Simulation and Design of the ZW Type Liquefied Petroleum Gas Compressor Crankshaft

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Abstract—The crankshaft is the key parts of piston compressor, the fatigue strength of the crankshaft of compressor for the accurate analysis can greatly enhance the operation reliability of the compressor. Taking ZW-0.8/10-16 air-cooled vertical two-cylinder single-acting liquefied petroleum gas compressor as an example, the Solidworks software has been utilized to carry out the three-dimensional model of the crankshaft firstly, and then the risk point of fatigue failure of the crankshaft may be found by the compressor crankshaft finite element analysis. The optimization model by ANSYS software has been established, and the optimized crankshaft strength are verified at last. The results show that the finite element method can be more accurately for crankshaft fatigue analysis, and thus it provides certain theoretical basis for the design and improvement of the crankshaft.

1. INTRODUCTION
The crankshaft is one of the important components of the piston compressor, and its main function is to transform the rotary motion of the motor into the reciprocating motion of the piston in the cylinder. With the development of energy industries, such as petrochemical, fertilizer, oil refining, natural gas and other energy industries, reciprocating compressors are required to become larger and more parameterized, and adapt to variable operating conditions [1]. How to ensure the performance and life of key components such as crankshaft and prevent crankshaft’s failing and even the happening of fracture accidents has become the core issue of the design [2].

The crankshaft of the ZW-0.8 / 10-16 compressor uses an integral crankshaft. The main technical performance parameters of this model are shown in Table 1. When the crankshaft is working, the working conditions are harsh, and it can withstand the alternating composite load of tension, compression, shear, bending, and torsion [3]. When the compressor is in operation, the bending and twisting crankshaft will cause crankshaft fatigue failure, which once may affect the operation of the entire compressor. Therefore, it is of great guiding significance for the design and optimization of crankshaft to obtain the magnitude of the stress and deformation of the crankshaft more accurately [4].

| Table 1 Main technical performance parameters of compressor |
|-------------------------------|-----------------|
| **Model** | **ZW-0.8 / 10-16** |
| Main compressed medium | Liquefied petroleum gas |
| Nominal volume flow (m3 / min) | 0.8 |
| Speed (r / min) | 550 |
| Inspiratory pressure (MPa) | 1.0 |
| Exhaust pressure (MPa) | 1.6 |
| Suction temperature (℃) | ≤50 |
| Exhaust temperature (℃) | ≤110 |
2. EASE FORCE ANALYSIS AND THEORETICAL CALCULATION OF CRAN

2.1. Motion analysis
The schematic diagram of the movement of the crank linkage mechanism of ZW compressor is shown in Figure 1 [5].

\[ x = r \left( 1 - \cos \alpha \right) + \frac{\lambda}{4} \left( 1 - \cos 2\alpha \right) \]  

Equation (1) can be obtained from the movement law of the crank link mechanism:
- \( x \) - the distance from the center point of the piston to the outer dead center
- \( r \) - crank radius
- \( \alpha \) - Crank angle
- \( \beta \) - connecting rod swing angle, that is, the angle between the centerline of the cylinder and the centerline of the connecting rod
- \( \lambda \) - The ratio of the crank radius to the radius of the connecting rod, which is 1/4 for this compressor.

2.2. Theoretical calculation of the main force of the crankshaft
In order to make a simple analysis of the force, the force of the crankshaft is simplified as follows:
1) For the double-supported crankshaft, consider the bearing's elastic support of the crankshaft;
2) Consider the action of the connecting rod on the crankshaft at the midpoint of the crank pin;
3) Do not consider crankshaft’s dead weight.

2.2.1 Reciprocating inertial force
\[ F_i = (m_p + m_r)a = m_p r \omega^2 \sqrt{(\cos \alpha + \lambda \cos 2\alpha)} \]  

Equation (2) can be written as:
- \( m_p \) - mass of reciprocating parts
- \( m_r \) - unbalanced rotating mass
- \( \omega \) - crank angular velocity

2.2.2 Comprehensive piston force
\[ F_p' = F_p + F_i \]  

2.2.3 Crank pin force. As can be seen from Figure 1, the crank pin force \( F_i \) is the component of the integrated piston force along the centerline of the connecting rod [5]:
\[ F_{L} = \frac{F'_p}{\cos \alpha} \]  

Equation (4) can be written as:
- \( F_{L} \) can be decomposed into two directions of force, namely the tangential force \( F_T \) perpendicular to the crank and the normal force \( F_R \) [6] along the radial direction of the crank.
The magnitude and direction of FT and FR change with the change of the crankshaft angle $\alpha$. When the tangential force is opposite to the rotation direction of the crankshaft, the value of FT is positive; when the normal force is directed from the inside to the outside, the value of FR is positive.

2.3. Pulley axial force

ZW type compressor transmission mode uses belt transmission, and the force of Fr will be acted on the crankshaft by the pulley.

$$F_r = 2F_0 z \sin \alpha_i$$  \hspace{1cm} (7)

$F_0$-Single tension of single V-belt
$z$-band number
$\alpha_i$-Roll wrapping angle

$$F_0 = 500 \frac{P_0}{v^2} \left( \frac{2.5}{k} - 1 \right) + qv^2$$  \hspace{1cm} (8)

$v$-line speed of the belt
$P_0$-power of belt drive
$k$-wrapping angle coefficient
$q$-mass with unit length

3. Finite element analysis of crankshaft based on ANSYS software

3.1. Crankshaft parameter attributes

The crankshaft is supported by double-row roller bearings, the diameter of the main journal is 60mm, the diameter of the connecting rod journal is 62mm, and the total length of the crankshaft is 458mm. The material used for this crankshaft is 45 carbon structural steel, and its mechanical properties [7] are shown in Table 2.

| Attribute name         | Value                  |
|------------------------|------------------------|
| Modulus of elasticity  | $2.05 \times 10^{11}$N/m$^2$ |
| Modulus of elasticity  | 2.9                    |
| Shear modulus          | $8.0 \times 10^6$N/m$^2$|
| Mass density           | 7850kg/m$^3$           |
| Tensile strength       | $6.25 \times 10^8$N/m$^2$|
| Yield strength         | $5.3 \times 10^8$N/m$^2$|

3.2. Applying loads and constraints

According to theoretical calculations and practical experience, it is known that when the compression process ends, the stress and deformation of the crankshaft is the largest, that is, when the piston is at the top dead center position, the comprehensive force on the crankshaft reaches the maximum. Therefore, when the crankshaft is subjected to a force analysis to obtain the maximum value, we only need to consider the force situation at the top dead center position [8].

The restriction of the crankshaft mainly depends on the roller bearings at both ends of the crankshaft. The bearings will support the crankshaft. The bearing's support of the crankshaft can be considered as an elastic support. For this type of compressor, the value of the spring stiffness $k$ is 1.5.

3.3. Selection and division of grids

A three-dimensional model of the crankshaft has been built using Solidworks software and imported into ANSYS Workbench [9]. The structure of the crankshaft is relatively simple and belongs to the
geometric specification. Therefore, a hexahedron is used as the main grid. In order to improve the accuracy, the coarse is set to be 60. The finite element mesh model of the double crank crankshaft is shown in Figure 2.

![Finite element mesh model of double crank crank.](image)

### 3.4. Results and data analysis

The equivalent stress distribution of the crankshaft is shown in Figure 3. Under actual operating conditions of the crankshaft, the stress is mainly concentrated at the connection between the crankpin and the crank. The maximum stress is 422 MPa, which is less than the yield limit of the crankshaft 530 MPa, so the strength of the crankshaft meets the requirements of the working conditions. However, a too long working cycle will cause fatigue damage to the crankshaft. For the above-mentioned stress concentration areas, it may be considered to appropriately handle the connection between the crankshaft pin and the crank, such as adding a boss, implementing an excessive arc, changing the diameter of the crankpin, and achieving the effect of reducing stress [10].

![Equivalent stress distribution of crankshaft.](image)

### 4. Optimization of the Crankshaft

#### 4.1. Optimization of the crankshaft

Based on the consideration of the damage caused to the crankshaft by stress concentration, the height of the boss, the radius of the transition arc and the diameter of the crankpin are defined as the input parameters, and the maximum stress is the output parameter. After the simulation is completed, the relationship curve of the influence of the height of the boss, the radius of the transition arc, and the diameter of the crankpin on the stress can be obtained, as shown in FIG. It can be seen from the figure that DS-3 has the greatest impact.

#### 4.2. Optimization of the crankshaft

The ANSYS software was used to analyze the equivalent stress of the crankshaft with reloaded input parameters. The equivalent stress distribution is shown in Figure 5. It can be obtained through
optimization results, the maximum stress of which is 371 MPa. Therefore, under actual working conditions, appropriately increasing the boss at the joint between the crankshaft pin and the crank, and selecting an appropriate transition arc radius and diameter of the crankpin will reduce the crankshaft stress and the stress concentration. The single increase of the height of the boss has little effect on reducing the stress. The main factors are the radius of the transition arc and the diameter of the crankpin. When the radius of the transition arc is constant, the larger the diameter of the crankshaft pin is, the smaller the stress will be. When the diameter of the crankpin is constant, the smaller the radius of the transition arc is, the smaller the stress will be. This greatly reduces the fatigue damage during the working process of the crankshaft, thereby increasing the life cycle of the crankshaft.

Figure 4. Influence of Boss Height, Transition Arc Radius, Crankshaft Pin Diameter on Stress.

Figure 5. Equivalent stress diagram of crankshaft with reloaded input parameters.

5. CONCLUSION
In order to verify whether the strength and stiffness of the crankshaft meet the requirements, we first use Solidworks software to establish a solid model of the crankshaft, and then import ANSYS software to perform a static analysis on the crankshaft to obtain a stress distribution cloud diagram of the crankshaft and perform data analysis.

Through theoretical calculation and finite element analysis of the crankshaft of ZW-0.8 / 10-16 compressor under variable working conditions, its dangerous cross-section location is determined, which provides a basis for the proposal of the optimization scheme.

Based on the results of static analysis, an optimization scheme for crankshaft is proposed. Then use ANSYS software to carry out optimization analysis to obtain the optimized stress distribution cloud diagram, verify the rationality of the optimization scheme, and provide a theoretical basis for the design and improvement of the crankshaft.
Using Solidworks software and ANSYS simulation software to model and simulate the crankshaft of ZW-0.8/10-16 compressor. Compared with traditional theoretical calculations, it can more intuitively reflect the movement process and characteristics of the crankshaft. It provides a reference for further analysis, optimization and design.

ACKNOWLEDGMENT
This work is supported by the Key Research Project of Natural Science of Bengbu University under Grant No.2017ZR09.

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