

Optimization and Analysis of Gear Set Structure for Continuous Loading and Unloading Hydraulic Power Tongs

Xiang Zhang^{1,*}, You Yang¹, Pengfei Zhang¹

¹ Mechanical Engineering College, Xi'an Shiyou University, Xi'an 710065, China

* Corresponding author: Xiang Zhang (Email: cuoguofeng@163.com)

Abstract: Optimizes and analyzes the gear set structure of hydraulic power pliers used in continuous loading and unloading operations. The optimization design process of gear set structure was elaborated in detail, including the selection of independent variables, determination of constraint conditions, and establishment of optimization objectives and fitness functions. The simulated annealing particle swarm optimization algorithm (SAPSO) was used for optimization design, and the optimization results showed that the total volume of the gear set was reduced by 17.1%. Finally, the strength of the optimized gear set was verified through finite element analysis, and the results showed that the strength of the optimized gear set met the actual working requirements.

Keywords: Hydraulic power plier, SAPSO, gear, optimized, strength calibration.

1. Introduction

Hydraulic power tongs are mainly used for upward unbuckling operation of tubular columns in well workover operations. The optimized design of hydraulic power tongs can effectively improve the efficiency of well workover operations, reduce production costs and increase the service life of power tongs [1-4].

Scholars have conducted research on the optimal design of gearing mechanism in mechanical systems. Lv Y et al. [5] optimized the gear material and surface heat treatment process, and discussed the methods of surface hardening, nitriding, and coating treatment of gears, aiming to improve the wear resistance of gearing system. Peruñ G et al. [6] provided theoretical guidance on reducing gear vibration by designing a dynamic model of a power-cycling gearing test rig, and comparing experimental results with computer simulation results. Habermehl C et al. [7] provided a new modeling method for gearing efficiency, which can optimize gearing vibration reduction. computer simulation results to provide theoretical guidance for reducing gear vibration. Habermehl C et al. [8] provided a new method for modeling gearing efficiency, which can optimize the loss behavior of gear trains under variable speed conditions.

From the above it can be seen that the optimization of

scholars with the gear transmission mechanism is mainly optimized from the aspects of energy consumption, wear and structure, and due to the special characteristics of the hydraulic power tongs open gears, there are fewer studies on its related optimization. The author through the use of SAPSO algorithm on the team designed by the hydraulic power clamp main clamp gear transmission mechanism, the total volume of the minimum as the optimization objective for the optimization design, for the open gear transmission mechanism optimization design provides a new way of thinking.

2. Optimized Design of Hydraulic Power Tongs Main Tong Drive Gear Set Structure

The transmission gear set structure used in this optimization design is the author's team independently designed for well dressing machine tubing column continuous on the buckling hydraulic power clamp main clamp gear transmission mechanism.

2.1. Original design parameters

The main design parameters of the hydraulic power tongs are shown in Table 1

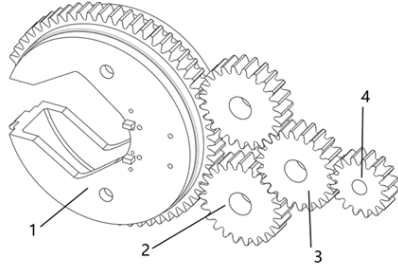
Table 1. Hydraulic Power Clamp Main Design Parameters

Applicable pipe diameter/mm	Large gear speed/rpm	gear ratio	Output torque/ kN m
60-140	14-60	2.8	20

Hydraulic power tongs in the process of unloading buckle according to different working conditions will be selected low-speed high-torque or high-speed low-torque mode to work. In the low-speed high-torque mode, the gear drive system is subjected to the largest torque, and it is necessary to carry out simulation calculations on the gear structure in this

mode to complete the strength check of the gear drive system.

After the gear drive installation data is known, the gear drive system is modeled in three dimensions by SolidWorks software, and the specific structure is shown in Fig.1 .



1-open gear; 2-idler set; 3-transmission wheel; 4-driving wheel.

Figure 1. Gear train

2.2. Selection of independent variables

In order to reduce the complexity of computation and solution, we only focus on the parameters that have a significant effect on the optimization objective to reduce the number of variables in the optimization problem. That is: the opening width b of the open gear is selected according to the selected diameter of the pipe column, so this parameter is not considered; the tooth width coefficient of the open gear; ϕ_d is set as an independent variable, and the tooth width of the pinion gear should be slightly larger than that of the large gear (5mm in this paper) in order to satisfy the conditions of gear mesh and transmission; the other parameters of the gears are all selected in accordance with the standard gears. Finally, the hydraulic power clamp main clamp transmission gear set structure optimization independent variable selection results as shown in equation (1).

$$X = [m, Z_1, Z_2, Z_3, \phi_d] = [x_1, x_2, x_3, x_4, x_5] \quad (1)$$

2.3. Determination of constraints

2.3.1. Gear module

The open gear needs to provide large torque in the process of punching buckle, so it is necessary to select a larger gear modulus to meet the actual practical needs, the other gear modulus are selected according to the open gear modulus, the final gear modulus value range as shown in equation (2).

$$9 \leq x_1 \leq 20 \quad (2)$$

and x_1 takes values chosen according to the standard modulus.

2.3.2. Number of teeth

The number of teeth of each gear must be an integer, and the transmission process can not occur root cut phenomenon, according to the relevant working conditions of the number of teeth selection conditions are as follows:

$$x_2 \geq 18 \quad 20 \leq x_3 \leq 30 \quad 60 \leq x_4 \leq 70 \quad (3)$$

A 5% error before and after the transmission ratio is allowed, i.e.:

$$|x_4 / x_2 - 2.8| \leq 0.05 \quad (4)$$

2.3.3. Tooth width

Gear tooth width coefficient ϕ_d is selected in the range of 0.2-2.4, i.e.:

$$0.2 \leq x_5 \leq 2.4 \quad (5)$$

2.3.4. Tooth contact strength conditions

In order to ensure the normal operation of the gears involved in the work, involved in contact with the gears to meet the fatigue strength of the meshing tooth surface and the root of the tooth bending fatigue strength, that is, the gears of their respective contact strength is less than the permissible contact strength. Due to the mutual meshing of the gear contact stress size is the same, in order to reduce the number of constraint boundaries only need to calculate which the number of teeth less gear contact strength can be.

The tooth root bending fatigue strength calibration formula is shown in equation (6).

$$\frac{2KT_1}{bd_1m} Y_{Fa} Y_{Sa} Y_{\epsilon} \leq \sigma_{FP} \quad (6)$$

In above: Y_{Fa} is the toothing coefficient, Y_{Sa} is the stress correction factor, Y_{ϵ} is the overlap coefficient, and σ_{FP} is the permissible bending stress.

The calibration formula for tooth contact fatigue strength is shown in equation (12).

$$\sigma_H = Z_E Z_H Z_{\epsilon} \sqrt{\frac{2KT_1}{bd_1^2} \frac{\mu+1}{\mu}} \leq [\sigma_H] \quad (7)$$

In above: Z_E is the elasticity coefficient; Z_H is the area coefficient; Z_{ϵ} is the overlap coefficient; K is the load coefficient; T_1 is the pinion torque; μ is the ratio of the number of teeth of the contacting large gear to the number of teeth of the pinion.

2.3.5. Optimization objective and fitness function

The size of the open gear set is mainly determined by the module and tooth width of each gear, so the minimum volume of the four gears is used as the optimization design objective function. The optimized mathematical model of gear transmission system established by these parameters is shown in equation (12).

$$f(x) = \frac{1}{4} \pi x_1^2 x_2^2 (x_5 x_1 x_4 + 10) + \frac{1}{2} \pi x_1^2 x_3^2 (x_5 x_1 x_4 + 5) + \frac{1}{4} \pi x_1^3 x_4^3 x_5 \quad (8)$$

The smaller the function $f(x)$ the smaller the total volume of the four gears, the optimization objective in this case is to solve for the minimum value of the function under the above constraints.

2.4. Simulated Annealing Particle Swarm Algorithm (SAPSO)

The SAPSO algorithm incorporates the ideas of particle swarm optimization algorithm and simulated annealing algorithm. It does this by randomly generating new candidate solutions x_i' around the current solution x_i and evaluating the fitness of these new solutions to the current solution. Then, the decision to accept or not accept these new solutions is based on the Metropolis^[9]. This criterion allows the admission

of non-optimal solutions with some probability^[10]. By introducing the Metropolis rule from the simulated annealing algorithm into the PSO algorithm, it helps to prevent the algorithm from achieving only local optimality, which in turn enhances its ability to find the optimal solution on a global scale.

2.5. Algorithm implementation

The parameters of SAPSO optimization algorithm are: 5 independent variables, particle swarm size is 50, in order to improve the optimization results the number of iterations will be set to 500, the learning factors are all set to 2.05, and the annealing constant $\lambda = 0.5$. The particle swarm optimization algorithm is mainly used to solve the optimization problems without constraints. However, the gear structure optimization studied in this paper has multiple constraints, which need to be ignored first and then the penalty conditions are introduced. This is done by adding a penalty term to the objective function in the code, so that if the resulting solution fails to satisfy all the constraints, the penalty condition will be triggered, leading to a significant increase in the value of the objective function, which will make the solution obsolete.

Since the optimized data is not integer, it needs to be rounded up and the processed results are compared with the pre-optimization parameters as shown in Table 2 :

Table 2. SAPSO algorithm optimized results

Design parameters	m	Z_1	Z_2	Z_3	ϕ_d	Objective function value
Original parameter	9	23	24	64	0.3	6.421e+7
Optimized parameters	9	22	24	63	0.26	5.324e+7
Optimization effect						17.1%

From the Table 2 it can be seen that the value of the objective function is reduced by 17.4% after optimization, and next the parameters after optimization are verified by finite element analysis to be in accordance with the design requirements.

3. Strength Calibration of Optimized Gears

Hydraulic power tongs in the upper unloading work on the tube column to produce a certain torque or force, in order to ensure that the hydraulic power tongs clamp head and transmission mechanism in the repair operation of the upper unloading buckle stability and worker safety, I use three-dimensional solid modeling software SolidWorks on the hydraulic power tongs of the main clamp for solid modeling, the use of Workbench on the main clamp in a few of the main structural components such as clamping device, gear drive system, etc. under static load stress analysis. Workbench is used to analyze the force under static load on several major structural components of the main clamp, such as the clamping device and the gear transmission system.

3.1. Strength check of master clamp clamping support plate

The main clamp clamping device is involved in the whole

process when unbuckling on the hydraulic power clamp. Among them, the support plate in the clamping device is subjected to the most severe force when the crumbling or tightening is carried out. In order to make the solution process more accurate, the clamping device and the open gear are assembled as a whole for hydrostatic analysis. Cylindrical support is applied at the position of the rotating shaft and pin of the support plate, fixed constraint is applied at the curved tooth plate, torque is applied at the open gear, and thrust of the piston rod is applied at the contact area between the support plate and the piston rod of the hydraulic cylinder. The final analysis results are shown in Fig.2.

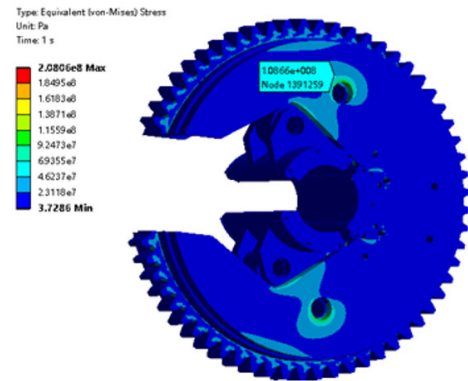


Figure 2. Stress diagram of the main jaw cotter gear and the clamping device

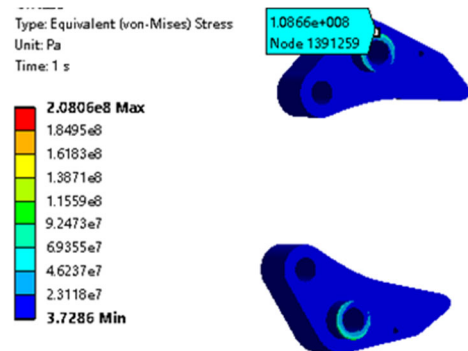


Figure 3. Clamping Device Support Plate Stress Diagrams

From Fig.2 and Fig.3 the maximum stress on the main jaw opening gear during chipping or tightening is 208 MPa and the maximum stress on the support plate is 108.66 MPa.

Since the selected gear and support plate materials are plastic materials, the allowable stresses are:

$$[\sigma] = \frac{\sigma_s}{n_s} \quad (9)$$

In above: σ_s -yield limit of material; $n(s)$ -factor of safety.

The materials and yield limits for each part are shown in the following Table3 :

Table 3. Material parameters related to main parts

	Name of material	Safety factor	Permissible stress
Pinion (gear wheel)	20CrNiMo	1.8	433
Support plate	structural steel	1.4	225

From the above table, it can be seen that the main clamp open gear and its clamping device support plate in the hydraulic power clamp maximum torque conditions are subjected to the stress is less than the permissible stress of its material, so it meets the strength requirements.

3.2. Strength check of gearing system

Cylindrical Support is applied at each gear hole and the tangent direction is set to free during simulation analysis. Then the corresponding loads are added to the gear train, the torque added to the low speed gear is $2\text{kN}\cdot\text{m}$, and the torque added to the open gear is $35\text{kN}\cdot\text{m}$. It is known that there are two meshing states in the double idler gear train: the open gear 1 meshes with two idler pulleys 2, and the open gear meshes with one idler pulley 2. These two states of engagement correspond to different stresses on the components of the drive train. Figures 4 and 5 show the stress analysis results of the driveline in two idler gears and one idler gear respectively.

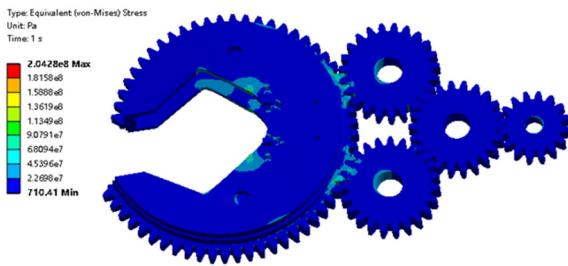


Figure 4. Stress diagram for double idler engagement case

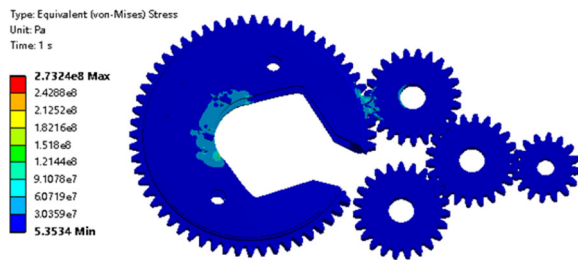


Figure 5. Stress diagram for single idler engagement case

When the open gear receives the maximum torque, Fig.4 shows the equivalent force diagram for the case of double idlers meshing with the open gear, at this time, the maximum stress value of the gear transmission system is 204.28MPa ; Fig.5 shows the equivalent force diagram for the case of single idler meshing with the open gear, at this time, the maximum stress value of the gear transmission system is 273.24MPa . The design material of the gear is 20CrNiMo , the permissible stress of the steel is 433MPa . The permissible stress of the steel is 433MPa , which can be obtained that the stress on the gears in the gear transmission system is much less than the permissible stress of the material, so the structural design of the gear transmission system is reasonable, and the optimized design of the gear's relevant parameters

meets the requirements of actual use.

4. Conclusion

This article selects the hydraulic power tong transmission mechanism for the continuous up and down coupling of the workover rig column independently developed by the team, and optimizes it based on the goal of minimizing the overall volume. The main conclusions are as follows:

- 1) Based on the SPASO algorithm, optimize the design of the open gear transmission group, and compare the final results with those before optimization to obtain the conclusion that the volume after optimization is reduced by 17.1% .
- 2) The strength of the optimized gear set was verified through finite element simulation, and the conclusion was drawn that the strength of the optimized transmission gear set meets the actual working requirements.

References

- [1] Artoni A. A methodology for simulation-based, multiobjective gear design optimization[J]. Mechanism and Machine Theory, 2019, 133: 95-111.
- [2] Tavčar J, Černe B, Duhovnik J, et al. A multicriteria function for polymer gear design optimization[J]. Journal of Computational Design and Engineering, 2021, 8(2): 581-599.
- [3] Qin Z, Wu Y T, Lyu S K. A review of recent advances in design optimization of gearbox[J]. International Journal of Precision Engineering and Manufacturing, 2018, 19: 1753-1762.
- [4] Atila Ü, Dörterler M, Durgut R, et al. A comprehensive investigation into the performance of optimization methods in spur gear design[J]. Engineering Optimization, 2020, 52(6): 1052-1067.
- [5] Lv Y, Lei L, Sun L, et al. Improvement of the wear resistance of 20CrMnTi steel gear by discrete laser surface melting[J]. Optics & Laser Technology, 2023, 165: 109598.
- [6] Peruñ G, Łazarz B. Modelling of power transmission systems for design optimization and diagnostics of gear in operational conditions[J]. Solid State Phenomena, 2014, 210: 108-114.
- [7] Habermehl C, Jacobs G, Neumann S. A modeling method for gear transmission efficiency in transient operating conditions[J]. Mechanism and Machine Theory, 2020, 153: 103996.
- [8] Liu Yongping, Ren Zhongtao, Wei Yongqiao, et al. Research on the Error of Involute Variable Tooth Thickness Straight Gear Transmission. Machine Tool and Hydraulics, 2023, 51(04): 1-6.
- [9] Pei Junfeng, Song Yupeng, Liu Zhigang, et al. Simulation Analysis of the Gear Transmission System of a Tubing Tong. Journal of Changzhou University (Natural Science Edition), 2018, 30(02): 63-68.
- [10] Fang Kai, Wang Huiliang, Su Jianxin. Geometric Backlash Analysis and Experimental Research of RV Reducer. Mechanical Design and Manufacturing, 2024-09-13, 1-6.