Design and Verification Test of the Primary and Secondary Stage Compression Gap Size of Miniature High Pressure Air Compressor

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Abstract. Aiming at the prominent problem of the primary and secondary pistons hitting the cylinder head in the current miniature high-pressure three-stage air compressor, a design study on the size of the primary and secondary compression gap is carried out. Through the theoretical force analysis of the connecting rod, the force situation of the connecting rod at the top and bottom dead centers is obtained. The three-dimensional models of connecting rod and cylinder are established based on SolidWorks software, and the three-dimensional models are simplified. Based on Ansys Workbench software, the connecting rod and cylinder models are pre-processed, model discretized and post-processed. The structural statics and steady-state thermal comprehensive analysis of the connecting rod at the upper and lower dead points and the steady-state thermal analysis of the cylinder under working conditions are completed. Through analysis, the total axial deformation of the connecting rod and the total axial deformation of the cylinder under high temperature and high speed working conditions are obtained. According to the simulation analysis results, the first and second compression gap size is designed. Based on the design results, ten sets of air compressors are tested for dimensional design verification. The test results show that the air compressors of each test group can complete the charging requirements within the specified time, and there is no phenomenon of piston hitting the cylinder head, indicating the size design reliability.

Keywords: Mini high-pressure air compressor, connecting rod, cylinder, finite element analysis, verification test.

1. Introduction
High-pressure gas has a wide range of applications in petroleum, chemical, shipbuilding, military, rescue, food and other fields. It is mainly used for cooling, diving breathing, pneumatic ejection, natural gas systems, etc. [1]. The main characteristics of compressors currently used to output high-
pressure gas are large size, heavy weight, and oil lubrication. Mobile devices with strict requirements on volume and weight cannot be applied; at the same time, they cannot adapt to high power density ratios, adapt to large temperature ranges, and avoid post-production. Requirements for special conditions such as maintenance. The application of micro high-pressure compressors in aerospace, marine and natural gas fields has significant advantages such as fast response and high efficiency [2]. With the development of national defense technology, there are more and more customized designs based on functional requirements. Among them, high-pressure and lightweight small and micro compressors have always been high-precision products for compressor equipment [3].

Compared with traditional high-pressure compressors, micro-high-pressure compressors are small in size, compact in structure, short in piston stroke and small in diameter of the high-pressure stage piston, so its performance is more sensitive to clearance volume [4]. The assembly of the connecting rod and the cylinder determines the compression gap of the compressor, and the compression gap design of the air compressor directly affects the clearance volume. The connecting rod is one of the key power components of the micro high-pressure air compressor, and it bears complex loads. The design of the connecting rod plays a vital role in the overall operation of the air compressor. Liu Xiaoxu et al. [5] used SolidWorks software to perform 3D modeling, dynamic simulation, finite element analysis and structural optimization on the connecting rod of a reciprocating piston air compressor. Hou Fangyong et al. [6] calculated and analyzed the inertial force of the piston compressor connecting rod. Yang Jian [7] used CATIA V5 software to carry out three-dimensional modeling, finite element analysis and structural optimization of the connecting rod of a small general gasoline engine. Chen Xuyang [8] analyzed the static and dynamic strength of the connecting rod of the internal combustion engine through ANSYS software, and used two methods of topology optimization and parameter optimization to optimize the structure of the connecting rod. Luo Jinghui [9] used Pro/MECHANICA as the analysis platform to optimize the connecting rod quality of the diesel engine to the minimum while meeting the work requirements. The above research provides a reference for the actual force analysis, boundary conditions, load application and post-processing of the connecting rod. Li Qiming [10] analyzed the mechanical thermal load and mechanical load of the compressor cylinder block and optimized the cylinder structure. Yang Tao [11] used the finite element method to analyze the cylinder liner deformation and influencing factors of the diesel engine. The above research provides reference for the actual force analysis, boundary conditions, load application and post-processing of the cylinder.

In this paper, the first-stage and second-stage compression of the miniature high-pressure three-stage piston air compressor is performed through the end and cone surfaces of the first-stage and second-stage pistons to achieve single-cylinder two-stage compression. As the main power mechanism of the air compressor, the connecting rod works under high temperature and high speed. The connecting rod is forced in the axial direction and thermally expands, which causes the axial size of the connecting rod to change, so that the piston actually has a bottom dead center. Offset, eventually leading to too large or too small compression gap. If the first-stage compression gap is too large, the compression ratio will be reduced. If the first-stage compression gap is too small, the first- and second-stage pistons will hit the cylinder head; the second-stage compression gap will cause the first- and second-stage piston cones to hit the cylinder, and the second-stage compression Too small a gap will result in a lower secondary compression ratio. Through the Ansys Workbench software, the structural statics and steady-state thermal comprehensive analysis of the connecting rod and cylinder under working conditions are carried out, and the total axial deformation of the connecting rod at the upper and lower dead points is obtained, and then the primary and secondary clearance dimensions are designed. So as to provide ideas for the overall structure design, reduce the number of tests and reduce the development cost.

2. Theoretical force analysis of connecting rod
Under actual working conditions, the axial resultant force borne by the primary and secondary cylinders is small and can be ignored, and the load borne by the connecting rod is more complicated,
so the main analysis of the force on the connecting rod is, in the static strength analysis of the connecting rod, three types of loads are mainly considered:

1. The reciprocating inertia force of the piston assembly and connecting rod assembly;
2. Centrifugal inertia force of connecting rod assembly;
3. The gas force acting on the piston.

2.1. Calculation of theoretical force of connecting rod
Calculation of the reciprocating inertia force of the piston assembly and the connecting rod assembly: To facilitate the actual calculation, the mass distribution of the connecting rod in the crank connecting rod mechanism composed of the connecting rod and the eccentric wheel can be simplified as shown in Figure 1. The M1 part will be used for linear transmission Movement, the M2 part does a uniform circular motion with the o point as the center.

![Figure 1. Mass distribution diagram of crank connecting rod.](image)

The sum of the masses of M1 and M2 should be equal to the total mass of the connecting rod M3, namely:

\[ M_1 + M_2 = M_3 \]  \hspace{2cm} (1)

In the formula, M3 is the total mass of the connecting rod; M1 is the mass of the upper part of the simplified connecting rod; M2 is the mass of the lower part of the simplified connecting rod.

\[ M_1L_1 - M_2L_2 = 0 \]  \hspace{2cm} (2)

In the formula, L is the center of gravity distance between M1 and M2; L1 is the center of gravity distance between M1 and M3; L2 is the center of gravity distance between M2 and M3.

The calculation result is L1=41.81mm, L2=26.19mm, M1=38.86g, M2=62.05g.
Piston acceleration \( a \) can be calculated from Figure 2:

\[
a = R\omega^2 (\cos\alpha + \lambda \cos 2\alpha + \frac{\lambda^3 \sin^2 2\alpha}{4 \cos^2 \beta})
\]

(3)

In the formula, \( R \) is the crank radius; \( \omega \) is the angular velocity of the crank; \( \alpha \) is the angle of rotation of the crank relative to the top dead center; \( \lambda \) is the rod diameter ratio, \( \lambda=R/F=0.2059 \); \( \beta \) is the deviation of the connecting rod from top dead center Angle.

The piston assembly and the connecting rod assembly M1 make reciprocating linear motion with piston acceleration \( a \), and the acceleration can be approximated by the following formula:

\[
a = R\omega^2 (\cos\alpha + \lambda \cos 2\alpha)
\]

(4)

When \( \alpha=0^\circ \), the piston and connecting rod move to the top dead center, \( a=740.554 \text{ m/s}^2 \); when \( \alpha=180^\circ \), the piston and connecting rod move to the bottom dead center, \( a=-487.67 \text{ m/s}^2 \).

The reciprocating inertia force \( F_m \) of the piston assembly and connecting rod assembly M1 is calculated as follows, where \( m_h \) is the mass of the piston assembly, \( m_h=0.223 \text{ kg} \):

\[
F_m = -(m_h + M_1)a
\]

(5)

When \( \alpha=0^\circ \), \( F_m=-214.39 \text{ N} \).
When \( \alpha=180^\circ \), \( F_m=141.18 \text{ N} \).

Calculation of the centrifugal force of the connecting rod assembly: The M2 part of the connecting rod assembly performs a uniform circular motion, and the centrifugal force \( F_r \) of the connecting rod assembly is calculated as follows:

\[
F_r = M_2 \omega R
\]

(6)

When \( \alpha=0^\circ \), \( F_r=-62.14 \text{ N} \); when \( \alpha=180^\circ \), \( F_r=62.14 \text{ N} \).

Calculation of the gas force acting on the piston: when \( \alpha=0^\circ \), the first-stage compression gap is exhausted, and the second-stage compression gap is inhaled. Therefore, the first-stage compression gap air pressure \( p_1 \) is basically the same as the second-stage compression gap pressure \( p_2 \), so the piston the gas force received is zero.

When \( \alpha=180^\circ \), the first-stage compression gap is inhaled, and the second-stage compression gap is exhausted. Therefore, the first-stage compression gap air pressure \( p_1 \) is consistent with atmospheric
pressure, which is 0.101MPa. According to the theoretical compression ratio, the second-stage compression gap air pressure $p_2$ is 2.786MPa, the calculation formula of gas force $F_q$:

$$F_q = \frac{\pi}{4} \left( D^2 - d^2 \right) p_2 - D^2 p_1$$

(7)

In the formula, $D$ is the major diameter of the piston; $d$ is the minor diameter of the piston. $F_q = -961.217$N

2.2. Analysis of connecting rod force

In summary, the load on the connecting rod is shown in Table 1, and the calculation formula for the force $F$ on the connecting rod is as follows:

$$F = F_m + F_r + F_q$$

(8)

When $\alpha = 0^\circ$, that is, when the connecting rod is at the top dead center, $F = -276.53$N, so the connecting rod will be subjected to a pulling force of 276.53N. When $\alpha = 180^\circ$, that is, when the connecting rod is at the bottom dead center, $F = -757.97$N, and the connecting rod is subject to a pulling force of 757.97N.

Table 1. Connecting rod bearing load table.

| Load type | Reciprocating inertia $F_m$ | Centrifugal inertia force $F_r$ | Gas force $F_q$ |
|-----------|---------------------------|-----------------|----------------|
| Link position | Top dead center | Bottom dead center | Top dead center | Bottom dead center | Top dead center | Bottom dead center |
| Load size /N | -214.39 | 141.18 | -62.14 | 62.14 | 0 | -961.217 |

3. Ansys Workbench simulation

Based on the ANSYS Workbench simulation platform, it can realize the analysis and simulation of the complex mechanical system's structural statics, structural dynamics, rigid body dynamics, fluid dynamics, structural heat, electromagnetic field and coupled fields. The simulation in this paper mainly involves the comprehensive analysis of connecting rods in the structural statics and structural thermal modules in ANSYS Workbench, and uses the structural thermal modules to analyze the cylinder blocks of the first and second cylinders. By studying the difference between the axial deformation of the connecting rod and the primary and secondary cylinders under high temperature and high load, it provides a reference for the design of the primary and secondary compression gap size. The simulation flowchart is shown in Figure 3.
3.1. 3D model creation and simplification

In order to accurately establish a three-dimensional model, first establish a three-dimensional model of the connecting rod and the first and second cylinders in SolidWorks. At the same time, in order to improve the quality of subsequent meshing and the accuracy of simulation results, the model features need to be simplified. By removing the rounding, chamfering and holes with smaller diameters of the model, the problem of poor convergence of analysis results caused by low mesh quality can be effectively avoided. The simplified model is shown in Figure 4.

![Simplified 3D models](image)

3.2. Pre-processing of CAE model

Select the "Static Structural" module in Ansys Workbench, and import the established model in "Geometry", and at the same time complete the division of the force area. This article claims that the connecting rod of the micro air compressor is forged with 12CrNi3A alloy steel, and the primary and secondary cylinders are casted with ZL105 aluminum alloy. The properties of the two materials are shown in Table 2.
Table 2. Material attribute table.

| Material  | tensile strength/MPa | Yield Strength/MPa | Elastic Modulus/GPa | density/kg·m⁻³ | Poisson's ratio | Thermal expansion coefficient 1/°C |
|-----------|----------------------|--------------------|---------------------|-----------------|----------------|-------------------------------------|
| 12CrNi3A  | 930                  | 685                | 200                 | 7850            | 0.275          | 1.9x10⁻⁵                           |
| ZL105     | 225                  | /                  | 70                  | 2750            | 0.34           | 2x10⁻⁵                             |

According to the data in Table 2 and Table 1, add the new material "12CrNi3A" in "Engineering Data", and add constraints and loads in "Model". When the connecting rod moves to the top dead center, in order to simulate its actual working conditions, force is applied to the contact surface of the connecting rod and the piston pin, as shown in Figure 5(a); by applying displacement constraints on the contact surface of the connecting rod and the crankshaft, so that the displacement in the three directions of X, Y, and Z is 0, as shown in Figure 5(b); referring to the temperature in the cylinder under the actual test conditions, set the connecting rod thermal load to 150°C, as shown in Figure 5(c). As shown. When the connecting rod moves to the bottom dead center, in order to simulate its actual working conditions, force is applied to the contact surface of the connecting rod and the piston pin, as shown in Figure 6(a); the displacement constraint is imposed on the contact surface of the connecting rod and the crankshaft, As shown in Figure 6(b); make the displacement in the three directions of X, Y, and Z to be 0, and finally apply a 150°C thermal load, as shown in Figure 6(c).

![Figure 5](image1.png)

Figure 5. Connecting rod load and restraint setting at top dead center.

![Figure 6](image2.png)

Figure 6. Connecting rod load and restraint setting at bottom dead center.
Similarly, add new material "ZL105" in "Engineering Data", and add constraints and loads in "Model". Appropriate displacement constraints are imposed on the model, as shown in Figure 7(a). The cylinder temperature is 90°C under the actual test conditions, so the corresponding thermal load is applied to the cylinder model, as shown in Figure 7(b).

3.3. Model discretization
According to the actual size of the connecting rod model, the mesh size is selected as 0.50mm, the mesh transition is selected as slow, and the tetrahedral mesh is selected to mesh the model through the intelligent meshing tool. The number of mesh nodes in the divided connecting rod model is 619515 and the number of units is 405794. The result of the division is shown in Figure 8(a). According to the actual cylinder model size, the mesh size is selected as 0.80 mm, the mesh transition is selected as slow, and the tetrahedral mesh is selected to mesh the model through the intelligent meshing tool. The number of mesh nodes in the cylinder model after division is 3574399, and the number of units is 2320304. The division result is shown in Figure 8(b).

3.4. Post-processing and analysis of results
Post-process the model, add "Directional Deformation", set the direction to the X axis, that is, the displacement of the connecting rod in the X-axis direction, post-process the connecting rod model, and get the post-processing result as shown in Figure 9. It can be seen from the simulation results that
when the connecting rod is at the top dead center, its maximum displacement is 0.25998 mm, and its minimum displacement is -0.018683, that is, the connecting rod is elongated by 0.278663 mm in the X axis. When the connecting rod is at the bottom dead center, its maximum displacement is 0.25114 mm, and its minimum displacement is -0.017329 mm, that is, the connecting rod is elongated by 0.268469 mm in the X-axis. After post-processing the primary and secondary cylinder models, the simulation results shown in Figure 9c) can be obtained. The results show that the maximum displacement of the primary and secondary cylinders is 0.14095 mm, the minimum displacement is -0.00015584 mm; the displacement at the secondary gap is 0.062556 mm, and the primary and secondary cylinders are elongated by 0.14111 mm in the axial direction as a whole, and at the secondary compression gap it is elongated to 0.0622711 mm.

Figure 9. Post-processing results of axial displacement.

4. Gap size design and verification test

Figure 10 shows the schematic diagram of adjusting the compression gap of the air compressor. It can be seen from the figure that a reasonable gap size design needs to adjust the primary and secondary compression gaps to a reasonable height, that is, to ensure a higher compression ratio under the premise of no cylinder collision. According to the simulation results of Ansys Workbench, when the connecting rod travels to the bottom dead center, the secondary compression gap is 0.20591 mm, and when the connecting rod is at the top dead center, the primary compression gap is 0.13755 mm. Therefore, based on the results of the simulation compression gap and taking into account the actual assembly compression of various gaskets, the asbestos gasket with a thickness of 0.3 mm was selected to adjust the primary compression gap, and the paper gasket with a thickness of 0.2 mm was placed to achieve the adjustment. The purpose of adjusting the secondary compression gap.
In order to further verify the effectiveness of the size design, the assembled air compressor was assembled to a self-built test bench to carry out the inflation test. The test bench is composed of a test system and a lubricating oil system, and its structure is shown in Figure 11. During the test, the air compressor will charge a gas cylinder with a capacity of 8L under the condition of rotating speed $n=2000\pm 50\text{r/min}$ and inlet pressure at standard atmospheric pressure. At the same time, the test requires that the time required for the pressure in the cylinder to change from 0 to 14.7MPa should not exceed 36min.

After testing, it was found that no cylinder collision occurred during the inflation test of 10 sets of air compressors. Figure 12 shows the change of cylinder pressure with the inflation time of each test air compressor. It can be seen from the figure that although there are certain differences in the inflation efficiency between different air compressors, they can all complete the inflation of the 8L cylinder.
within 36 minutes and increase the air pressure to 14.7MPa. This shows that the size of the primary and secondary compression gap designed based on the finite element simulation results is reasonable and effective, and can provide new ideas and methods for the future air compressor assembly size design.

![Figure 12](image.png)

**Figure 12.** The results of the first and second cylinder clearance design verification test.

5. **Conclusions**

Through the theoretical force analysis of the connecting rod of the micro high pressure three-stage piston air compressor and the comprehensive finite element analysis of structural statics and steady-state thermal, the steady-state thermal finite element analysis of the first and second cylinders is performed, and the analysis results are finally analyzed. Carrying out size design and testing, the main conclusions are as follows:

1. Through the theoretical force analysis of the connecting rod, when the connecting rod is at the top dead center, the connecting rod is subjected to a tension of 276.53N, and when the connecting rod is at the bottom dead center, the connecting rod is subjected to a tension of 757.97N;

2. Through the simulation analysis of the connecting rod by Ansys Workbench software, the connecting rod will be axially elongated by 0.278663mm at the top dead point and 0.268469mm at the bottom dead point under high temperature and high speed conditions. The secondary cylinder is elongated by 0.14111mm in the axial direction as a whole, and is elongated to 0.0622711mm at the secondary compression gap;

3. According to the simulation results, ten sets of air compressors with the first-stage compression gap adjusted to 0.3mm and the second-stage compression gap adjusted to 0.2mm are carried out for inflation tests. The experimental results are qualified and the design dimensions meet actual needs.

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