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Finite Element Analysis and Calculation of HTC15J01 Petrochemical Industry Crane Pulley Design

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The pulley is a small part of complex machine. Its lifetime and quality directly affect the reliability and safety of the entire equipment. Firstly, this paper introduces the design requirements for the main pulley of the oil rig HTC15J01 for analysing and calculating the stress and load distribution of the main pulley. Then, based on the design calculation in terms of stability, strength, and extruding strength etc. of the pulley, the key structural sizes of the main pulley were obtained. Finally, through checking calculations and finite element calculations in terms of stability, strength, and extruding strength etc., the analysis shows that the structure size parameters of main pulley meet the requirements of the oil rig. The calculation results of the HTC15J01 rig crane pulley provide a theoretical basis for the application of the pulley in the project.

1. Introduction

The manufacturing methods of the pulley mainly include casting, welding, stamping, rolling, and riveting etc. Casting is the most typical method for pulley, and it generally uses cast iron or cast steel with simple process. However, the pulley requires continuous operation, the frequency of load changes is relatively high, and the failure modes are mainly cracking, excessive wear on the rope groove, and decreased rigidity, etc., therefore, for the cast pulley product, the replacement must be made for the whole pulley, thereby causing high product cost (Zhao, 2010).

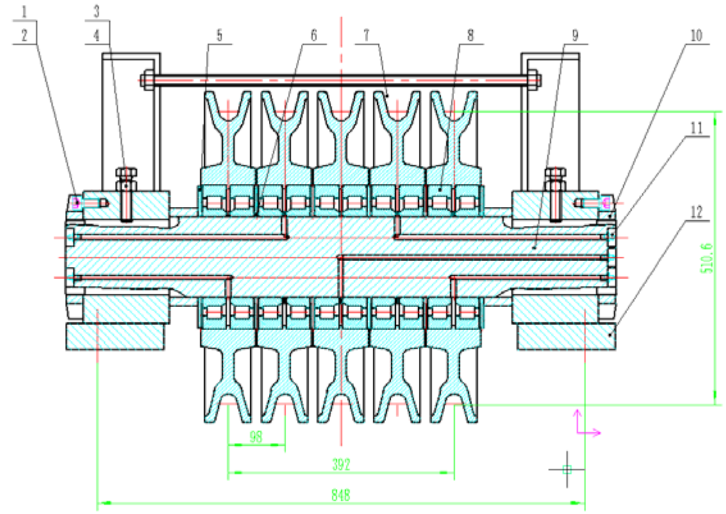
Relatively speaking, the hot-rolled pulley mainly relying on plastic forming has become the trend of development, because the hot-rolling reduces the deformation of the welding method and the connection looseness of the riveting. Besides, the pulley failure is mainly caused by the abrasion or cracking under repeated alternating load acting on the rope groove, while the design of the wheel web can basically meet the stability requirements at small failure probability. During the hot rolling process, the metal material structure in the rope groove portion changes, improving the anti-wear and anti-pressure capability. In addition, for the split-type pulley, different materials can be used for the rims, webs, and hubs, which are convenient to process at lower manufacturing cost. Therefore, the low-cost, long-life, high-stability pulley design method has become an essential concept for designers (Li, 2017).

By taking the main pulley of oil drilling rig HTC15J01 as object of study, this paper discusses the design and verification of the pulley in terms of the force analysis, design verification, stability analysis and verification of the pulley. This shall provide reference for the engineering design of the pulley (Jiang et al., 2018).

2. HTC15J01 crane pulley force and checking

2.1 Main parameters of the crane pulley

The maximum hook load of the crane is 950KN, and the total weight of the traveling system and the wire rope is 90KN. The pulley 5×6 wheel was selected for the traveling crane and crane (Figure 1 and 2). The actual maximum tension of the wire rope is F=(950+90)/12=86.7KN.



1.Screw 2. Washer 3. Bolt 4. Nut 5. Spacer 6. Inner spacer 7. Main pulley 8. Bearing 9. Shaft 10. Gland 11. Straight hydraulic grease nipple 12. Bearing seat

Figure 1: Structure of pulley assembly

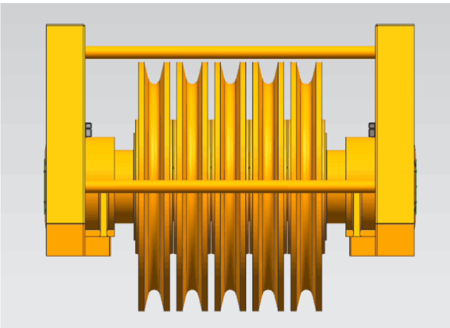


Figure 2: 3D solid model of pulley assembly

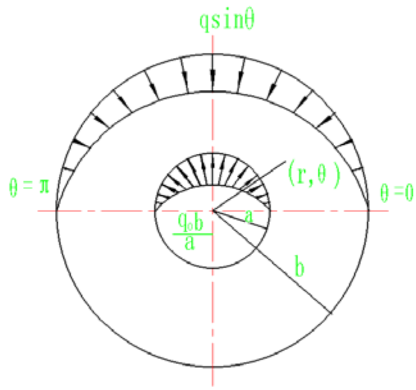


Figure 3: Load distribution of pulley

2.2 Pulley force analysis

Let F be the force of block line on the pulley, and β be the angle between the tensions (Figure 3). The total pressure P on the pulley shaft is given as:

$$P = \sqrt{F^2 + F^2 - 2F^2\cos(180^\circ - \beta)} = 2F\sin\frac{\alpha}{2} \tag{1}$$

where:

$$\frac{\alpha}{2} = 90^\circ - \frac{\beta}{2}.$$

At α=180°, the tension at both ends of the rope was parallel and P reached a maximum value $P_{rmax}=2F$. Therefore, in the mechanical model, α=180° was taken as the stress state, ignoring the influencing factors such as the line deflection. The mechanical model is in Figure 1. The force of the rope on the pulley was distributed in sine curve $q\theta=q_0 sin\theta$, and the component of $q\theta$ in the vertical direction was $q\theta sin=q^2\theta$; then the resultant force is given as:

$$2F = 2\int_0^{\frac{\pi}{2}} q_0 \sin^2\theta d\theta \tag{2}$$

where: $q_0 = \frac{F}{\int_0^{\frac{\pi}{2}} sin^2\theta d\theta} = \frac{F}{\frac{\pi}{4}\times b} = \frac{4F}{\pi b}$, $q\theta$ is the distribution load per unit length, q_0 is the distributed load strength when θ is equal to 90°, and b is the outer radius of the pulley web.
The inner load and the outer load of the pulley are in balanced state, and the lower ring is not affected by the load. The tangential equal force and equal effective force moment are all zero.

2.3 Pulley strength design and extrusion checking conditions

$$\sigma_{\max} = \frac{4F}{\pi ah} < [\sigma]/n \tag{3}$$

where: [σ] is the basic allowable stress for the pressure, a is the inner radius of the pulley web, n is the safety factor of the manufacturing process. In the design rules for cranes, it's recommended that the allowable declination angle of the wire rope is not more than 5 degrees, and its axial component is sin5°=0.0872. This component can cause the web to bend, but in the limited way, so it can be properly considered when selecting the allowable stress. For the maximum stress, the extruding stress of inner bore will be redistributed when the steel reaches its yield limit, thus, no danger will occur.
Considering that the wheel grooves of pulley is subjected to repeated extrusion stress of rope, it's usually checked by the extrusion stress, which must satisfy the formula:

$$\sigma = \frac{F}{\mathrm{bd}} \le [\sigma] \tag{4}$$

where: d is the diameter of wire rope, [σ] is the material extrusion stress, and b is the outer radius of the pulley web

2.4 Stability design of the pulley

For the stability design of the pulley, the Ritz method of minimum energy principle was applied to calculate the critical load formula for the sinusoidal distribution pressure of the circular plate (Qian and shen, 1989):

$$q_{0cr} = \frac{9(b-a)}{16b}\left[\frac{\pi Db^2 A}{(b-a)^4} + \frac{\pi EI}{2b^3} + \frac{2\pi GJ_k}{b(b-a)^2}\right] \tag{5}$$

where: D is the flexural rigidity of pulley web, h is the thickness of pulley web, GJ_K the torsional rigidity on both sides of pulley rope groove cross section, I the moment of inertia on both sides of rope groove cross section, A the coefficients related to Poisson's ratio of pulley web material and a/b, a is the inner radius of the pulley web, and b the outer radius of the pulley web.
Regardless of the energy for rope groove bending and twisting, the critical value is given as:

$$q_{0cr} = \frac{9\pi DbA}{16(b-a)^3} \tag{6}$$

Considering the energy of rope groove bending and twisting, the critical value will increase by two:

Bending:

$$q_{0cr} = \frac{9\pi EI(b-a)}{32b^4} \tag{7}$$

Twisting:

$$q_{0cr} = \frac{9\pi GJ_k}{8b^2(b-a)} \tag{8}$$

When the pulley is under the action of wire rope tension $F = \frac{\pi b q_0}{4}$, the full energy formula can be used for stability calculation. If the rope tension on the pulley is known, the thickness of the pully web can be designed from the stability formula. Substituting $D = \frac{Eh^3}{12(1-\mu)^2}$ into formula (5), it's calculated as:

$$h=\sqrt[3]{\frac{12(1-\mu^2)(b-a)^4}{EbA}\left[\begin{array}{l}\frac{64F}{9\pi(b-a)}\\-\frac{EI}{2b^3}-\frac{2GJ_k}{(b-a)^2b}\end{array}\right]} \tag{9}$$

For the convenience of engineering calculation and partial safety design, (6) can be adopted to obtain:

$$h=\sqrt[3]{\frac{256(1-\mu^2)(b-a)^3F}{3\pi^2EbA}} \tag{10}$$

Taking into account the stability factor n, it's simplified as:

$$h=\sqrt[3]{\frac{\xi bFn}{A^*}} \tag{11}$$

where: $A^*=\frac{A}{(1-\frac{a}{b})^3}$, $\xi=\frac{256(1-\mu^2)}{3\pi^2E}=3.7466\times10^{-5}$, μ=0.3, E=2.1×10^5 N/mm^2.

3. Pully design

3.1 Wire rope and pulley groove design

According to API Spec 8C-2012 version 5, the total depth G of the rope groove should be 1.33d at the minimum, and the maximum should be 1.75*d*, and *d* is the nominal diameter of the wire rope. According to the item 4.7 in API Spec 8C-2012 Version 5, the design safety factor of the pulley is not less than 3, the material of the pulley is Q235, and the basic allowable stress for compression is 113*MPa*.

$$R_{\min}=R_{rope}\times1.06 \tag{12}$$

$$R_{\max}=R_{rope}\times1.1 \tag{13}$$

where: R_{min} is the minimum radius of the new pulley groove, R_{max} is the maximum radius of the new pulley groove, and R_{rope} is the nominal radius of the wire rope.
According to the maximum hook load and 5×6 wheel train of the crane, the wire rope at radius 13*mm* was selected with reference to API8C. Based on formula (12) and (13), the maximum and minimum pulley radii are 14.3mm and 13.78mm, respectively. Considering that the crane pulley will be lifted and lowered frequently as the rig crane system during work. To avoid the wear of the wire rope, the radius of the groove was selected to be 13.95mm.
3.2 Design for outer diameter and thickness of pulley web
According to the rule 9B in American API Standard, the minimum diameter of the pulley groove has a certain relationship with the wire rope diameter as follows (Song et al., 2003; Jiang et al., 2003):

$$D\geq e\times d \tag{14}$$

where: *d* is the diameter of the wire rope, and *e* is the outer diameter coefficient of the pulley web.
For the rope diameter, it's shown *e*=21 in the *Table 5-Sheave-diameter factors API-9B*.
$D\geq e\times d$=21×26=546*mm*; in view of the wears of the wire rope and the pulley, *D*=550*mm*, and the outer diameter of the pulley is taken as $D_0=D+2(1.33d\sim1.75d)=620\sim641$, where, D_0=630*mm*. According to the load on the pulley bearing, double row cylindrical roller bearing LM294710D is selected, the basic static load C_0 is 1650*KN*, the bearing outer diameter 347*mm*, and inner diameter 245*mm*.

$$P_0=\frac{c_0}{s_0}=\frac{1650}{2}=825>2\times F=550KN$$

Where: C_0 is the basic static load rating; S_0 is safety factor for static strength; P_0 is the equivalent static load.
It meets the requirements.

In line with the maximum load of the traveling crane and the bearing carrying capacity of travelling system, it's taken *a*=107.8*mm* (radius of the inner radius of the pulley web), and the safety factor for the pulley manufacture is 1.5. Based on formula (3), the pulley strength is designed as:

$$h = \frac{4Fn}{\pi a[\sigma]} = \frac{4\times 86700\times 1.5}{3.14\times 113\times 107.8} = 13.6mm$$

In order to prevent the internal defects of the web material from affecting the material properties and ensure the stability of the pulley, it's taken *h*=15*mm*.

4. Results and analysis

4.1 Maximum tension and compression strength checking

The maximum tension of the wire rope must meet the requirements of (3), and $F \leq \frac{\pi h a[\sigma]}{4n} = [F]$:

$$[F] = \frac{\pi \times 20\times 107.8\times 113}{4\times 1.5} = 127.5KN$$

According to the requirements of the oil rig equipment and the US API standard, wire rope selection should be:

$$\phi 26 - 1960\left(6\times 19s + \text{IWRC-IPS}\right),$$

With the minimum breaking force of 512.7*KN*, and the minimum safety factor of 3, it's given as:

$$F = \frac{512.7}{3} = 170.9KN > 86.7KN,$$

So, F<[F] and the design meets the requirements of the web design.

Based on the extrusion strength formula (3), the allowable stress of the reference bond extrusion is [σ]=30MPa due to the dynamic extrusion of the pulley and the rope, and then:

$$\sigma = \frac{F}{\text{bd}} = \frac{86700}{275\times 26} = 12.1MPa < [\sigma] = 30MPa$$

It meets the design requirements.

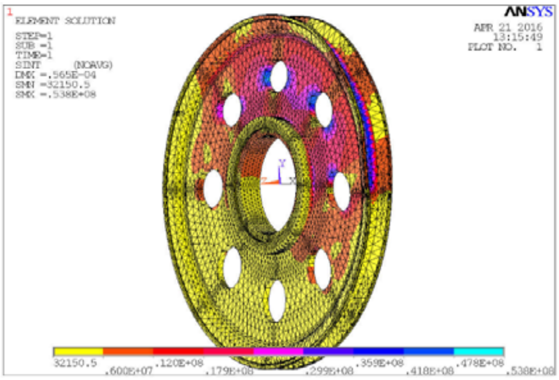
4.2 Stability analysis

Based on the formula of stability:
Regardless of the bending energy of the rope groove, the critical load is:

q_{0cr}=38952.42,$[F]_{cr} = \frac{\pi b q_{0cr}}{4} = 1329.15KN$.

Considering the rope groove bending energy, the critical load is: q_{0cr}=51520.31, $[F]_{cr}$=1112.19*KN*.
Considering the full energy of the strain: q_{0cr}=53100.14, $[F]_{cr}$=1146.3*KN*.

In order to further understand the stability of the pulley, ANSYS software was used to establish the three-dimensional model of the pulley based on the parameters of the designed pulley. This model was loaded according to the load distribution to make strength calculation and stability analysis. Figure 4 and 5 show the stress and displacement obtained through ANSYS finite element loading. It's found that the maximum stress of the pulley was 53.8MP, which occurred on the web, the design stress of the web was 120*MPa*, and the maximum deformation was 0.0000565mm, all meeting the design requirements. It indicates that the finite element analysis results have certain reliability.



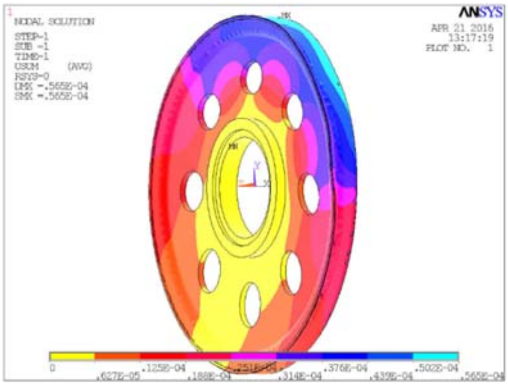


Figure 4: Stresst analysis of the pulley

Figure 5: Displacement analysis of the pulley

5. Conclusions

This paper analyses and calculates the stress and load distribution of the main pulley of the oil rig HTC15J01 crane during the working process. Then, based on the pulley requirements for stability, strength, and extruding strength etc., the key structural sizes of the main pulley were obtained. Finally, through the checking calculation and finite element analysis in terms of stability, strength and compression strength, it is found that the designed main pulley of the crane meets the requirements for use.

Acknowledgement

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Reference

API SPEC 8C, (2012): “Specification for Drilling and Production Hoisting Equipment” (PSL1 and PSL2).

API SPEC 9B, (2012): “Specification for Drilling and Production Hoisting Equipment” (PSL1 and PSL2).

Jiang P., Kong Y., Tang S., 2018, Nitrogen injection pyrolysis simulated experiment and pyrolysis characteristics analysis of huadian oil shale, Chemical Engineering Transactions, 66, 421-426 DOI:10.3303/CET1866071

Jiang Z.Y., Tieu A.K., Zhang X.M., 2003, Finite element simulation of cold rolling of thin strip, Journal of Material Processing Technology 2003, 140, 542-547, DOI; 10.1016/s0924-0136(03)00832-x

LI F.Z., 2017. Finite element analysis and calculation ofTC315-PSL2 oil drilling rig crane pulley design, Manufacturing Automation, 39(3), 93-96, DOI: 10.3969/j.issn.1009-0134.2017.03.025

Qian Z.Y., Shen J.Z., 1989, Design and calculation of the strength and stability of the pulley radial plate, Journal of Shanghai Maritime Institute, 1(3), 17-24.

Song Y.H., Zhang K.F., Wang Z.R., 2003, 3D FEM analysis of the temperature field and thermal stress for plastic thermal forming, Journal of Materials Processing Technology 2003, 97, 25-43, DOI: 10.1016/S0924-0136(99)00097-7

Zhao N.T., Cao J.G., Zhang J., 2008, Work roll thermal contour prediction model of nonoriented electrical steel sheets in hot strip mills. Journal of University of Science and Technology, 15, 352-358, DOI: 10.1016/S1005-8850(08)60066-3