Finite Element Analysis of New Crankshaft Automatic Adjustment Mechanism of Pumping Unit

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Abstract. In this paper, the crankshaft automatic adjustment mechanism designed on CYJY10-4.2-53HF pumping unit is used as the research object. The simulation of the friction and bending moment of the crank is carried out by ANSYS Workbench, and the finite element simulation results are compared with the theoretical calculation results to verify the theoretical calculation. The final result is that the finite element analysis of the friction of the crank is basically consistent with the theoretical calculation; The analysis and calculation of the stress and deformation about the two kinds of ultimate conditions of the guide platform are carried out too; The dynamic state analysis of the mechanism is carried out to obtain the vibration modes and natural frequencies of the vibration of the different parts of the counterweight under the condition of no preload force so that the frequency of the array can avoid the natural frequency, and can effectively avoid the resonance phenomenon, and for different modes we can improve the stiffness of the structure.

Keywords. Automatic Balance, Friction Analysis, Size Optimization, Modal Analysis.

1. Introduction
So far, the number of mechanical wells in China accounted for more than 90% of the total number of wells, while the beam pumping unit accounted for more than 98% of the total pumping units [1]. As the pumping unit in the upper and lower stroke to bear the different load, resulting in the load on the upper and lower strokes of the motor is not balanced, created energy waste, reduced the efficiency of the motor, shorten the life of the motor, affecting the machine, rod, pump properly work, so the pumping unit needs the automatic balancing device to ensure the smooth operation of the pumping equipment. The traditional way of adjusting the balance was cumbersome, the labor of workers was very intensive, and cannot achieve real-time adjustment [2]. In order to solve the above problems, a crank automatic adjustment mechanism is designed. The device is designed on the basis of CYJY10-4.2-53HF type pumping unit. As shown in Fig. 1, constructed the active crank block [3, 4], to meet the needs of the crank automatic balance. The device breaks the original automatic balancing mechanism and references the real-time balancing device on the crank, to reduce energy consumption, improve motor efficiency, and increase the life of the pumping unit. Based on the finite element method, the static and dynamic

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Analysis of the structure is carried out by using ANSYS Workbench to verify the reliability of the whole structure design, and the orientation platform is optimized, providing technical support for subsequent research designs.

2. Finite element analysis of balance blocks
If the material of the two solid surfaces is improperly chosen or the pressure applied between them is very large, the solid friction will cause wear; the balance block is connected with the guide platform through the dovetail groove, the upper end size of the dovetail is small, with the up and down movement of suspended load and the crank is turned back and forth, the balance block will produce bending moment, thus affecting the balance of the force and deformation. It is necessary to calculate the frictional force \( f \) between the guide plate and the counterweight and the lateral bending moment of the balance block \( M \).

There is a transition fit between the counterweight and the guide platform, and the counterweight moves at a speed of 1 mm / s on the guide platform, So theoretically the three contact surface friction force is similar to the static friction, that \( f_{\text{theory}} = N = 1500 \text{N} \).

2.1. Calculation of frictional force of balance block
The actual balance block is subjected to sliding friction, and the theoretical calculation takes the static friction force, and the finite element is used to check its desirability. Create a three-dimensional model with SolidWorks, importing it into ANSYS Workbench, the material of the balance block is HT150, and its material properties [5] are set as follows.

![Fig.1 Automatic adjustment of crank structure](image)

**Table 1.** Ht150 material properties

| Property | Mass Density / kg m\(^{-3}\) | Elastic Modulus / GPa | Tensile Strength / GPa | Poisson's Ratio |
|----------|-------------------------------|-----------------------|------------------------|----------------|
| Value    | 7340                          | 120                   | 150                    | 0.25           |

**Table 2.** 45 steel material properties

| Property | Mass Density / kg m\(^{-3}\) | Elastic Modulus / GPa | Tensile Strength / GPa | Bow to Extremes / GPa | Poisson's Ratio |
|----------|-------------------------------|-----------------------|------------------------|-----------------------|----------------|
| Value    | 7890                          | 209                   | 355                    | 600                   | 0.269          |

The material of oriented platform is 45 steel, the material properties of the settings shown in Table 2. The frictional contact is established and the coefficient of friction is set to 0.2. As can be seen from Figure 2, the establishment of the frictional connection is correct. Setting the grid size to 10mm, and then meshing as shown in Figure 3, we can see the grid quality is uniform and density is fine. Apply the constraint and load, the results shown in Figure 4.
The friction calculation values are shown in Table 3. Stress and deformation as shown in Fig. 5 and Fig. 6, it is necessary to note that the actual deformation of the frictional force is small and very slow.

**Fig.2** Friction connection results.

**Fig.3** Friction connection results.
Fig. 4 Balanced blocks apply cand loads.

Table 3. Balance block friction results

| Direction Axis | Bottom Friction / N | Side Friction / N | The Other Side Of The Friction / N |
|----------------|---------------------|-------------------|-----------------------------------|
| X              | -3971.6             | 90.283            | 94.213                            |
| Y              | 13403               | -435.25           | -446.04                           |
| Z              | 4.214               | 679.03            | -676.43                           |

Fig. 5 Friction Equivalent Stress Diagram.
Fig. 6 Friction Equivalent Deformation.

The results of the analysis, the bottom friction mainly depends on the Y axis direction, the size is 13403N, both sides of the friction mainly depends on Z-axis direction, the size is 679N and 676N, the total friction is the sum of the three, and the size is 14758N, compared with the theoretical calculation, the relative error is:

\[ e = \frac{15000 - 14758}{15000} = 1.6\% \]

The theoretical calculation can be considered correct. The error exists because the model simplification is not precise enough or the frictional contact between the counterweight and the guide platform is different from the actual conditions with the instability of the operating conditions. Under the action of friction, the maximum stress of the counterweight \( \sigma_{\text{max}} = 10.9 \text{MPa} \), HT150 tensile strength \( \sigma_b = 150 \text{MPa} \), compressive strength is generally greater than the tensile strength, take the safety factor of \( n_{\text{block}} = 2.0 \), then the allowable stress:

\[ \left[ \sigma_{\text{block}} \right] = \frac{\sigma_b}{n_{\text{block}}} = \frac{150}{2} = 75 \text{MPa} \]

The allowable stress is much larger than the maximum stress of the material, so the material strength meets the requirements; maximum deformation is 0.49mm, almost no impact on the material, so the balance block can withstand the friction load. In order to reduce the friction, in practice, you can also add a thin layer of nylon between the guide platform and the balance block.

2.2. Simulation of lateral bending moment of balance block
As the mechanical model of the two sides of the bending moment is not easy to simplify in the theoretical calculation, ANSYS Workbench finite element software is used to simulate the calculation. The results of the final solution are shown in Table 4.

It can be seen from the table that the bending moments of the sides around the X-axis are 11492N • mm and 11671N • mm, respectively, and the total bending moments are 46978N • mm and 51212N • mm, respectively.

3. Finite element analysis of oriented platform
According to the weight of the balance block and the weight of the screw, the nuts and other parts on the platform, The final weight of the guided platform is determined to be about 1.5 tons. The two most dangerous conditions are discussed: the first condition: the balance block is perpendicular to the
guide platform, above the guide platform, the angle between the guide platform and the horizontal direction is 0°; the second condition: the balance block is perpendicular to the guide platform, under the guide platform, at this point the guide platform is rotated 180°, The force of the guide platform is shown in Table 5.

3.1. Oriented platform strength check
According to the above steps, the three-dimensional model is introduced, and the material attributes [5] are setted according to Table 2. The constraints and loads are applied according to the two cases respectively. The final simulation is shown in Fig7.

### Table 4. Balance block bending results

| Direction Axis | One Side / N · mm | The Other Side / N · mm |
|----------------|-------------------|------------------------|
| X              | 11671             | -11492                 |
| Y              | -22815            | 23524                  |
| Z              | 44339             | 39006                  |

### Table 5. Guide platform load conditions

| Condition | Force/N | Area / mm² | Pressure / MPa |
|-----------|---------|------------|----------------|
| Case 1    | 15000   | 193301.18  | 0.0776         |
| Case 2    | 15000   | 86602.42   | 0.1732         |

A: Static Structural
Equivalent Stress
Type: Equivalent (von Mises) Stress
Unit: MPa
Time: 1
2017-02-11 10:45

(a). Case1: Guidance platform simulation stress diagram
(b). Case 1: Guided platform simulation strain diagram

(c). Case 1: Guided platform simulation deformation diagram
(d). Case 2: Guidance platform simulation

(e). Case 2: Guided platform simulation stress diagram
Case 2: Guided platform simulation deformation

**Fig. 7** Force and restraint of transport platform

**Table 6.** Oriented platform analysis results

| Condition | Stress /MPa | Strain       | Deformation / mm |
|-----------|-------------|--------------|------------------|
| Case 1    | 0.33        | 1.90x10⁻⁶   | 2.1x10⁻⁴         |
| CASE 2    | 1.24        | 8.22x10⁻⁶   | 1.0x10⁻³         |

As can be seen from the figure, the stress and deformation are shown in the following table. Second conditions both occurs at the upper edge of the guide platform. The material of the platform is 45 steel, the yield strength of the material is 355MPa, the safety factor is 3.0, the allowable stress is 120MPa, much less than the yield strength, so it is safe in both cases, the impact of the deformation on the platform structure is very small. So the guide platform can withstand the load.

3.2. **Oriented platform structure optimization**

From the above analysis we can see that the force and deformation of the guide platform are very small under the two conditions, so the structural size is optimized to meet the strength conditions to save the economy. Taking into account the bending moment at the tail of the crank, combined with the actual situation, the force condition of guide platform after optimized are shown below.

**Table 7.** Optimized post - guided platform analysis results

| Condition | Stress /MPa | Strain       | Deformation / mm |
|-----------|-------------|--------------|------------------|
| Case 1    | 1.26        | 8.16x10⁻⁶   | 1.5x10⁻³         |
| Case 2    | 4.65        | 3.15x10⁻³   | 5.8x10⁻³         |
The optimized maximum deformation still occurs at the upper edge of the guide platform, and its strength and deformation are also meet the requirements. Overall quality is 33791kg before optimization, after optimization is 26025kg, weight reduction of 23%, so to a certain extent, save the economic costs.

4. Structural dynamics analysis
It is known from the vibration theory that the low-order mode is mainly used in the vibration process of the structure, So only the first six natural frequency analysis of the crank self-adjusting mechanism is extracted [6,7]. Take balance block from the guide platform dovetail groove left 300m, the balance block in the middle, balance block from the guide platform dovetail groove right 300m these three cases to discuss. The natural frequency of each order is shown in Table 8, and the contrast is shown in Fig. 9[8].

CYJY10-4.2-53HF pumping unit its crank speed is 3r/min, so solid excitation frequency is 0.05Hz. It can be seen from Table 8, regardless of where the balance block is, the first-order natural frequency is far greater than the array frequency, it can avoid the occurrence of resonance; balance block at different positions ,its first order natural frequency is basically same, so the position of the balance block has little effect on its first order natural frequency. The symmetrical position of the counter block its vibration modes are also symmetrical. The contrast pattern is shown in Fig.8, The overall shape of the assembly is not only twisting, bending but also rocking vibration, which will affect the service life of the whole body, so in the process of designing, we should consider the impact of vibration on the body, the greater impacted parts need to increase its stiffness to reduce the impact of vibration on the overall body.

Table 8. The first six order frequencies corresponding to the different positions of the balance block

| Frequency | First order /Hz | Second order /Hz | Third order /Hz | Fourth order /Hz | Fifth order /Hz | Sixth order /Hz |
|-----------|-----------------|-----------------|----------------|-----------------|----------------|----------------|
| Left      | 189.42          | 277.81          | 279.1          | 336.79          | 521.53         | 550.06         |
| Middle    | 188.96          | 334.33          | 514.44         | 516.33          | 521.73         | 568.05         |
| Right     | 188.8           | 285.17          | 294.33         | 334.88          | 474.05         | 521.61         |

a. First order mode
b. Second order mode

c. Third order modes
d. Fourth order mode

e. Fifth order mode
5. Summary
1) Based on the designer of the crank automatic adjustment mechanism, the paper uses ANSYS Workbench to analyze and calculate the friction and lateral bending moment of the balance block. The simulated friction force of the balance block is basically the same as the theoretical value, which verifies the desirability of the theoretical calculation, and simulates the lateral bending moments, which provides a theoretical reference for the subsequent optimization and research.

2) The stress analysis and optimization of the guide platform are carried out. It is concluded that the stress and deformation meet the strength requirements before and after the optimization of the guide platform. After optimization, the weight is reduced by 23% and the economic cost is saved.

3) The dynamic analysis of the structure shows that the different positions of the equilibrium block have little effect on the natural frequency of the structure. The vibration direction of the mechanism under low-order vibration mode is obtained, and the weak area of the organization is found, the design can be used for this area to increase the stiffness of the parts.

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