Optimum Design on Structural Parameters of Reciprocating Refrigeration Compressor Crankshaft

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ABSTRACT: Crankshaft is one of the main components of a piston refrigeration compressor. Its structure parameters influence the reliability and service life. [Research objective] Improve the reliability of the crankshaft and further understand the impact of structural parameters on the strength of the crankshaft. [Research method] The forces acting on crankshaft are analyzed on the basis of dynamic calculation of reciprocating compressor working process. Finite Element Method (FEM) is applied for three-dimensional modeling and static strength analysis. [Research result] The stress at the transition fillet of crankshaft journal under the maximum comprehensive piston force is obtained accordingly. The reliability and accuracy of this model is verified by the experimental results. The influence of the crankshaft with crankpin diameter, center distance of adjacent crankpins and fillet radius on stress concentration has been investigated in detail. The optimal undercut fillet is designed for reducing the maximum stress obviously. The results provide a reference for similar design and production of compressor crankshaft.

Keywords: crankshaft; crankpin; Finite Element Method (FEM); stress analysis; optimal design

1 INTRODUCTION

The crankshaft as the most important moving part of a piston compressor plays a very important role in the compressor. There are many reasons affect the overall performance of the crankshaft because its complex geometry and serious stress concentration phenomenon. Failure examples show that the main failure form is fatigue failure which caused by the alternate load though connecting rod plays on the crankpin. The alternating load includes gas power, friction, reciprocally rotating inertia force and input torque. According to the literature, crankshaft breakages mainly appear at the transition arc of main shaft and crankpin (Silva F S. 2003). Therefore, how to reduce the maximum stress concentration becomes an indispensable content in design of the crankshaft. With the development of finite element method (FEM) and computer technology, stress and deformation changes with structural parameters of crankshaft can be more clearly described by FEM software, which becomes a popular means in improving the level of crankshaft design.

Lots of studies have been made to analyze the stress and deformation of the crankshaft in compressors and engines. Li, H. Z. et al. (1991) presented the optimum design of crankshaft fillets by means of finite element and mathematical programming methods for 6120Q gasoline engine. The maximum stress at the recommended crankshaft fillet reduced considerably. Cordon (1997) analyzed the stress of crankshaft fillets by three-dimensional finite element method, and optimization values of the fillet radius were obtained. Hu, Y.P. (2006) developed an improved crankshaft simulation model of 6160 diesel engine to predict crack position and fatigue life.

Research on the static strength and fatigue strength analysis is the main content of crankshaft design. The static strength analysis is conducted to obtain the stress distribution under the maximum load permitted and identify the dangerous section. Su, T. X. et al. (1995) have done a numerical simulation focusing on the bending stress of crankshaft associated with the oil holes location and transition radius. The research found that stress at the crankpin jour-
nal fillet could be lightened by increasing oil holes diameter or lowered oil holes position appropriately. Wang, D.H. (1997) comparatively analyzed the static strength with four kinds of FEM model and boundary condition of crankshaft. He proposed that the bending moment and torsion moment in crankshaft journal have great influence on the calculation results. Y.KANG et al. (1998) established the finite element model of crankshaft both in entity and beam elements. By comparison with the experimental data, entity elements model is the better one. Feng, G.S. et al. (2003) investigated the distortion and stress of 4190 diesel engine’s crankshaft. Compared with the whole crankshaft model, the reliability of single crank stress analysis method was undesirable due to its simplified boundary conditions. Choi K.S. (2009) analyzed static strength, fatigue strength and vibration characteristics of the crankshaft combined with the finite element method and modal. The result verified the accuracy of numerical analysis calculations precisely.

Many scholars have studied the fatigue strength and fatigue life of crankshaft as accurately as possible. D. Taylor, W (1999) presented an approach based on equivalent stress intensity for analyzing the static stress and fatigue strength. Lee Y.L (2001) calculated the fatigue strength and safety coefficient of the crankshaft. Both the simulation results and experimental results are consistent. Zhang, G. Q et al. (2006) have done parametric studies on fatigue life of crankshaft using nominal stress method and local stress method respectively. The results indicated that the traditional design is too conservative and not accurate enough. Yang Z (2014) analyzed the dynamic characteristics and load spectrums of the crankshaft combined with the finite element method and multi-body dynamics technique. The fatigue life and failure danger section of the crankshaft were obtained. All of these works lay a foundation for the research of the crankshaft strength deeply.

In this paper, the influence of crankpin diameter, center distance of adjacent crankpins and crankshaft journal fillet radius on stress concentration has been studied based on Finite Element Method (FEM). The undercut fillet is designed for optimizing the life and reliability of crankshaft. It provides a theoretical basis for the optimum design of compressor crankshaft.

2 STRENGTH ANALYSIS OF THE CRANKSHAFT

A crankshaft used in a medium-sized semi-hermetic piston compressor is adopted for the research. The reciprocating mass of the compressor is 0.4756kg; the compressor rated speed is 1,450r/min and the rated power is 2.2kW. The main dimensions of the crankshaft are shown in Table 1.

The crankshaft material is 40Cr steel. The main physical constants of 40Cr steel are shown in Table 2.

Table 2. The physical characteristics constants of 40Cr steel.

<table>
<thead>
<tr>
<th>Material</th>
<th>Elastic modulus E/Gpa</th>
<th>Poisson’s ratio µ</th>
<th>Density ρ/g·cm-3</th>
</tr>
</thead>
<tbody>
<tr>
<td>40Cr steel</td>
<td>198</td>
<td>0.29</td>
<td>7.82</td>
</tr>
</tbody>
</table>

2.1 Three-dimensional modeling and meshing

Three-dimensional modeling is precisely constructed for calculating the strength of whole crankshaft based on Pro/Engineer software. Taking into account the meshing grid at the junction of oil channels and, shaft necks are prone to adverse elements, the oil passages are ignored in the model (Okamura H, 1995). Since hexahedral meshing has more advantages in calculation precision, deformation characteristics and anti-distortion degree than tetrahedral meshing, the finite element model of crankshaft is hexahedral meshed in HyperMesh software with a total of 219,095 nodes and 612,220 elements.

2.2 Force boundary conditions and constraints

The force boundary conditions of finite element model include loads, bearing radial constraints and input torque, etc (Tadashi N, 2002). The force acting on piston pin mainly contains air force $F_p$, reciprocating inertia force $F_j$ and friction force $F_f$. Since the forces’ directions are all along the central line of the cylinder, the equations of comprehensive piston force $F_p$ can be written as follows:

$$F_p = F_g + F_j + F_f$$  \hspace{1cm} (1)

Figure 1. Schematic of crank linkage force.

The maximum stress appears when comprehensive piston force is the largest. This paper focuses on static
stress analysis under the condition of the largest comprehensive force, which provides reasonable values to assure the simulated results are physically meaningful.

Figure 1 presents that the total force $F_a$ imposed on piston are decomposed into connecting rod force $F_l$ and lateral force perpendicular to the cylinder wall $F_h$. The $F_l$ as well as $F_h$ can be expressed by the ratio of crank radius and connecting rod length $\lambda$ and crank angle $\alpha$ as follows:

\[ F_l = F_a \cdot \frac{1}{\sqrt{1 - \lambda^2 \sin^2 \alpha}} \quad (2) \]

\[ F_h = F_a \cdot \frac{\lambda \sin \alpha}{\sqrt{1 - \lambda^2 \sin^2 \alpha}} \quad (3) \]

The force $F_l$ acting on crankpin center through the connecting rod are decomposed into the circumferential tangential force $F_t$ and radial normal force $F_r$. These two forces can be calculated as:

\[ F_t = F_a \left( \sin \alpha + \frac{\lambda \sin 2\alpha}{2\sqrt{1 - \lambda^2 \sin^2 \alpha}} \right) \quad (4) \]

\[ F_r = F_a \left( \cos \alpha - \frac{\lambda \sin^2 \alpha}{\sqrt{1 - \lambda^2 \sin^2 \alpha}} \right) \quad (5) \]

The simplified forces model of crankshaft is shown in Figure 2. The tangential force $F_t$ and normal force $F_r$ were applied on the center node of crankpin.

The constraints of crankshaft are specified as follows:

Displacement constraints are arranged to the short end of main shaft for limiting the axial movement.

The power input end of crankshaft needs to make steel processing for reducing stress concentration on the center node.

According to the formula $T = F \times D_1$, input torque $T$ can be described as the tangential force $F$ exerting on the circumference face of power input, the moment arm is $D_1/2$. The loads on crankshaft are shown in Table 3.

### Table 3. Loads on crankshaft.

<table>
<thead>
<tr>
<th>Crankshaft position feature</th>
<th>The first crankpin</th>
<th>The second crankpin</th>
<th>Input torque $T$ (Nm)</th>
<th>Input Force $F$ (N)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Normal force $F_r$ (N)</td>
<td>-595.05</td>
<td>-7325</td>
<td>356.78</td>
<td></td>
</tr>
<tr>
<td>Tangential force $F_t$ (N)</td>
<td>9949</td>
<td>244.2</td>
<td>8831</td>
<td></td>
</tr>
</tbody>
</table>

2.3 Result analysis

After arranged the load boundary conditions and constraints, the static stress distribution of crankshaft is calculated in ANSYS under the condition of maximum piston comprehensive force. It was found that the maximum stress was 83Mpa at the journal fillet of second crankpin, which is located in the vicinity of 300 degrees counter-clockwise viewing from motor. According to the experimental results, the destruction of crankshaft commonly occurs at transition fillet of crankshaft journal, which is consistent with the calculated results. Therefore, the finite element model of static stress analysis is valid correspondingly. Stress distribution of the crankshaft is shown in Figure 3:

Figure 2. Simplified forces model of crankshaft.

Figure 3. Stress distribution of crankshaft.
3 PARAMETERS OPTIMIZATION OF THE CRANKSHAFT

The structural parameters of crankshaft impact on the strength are studied by modifying the geometric dimensions of the finite element model. Optimization of the crankshaft structural parameters needs to integrate the consideration of crankshaft strength, stiffness, weight and reliability. Since the maximum stress emerged when comprehensive piston force is the largest, the static strength analysis is carried out under this condition. In this section, we emphasize our analysis in structure parameters as crankpin diameter, center distance of adjacent crankpins and common fillet radius inflecting on the maximum stress of crankshaft.

3.1 Crankpin diameter

The changes of maximum stress with the crankpin diameter increased have been studied based on the main shaft length and thickness of crank arm invariably. Figure 4 presents the maximum stress at the journal fillet, which is reduced with the increase of crankpin diameter. Crankpin diameter is also affected by bearing diameter and quality, the overall stiffness and weight of the crankshaft subsequently. The crankpin diameter could be improved only in permit conditions.

![Figure 4. Maximum stress changed with crankpin diameter increase.](image)

3.2 Center distance of adjacent crankpins

The changes of maximum stress with the increasing center distance of adjacent crankpins have been also studied based on main shaft length, crank thickness and crankpin diameter invariably. Figure 6 presents the maximum stress at the journal fillet, which is generally increased with the center distance growth. However, reducing the center distance of adjacent crankpins will shorten the stroke of connecting rod and resulted in lower volumetric efficiency of the compressor. Therefore, the structural parameters of stroke / bore size ratio should be reasonable chosen for the relative clearance volume from 1.5% to 6% in design of the small–and-medium-sized compressor.

![Figure 6(a-b). Stress distributions when center distance of adjacent crankpins is 68mm or 70mm.](image)

3.3 Common fillet radius

![Figure 7. Maximum stress changed with the increase of common fillet radius.](image)
The size, shape and surface quality of the crankpin journal fillet have significantly affected on stress concentration and fatigue strength. The maximum stress can be gradually reduced with the appropriate increase of common radius $R$, as shown in Figure 7. When $R$ increased over 3mm, the effect of stress reduction is not obvious, and length of effective pressure will be shortened correspondingly and unfavorable for the crankpin working. Therefore, the crankshaft fillet radius should not be too large.

3.4 Undercut fillet

In order to solve the above problem, the undercut fillet was designed for reducing the maximum stress. And the structure of undercut fillet is shown in Figure 8:

![Figure 8. Undercut fillet of crankshaft.](image)

The undercut fillet is arranged within the fillet of crank pin and crank, which is recommended when the axial length of the compressor is strictly limited such as angle-type compressors. Meanwhile, the undercut fillet should be avoided cutting too deep otherwise it will weaken the strength of the crank.

![Figure 9. Comparison of the maximum stress between undercut fillet and common fillet.](image)

Figure 9 shows the comparison of maximum stress between common fillet and undercut fillet. The black line represents the maximum stress varying with the increase of crankpin diameter when common fillet $R$ is 2mm. It was found that the maximum stress is very strong when crank pin diameter is small and then declined rapidly with the diameter increase. There is a large variation range of the maximum stress. The red line represents the maximum stress changed with the increase of crank pin diameter when undercut fillet $r$ is 0.8mm. However, the maximum stress at the journal of undercut fillet decreased generally and continued its downward trend when the diameter is less than 45.5mm. The maximum stress of undercut fillet exceeds that of common fillet when diameter is larger than 45.5mm. Therefore, undercut fillet could be selected for reducing the stress concentration according to the diameter of crank-pin under the premise of ensuring the overall strength of crankshaft.

![Figure 10(a-d). Stress distribution on crankshaft with the increasing of crankpin diameter (D) when undercut fillet r=0.8mm.](image)

Figure 10(a-d) shows the stress distribution on crankshaft with the increase of crankpin diameter (D) when undercut fillet $r=0.8$mm.
when undercut fillet (r) is 0.8mm. The maximum stress occurs in the undercut fillet of crankshaft journal, which is generally lower than common fillet. The maximum stress is 58MPa, 57MPa, 54MPa or 53MPa when crankpin diameter (D) is 44.4mm, 44.5mm, 45mm or 46mm.

4 CONCLUSIONS

In this study, static stress analysis is conducted for a whole crankshaft of reciprocating refrigeration compressor. The weak position of strength was found in crankshaft journal fillet, where reached 83MPa at the maximum comprehensive piston force. The influences of crankpin diameter, center distance of adjacent crankpins and journal fillet radius on the maximum stress of crankshaft have been investigated. The main conclusions are listed as follows:

The increase of crankpin diameter causes a significant decrease in fillet stress when the thickness of the crank arm is unchanged. With the center distance of adjacent crankpins shortened, the fillet stress decreases slowly.

The maximum stress can be decreased with the crankpin fillet radius R increase. When R increased over 3mm, the effect of stress reduction is not obvious, and the journal pressure area is reduced, which is unfavorable for the crankpin working. Therefore, the crankshaft fillet radius should not be too large.

To decrease the maximum stress of crankshaft, the undercut fillet structure of the crankpin was designed. The static stress analysis of common fillet and undercut fillet are compared and found that the maximum stress at the crankshaft journal was significantly decreased when crankpin diameter is 44.4mm. Once the diameter exceeds 45.5mm, the stress in common fillet is smaller than that of undercut fillet. Therefore, less stress can be obtained in the range of 44.4mm ~ 45.5mm of the crankpin diameter. Furthermore, more accurate stress analysis data of crankpin fillets was obtained in this study, which provides a theoretical basis for the improvement of small-and-medium-sized compressor crankshaft.

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REFERENCES


