of limiter, and it is hard to understand the actual movement and stress state of valve in the process of movement.

In this paper, a mechanical model of discharge valve considering the coupling effect with limiter is established, and some major parameters of the discharge valve are obtained. This model is used to analyze the valve movement under the two limiter profiles, and the accuracy of the model is verified by simulation and experiment. Furthermore, the profile of limiter B is optimized.

2. Theoretical model

In the process of valve opening and closing, the root of the valve is constrained by the limiter and rivet. The parallel part of the limiter can be considered as the fixed end constraint. The head of the valve is covered on the flange vent hole and bears the gas force, as shown in Figure 1a. Based on the above structure and mechanical characteristics, the valve can be simplified as a cantilever beam. One end is fixed, and the other end is under uniform load. In order to facilitate the calculation, the load is equivalent by using the superposition principle, as shown in Figure 1b. It should be noted that the model does not consider the variable cross-section of the valve head.

![Diagram of discharge valve component](image1.png)

(a) Diagram of discharge valve component

![Cantilever beam model](image2.png)

(b) Cantilever beam model

Figure 1. Diagram of discharge valve component and cantilever beam model.

Because the equivalent cantilever beam bears two different loads, it is necessary to carry out sectional analysis, and the moment $M$ of valve is

\[
M(x) = \begin{cases} 
q(l - a)x - \frac{q}{2}(l^2 - a^2) & 0 \leq x \leq a \\
-\frac{q}{2}(l - x)^2 & a \leq x \leq l
\end{cases}
\]

where $q$ is uniform load. According to the bending force and deformation formula of material mechanics, there is

\[
y'' = \frac{d^2w}{dx^2} = \frac{M(x)}{EI_0} \\
y' = \frac{dw}{dx} = \int \frac{M(x)}{EI_0} dx + C
\]
\[
y = w = \int \left( \frac{M(x)}{E I_0} dx \right) dx + C x + D
\]

(2)

where \( w \) is the lift of valve head center, \( E \) is Elastic modulus, \( I_0 \) is moment of inertia, \( C \) and \( D \) are undetermined coefficient.

According to geometric boundary conditions, there is

\[
y'(0) = 0, \quad y'(l) = 0
\]

\[
y'(0) = y_1', \quad y'(a) = y_2'
\]

(3)

Where \( y_1 \) and \( y_2 \) are displacements for \( 0 < x \leq a \) and \( a \leq x < l \), respectively. Combining equations (1), (2) and (3), we can achieve the deflection curve and moment of valve, that is

\[
y(x) = \begin{cases} 
\frac{96(l^2 - a^2)x^2 - 64(l - a)x^3}{17d^4 + 32ad(3l^2 + a^2)} & 0 \leq x \leq a \\
-\frac{384w[-16(l - x)^4 - (192ad + 64d^2)(x - a) + 16d(-6a^2l - 2a^3 + d)]}{17d^4 + 32ad(3l^2 + a^2)} & a \leq x \leq l
\end{cases}
\]

(4)

\[
M(x) = \begin{cases} 
\frac{384EI_0w}{17d^4 + 32ad(3l^2 + a^2)} \left[ (l - a)x - \frac{1}{2}(l^2 - a^2) \right] & 0 \leq x \leq a \\
\frac{192EI_0w}{17d^4 + 32ad(3l^2 + a^2)} (l - x)^2 & a \leq x \leq l
\end{cases}
\]

(5)

Then, the stress and stiffness can be achieved, namely

\[
\sigma(x) = \begin{cases} 
\frac{384EI_0w}{17d^4 + 32ad(3l^2 + a^2)} \left[ (l - a)x - \frac{1}{2}(l^2 - a^2) \right] \frac{t}{2I_0} & 0 \leq x \leq a \\
\frac{192EI_0w(l - x)^2}{17d^4 + 32ad(3l^2 + a^2)} \frac{t}{2I_0} & a \leq x \leq l
\end{cases}
\]

(6)

where \( t \) is thickness of valve.

\[
k = \frac{qd}{w} = \frac{384EI_0d}{17d^4 + 32ad(3l^2 + a^2)}
\]

(7)

The above theoretical process only considers the mechanical and deformation of the valve under the action of gas force on the head with the root restraint, and does not consider the coupling effect between the valve and the limiter. With the gradual opening of the valve, the radius of curvature gradually decreases. When the radius of curvature of the valve decreases to equal to the radius of curvature of the limiter, the two start to wind. Firstly, the initial contact position of valve and limiter (Initial winding point) can be determined based on the deflection curve of the valve and the profile line of the limiter. The limiter profile has two common types: continuous type and discontinuous type, as shown in Figure 2.

![Figure 2. Diagram of two limiter profiles.](image)

For continuous type, the center of limiter profile is just above the starting point, while the center of limiter profile is on the starting point left for discontinuous type. The continuous type limiter profile is


\[ y = r_1 - \sqrt{r_1^2 - x^2} \tag{8} \]

and the discontinuous type limiter profile is

\[ y = \sqrt{r_2^2 - c^2} - \sqrt{r_2^2 - (x + c)^2} \tag{9} \]

Because the analysis process of two limiters is same, taking the continuous type limiter as an example, the mechanical and deformation of valve is studied with the coupling effect. Combining the equations (4) and (8), the initial winding displacement \( w_0 \) at the center of valve top (Valve displacement) can be achieved. For the continuous type limiter, the initial winding point is always at the point \( x = 0 \).

Based on equation (7), it can be seen that the valve stiffness is mainly related to the material parameters and structural parameters of the valve. When the valve displacement is less than \( w_0 \), the valve and the limiter have not yet been wound. At this time, the valve stiffness is a constant value. When the valve displacement is greater than \( w_0 \), the valve and the limiter begin to wind, and the effective working length \( l \) gradually decreases, which is not the deformation length \( l \) any more. The valve stiffness analysis is divided into two stages, as shown in Figure 3. Stage I is before that the valve winding point does not reach the uniform load position, that is \( l' \geq l - a \). Stage II is when the valve winding point has reached the uniform load position, that is \( l' < l - a \). Therefore, the two stages have different stress forms.

The stiffness of valve in two stages can be calculated, namely

when \( l' \geq l - a \),

\[ k_1 = \frac{qd}{w} = \frac{384EI_0d}{17d^4 + 32d(a - x_c)\left[3(l - x_c)^2 + a^2\right]} \tag{10} \]

when \( l' < l - a \),

\[ k_2 = \frac{qd}{w} = \frac{24EI_0(l - x_c)}{(l - \frac{d}{2} - x_c)\left[(l - \frac{d}{2} - x_c)^2 - 4(l - x_c)(l - \frac{d}{2} - x_c) + 6(l - x_c)^2\right]} \tag{11} \]

The relationship between the valve stiffness and the length of deformation section can be established by equations (7), (10) and (11). In order to get the relationship between the stiffness and valve displacement, it is necessary to establish the relationship between the displacement and the length of the valve. During calculating the valve displacement, it can be divided into three parts, as shown in Figure 4. Part one \( y_1 \) is the height of limiter at the winding point, part two \( y_2 \) is the height of valve center due to the angle \( \alpha \) at the winding point, part three \( y_3 \) is the valve displacement because of the deformation of the rest length.
Based on the limiter profile, \( y_1 \) and \( y_2 \) can be easily obtained.

\[
\begin{align*}
    y_1 &= r_1 - \sqrt{r_1^2 - x_c^2} \\
    y_2 &= (l - x_c - d) \frac{x}{\sqrt{r_1^2 - x_c^2}} 
\end{align*}
\]  

(12) and (13)

Based on the mechanical model in Figure 3, \( y_3 \) can be obtained.

\[
y_3 = \begin{cases} 
    \frac{-17qd^4 + 96qd(a-x_c)(l-x_c)^2 + 32qd(l-x_c)^4}{384EI_o} & 0 \leq x_c \leq a \\
    \frac{q(l-d-x_c)^2}{24EI_o} \left[ (l-d-x_c)^2 - 4(l-x_c)(l-d-x_c) + 6(l-x_c)^2 \right] & a \leq x_c \leq l
\end{cases}
\]  

(14)

In equation (14), the load \( q \) is uncertain and need to be obtained. According to the define of bending moment,

\[
M(x_c) = \frac{EI_o}{r_1}
\]  

(15)

where \( r_1 \) is the curvature radius of limiter. Then the load \( q \) will be

\[
q(x_c) = \begin{cases} 
    \frac{EI_o}{r_1(l-a)x_c - \frac{(l^2-a^2)}{2}} & 0 \leq x_c \leq a \\
    \frac{-2EI_o}{r_1(l-x_c)} & a \leq x_c \leq l
\end{cases}
\]  

(16)

Combining equations (7), (10), (11), (12), (13), (14) and (16), we can achieve the relation of valve stiffness and valve displacement. So the elastic force is easily achieved by equation (17).

\[
F_{sp} = k \cdot w
\]  

(17)

The calculation of the maximum bending stress of the valve is divided into two processes, that is, the maximum bending stress of the valve occurs at the root of the valve before winding, as shown in equation (18), and the maximum bending stress of the valve occurs at the winding position after winding, and changes with the change of the winding position, as shown in equation (19).

\[
\sigma_{max} = \frac{96EI_oW(l+a)}{17d^4 + 32a(3l^2 + a^2)} I_o
\]  

(18)

\[
\sigma_{max} = \frac{M}{W_z} = \frac{EI_o}{r_1 W_z} = \frac{Et}{2r_1}
\]  

(19)

According to equation (18) and (19), we can get the change of the maximum bending stress of the valve with the opening process of the valve. It is worth noting that the maximum bending stress of the valve is a constant value when the valve is wined on the limiter, and it occurs at the real-time position of the winding point.
3. Results and Discussion

3.1 Calculated Results

Table 1 shows the parameters of the two valve components used in this calculation. The limiter in valve component A is a continuous limiter, and the limiter in valve component B is a discontinuous limiter. Based on the above theoretical principle, the results of two valve components can be obtained.

Table 1. The structural and material parameters of two valve components.

| Parameters                  | Valve component A | Valve component B |
|-----------------------------|-------------------|-------------------|
| Deformation length of valve/mm | 23.05             | 27                |
| Valve width/mm              | 4.2               | 6                 |
| Valve thickness/mm          | 0.254             | 0.381             |
| Valve head diameter/mm      | 12                | 13                |
| Elastic modulus/GPa         | 210               | 210               |
| Limiter lift/mm             | 2                 | 2.2               |
| Curvature radius of limiter/mm | 73.57           | 130               |
| Center offset of limiter/mm | 0                 | 3.55              |

As shown in Figure 5, the initial winding point of the valve component A under the constraint of continuous limiter occurs at \( x = 0 \) when the valve displacement reaches \( 1.321 \) mm. For valve component B, when the valve displacement reaches \( 2.115 \) mm, the initial winding point of the valve is located at the position of \( x = 14.8 \) mm. According to the two groups of results, it can be seen that the valve is gradually winding and opening under the continuous limiter profile, while the valve is directly hitting the limiter at a certain distance from the root under the discontinuous limiter profile.

![Figure 5](image1.png)

**Figure 5.** The initial winding position of two valve components.

![Figure 6](image2.png)

**Figure 6.** The change of valve stiffness with the deformation length of valve for two components.

Figure 6 shows the variation of valve stiffness of the two valve components with the effective working length. When the effective working length is the initial length, the initial stiffness of valve component A is \( 0.727 \) N/mm, and that of valve component B is \( 2.018 \) N/mm. The results show that the effective working length of the valve will decrease when the valve is winded or impacted on the limiter, and the valve stiffness will increase with the decrease of effective working length.

![Figure 7](image3.png)

**Figure 7.** The relation of valve displacement and effective working length.
In order to establish the relationship between the valve stiffness and the valve displacement, the relationship between the valve displacement and the effective working length is obtained, as shown in Figure 7. The results show that the total displacement \( H \) increases with the decrease of effective working length for both valve components, where \( y_1 \) increases gradually, \( y_2 \) increases first and then decreases, and \( y_3 \) decreases gradually.

The relationship between valve stiffness and displacement is shown in Figure 8a. The valve stiffness is constant when the displacement is less than the initial winding displacement, and increases sharply after the displacement exceeds the initial winding displacement. As the valve component A adopts continuous limiter, its initial winding displacement 1.321 mm is more forward than that of valve component B 2.115 mm with discontinuous limiter. The initial valve stiffness in valve component A is 0.727 N/mm, and that of valve B is 2.018 N/mm. Since the initial winding displacement of valve component B is 2.115 mm, which is close to the lift of 2.2 mm, the valve is basically in a constant state. According to the relationship between the valve stiffness and displacement, the relationship between the elastic force and displacement in Figure 8b can be calculated. It can be seen from the figure that the elastic force in valve component A first increases linearly and then increases sharply, while the elastic force in valve component B almost always increases linearly. According to the relationship between the elastic force and the displacement, the work done by the elastic force in the process of movement can be obtained. The work of valve from 0 to 2 mm in valve component A and B are 12.13 mJ and 4.04 mJ, respectively. Although the initial valve stiffness of valve component B is higher than that of valve component A, the work of valve component B is lower than that of valve component A because of the difference of limiter profile. In addition, the maximum bending stress is calculated. The results indicated that the maximum bending stress of valve component A increases linearly before the displacement of 1.321 mm, and keeps constant at 363 MPa when it reaches the winding point. However, the maximum bending stress of valve component B increases linearly all the time, and the maximum value reaches 601 MPa, which is higher 238 MPa than that of valve component A.

![Figure 8](image)

(a) Valve stiffness  
(b) Elastic force  
(c) Maximum stress

**Figure 8.** The relationship between the stiffness, elastic force and maximum stress of valve and the valve displacement.

It can be seen that the limiter profile has an importance on the winding displacement, valve stiffness, elastic force, work and maximum stress. For continuous limiter, the valve can show variable stiffness characteristics in the process of movement, while it is always in the state of equal stiffness for discontinuous limiter.

3.2 Simulation Verification

In order to verify the accuracy of the above theoretical methods and results, finite element method (FEM) is used to simulate the mechanical and deformation of the continuous limiter and discontinuous limiter coupling valve. By applying uniform load step from small to large on the valve head, the valve displacement under different loads is calculated, and then the valve stiffness is calculated and compared with the above theoretical results.

Figure 9 shows the present investigation and FEM results of the two valve components. For valve component A, the initial valve stiffness obtained by the FEM is 0.725 N/mm, which is slightly lower than the present valve 0.727 N/mm, and the calculation error is 0.28% based on the FEM data. For
valve component B, the initial stiffness of the valve obtained by the FEM is 2.100 N/mm, which is slightly lower than the present valve 2.018 N/mm, and the calculation error is 3.90% based on the FEM data. It can be seen that the valve stiffness value obtained by theoretical method is relatively accurate. In Figure 9, the elastic force obtained by the FEM also presents a process of linear increase first and then sharply increase, which is consistent with the present investigation. However, the results obtained by the two methods have some errors, and the FEM has greater deformation under the same elastic force after valve reaching the winding displacement, the maximum difference is less than 0.2 mm. This is because the elastic deformation of the limiter is considered in the FEM, while the theoretical calculation method treats the limiter as a rigid body, so it is normal to have this error.

3.3 Experimental Verification
In order to further verify the accuracy of the mechanical model in this paper, the high-speed photography (HSP) experiment of valve movement was carried out. The opening process of the valve under the action of continuous limiter and discontinuous limiter was observed, as shown in Figure 10. During the experiment, a thin rod was used to manually open the valve head from the flange hole. A high-speed camera was used to observe the valve component from the side. The sampling rate of the high-speed camera was set at 3000 s⁻¹, and the resolution was 1024 × 768. It can be seen from the results that the opening process of the valve under the continuous limiter is gradually around the limiter, while under the discontinuous limiter, the valve directly impacts with the limiter at the neck. Figure 10c shows the image of the valve and limiter for valve component B at the winding moment. The winding position is at 15.8 mm away from the starting point, which is very close to the theoretical value 14.8 mm. When the valve impacts on the limiter at the neck, clearly clearance can be found between the valve and limiter at the waist. This phenomenon is consistent with the theoretical analysis process, which proves the accuracy of the mechanical model in this paper again.
3.4 Structural Optimization

Due to the importance of limiter profile, the discontinuous limiter profile in valve component B is optimized. When the center distance and lift are kept unchanged, the limiter profile is adjusted to continuous type with the limiter curvature radius changed from 130 mm to 96.6 mm, as shown in Figure 11.

The initial winding point was obtained in Figure 11. Compared with original structure, the optimized structure changes the initial winding position from $x = 14.8$ mm to $x = 0$, and change the winding displacement from 2.115 mm to 1.453 mm. The optimization greatly advances the winding effect of valve and limiter, and avoids the impact of the valve on limiter.

![Figure 11. Optimization of limiter profile and the initial winding position of valves.](image)

Figure 11 shows the change relationship between the stiffness, elastic force and maximum stress of valve with the valve displacement.

Figure 12 shows the change relationship between the stiffness, elastic force and maximum stress of valve with the valve displacement before and after the limiter optimization. It can be seen that the stiffness and elastic force begin to change sharply after 1.453 mm, which makes the limiter have better restraint effect. The work of valve from 0 to 2 mm for original and optimized structure are 4.04 mJ and 7.42 mJ, respectively, and the work of valve in the optimized structure increased 83.7% than it in
the original structure. After optimization, the maximum bending stress of valve is reduced to 414 MPa, which is reduced by 187 MPa and 31.1% compared with 601 MPa before optimization.

4. Conclusions
In this paper, the mechanical and deformation characteristics of the discharge valve under the restriction of limiter are studied by means of theory, simulation and experiment. The main conclusions are as follows:

1. The mechanical model of discharge valve considering the coupling effect of limiter is established, and the important parameters such as displacement, stress, stiffness, elastic force and work of valve are obtained.

2. The results show that the limiter profile has a significant effect on the movement of valve. For continuous limiter, the valve is gradually winded with limiter and shows variable stiffness characteristic, while the valve directly impacts the limiter and always in the state of equal stiffness for discontinuous limiter.

3. The error between the present investigation and the FEM is less than 3.9%, and the HSP results show that the motions of valve are consistent with the model results.

4. The limiter profile in component B is optimized. After optimization, the initial position of the valve is advanced from 2.115 mm to 1.453 mm, which shows a better constraint effect.

Acknowledgments
The authors gratefully acknowledge the support of the Guangdong Basic and Applied Basic Research Foundation (2019A1515010787).

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