Transmission mechanism analysis of downhole drive reciprocating pump production system based on rack and pinion

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Abstract. At present, the rod production system is difficult to meet the requirements of directional wells and cluster wells with large slant, the rod and pipe of these pumping wells suffer from serious wear, high energy consumption and low system efficiency. Downhole drive reciprocating pump production system, cancelling the sucker rod for transferring forces and motion, utilizing downhole submersible motor drives the reciprocating pump through the downhole drive mechanism, which can avoid the wear of the rod and pipe and improve the system’s efficiency. The composition and working principle of downhole drive reciprocating pump production system are introduced. The scheme of downhole drive reciprocating pump production system based on rack and pinion drive is analyzed in detail. The dynamic model of rack and pinion is established. By the dynamic simulation of the downhole transmission mechanism, the vibration loads of the mechanism with start up different accelerations are analyzed, and it is concluded that it is better to start with the variable acceleration. The scheme can reduce the vibration load caused by the motion inertia to the mechanism and improve the service life of the whole oil production system.

1. Introduction
Currently, the artificial lifting method in China's oil field is still mainly beam pumping unit. However, for directional wells and cluster wells with large inclination, the phenomenon of pole and pipe eccentric wear is serious, the energy consumption is large, and the system’s efficiency is low[1][2]. The sucker rod of rod production system(mainly beam pumping unit) restricts the output of oil wells and the pump’s efficiency to a certain extent, and the phenomenon of eccentric wear of the rod and pipe is serious[3]. In order to solve the phenomenon of rod and tube eccentric wear, artificial lift with non-rod pump has become a new research direction[4]. Non-rod pump fundamentally avoids the restriction for oil extraction machinery due to the existence of sucker rod string, so that mechanical production can be more widely used in oilfield production.

In 2005, Daqing Oilfield Co., Ltd developed a downhole drive lifting device, the plunger pump is driven by a rotating motor for artificial lift[5]. In 2008, China University of Petroleum(East China) and Sinopec Shengli Oilfield Co., Ltd jointly developed a new type of rodless pumping system, this system is driven by a linear motor, which is connected to the oil pump[6]. In 2016, Wang Lei designed and invented a linear motor downhole drive oil production device, the linear motor is set at the center of
the oil extraction device, the plunger pump is located at both ends of the linear motor, the upper oil pipe is set on one side of the oil extraction device, the bottom end of the upper oil pipe is connected with the sucker rod, and the lower oil pipe is set at the bottom of the sucker rod[7].

The main difference between downhole drive reciprocating pump production system and rod pump production system proposed in this paper is that rod transfer force and movement are not required, thus completely avoiding eccentric wear of rod and pipe and improving system efficiency. Under different driving characteristics, the output response changes of drive mechanism of downhole reciprocating pump oil extraction system were studied, and the vibration effects of inertia of moving parts of oil extraction system on the mechanism under different driving modes were analyzed.

2. Composition and working principle of downhole drive reciprocating pump production system

2.1. System composition and principle

Downhole drive reciprocating pump pumping system is mainly realized in the downhole drive reciprocating pump by rotating motor to achieve the purpose of artificial lift, and the rack and pinion drive are applied to achieve the transformation of movement form. The system is divided into ground equipment and downhole unit, ground equipment includes ground frequency control cabinet, transformer, and power generating facilities and power grids etc; Downhole units include: submersible motor, reducer, changement, protector and reciprocating pump etc. Figure 1 shows the schematic diagram of downhole drive reciprocating pump production system. The working principle of the system is that the ground equipment transmits power through the submersible cable 6 to the submersible motor 11 installed with motor protector underground, the submersible motor transmits power to the reducer 10, after decelerating by the reducer, the power will be transferred to the changement 9, the output end of the changement is connected with a rack and pinion mechanism 8, a balance member is installed in the symmetrical position of the rack, which is connected with the piston rod of reciprocating pump through connecting pieces to complete the transmission of force and motion of the whole system.

1-wellhead, 2-ground frequency control cabinet, 3-transformer, 4-power generating facilities and power grids, 5-oil pipe, 6- oil-submersible cable, 7-reciprocating pump, 8-rack and pinion mechanism(includes balance mechanism), 9-changement, 10-reducer, 11-submersible motor(includes protector), 12-casing

Figure 1. Downhole drive reciprocating pump production system composition diagram.

2.2. Technical parameters of downhole drive reciprocating pump production system

Technical parameters of reciprocating pump and operating parameters:

- Stroke: 0.8 m; Jig frequency: 2min⁻¹; Pump diameter: 32mm; Maximum downpump depth: 1200 m;
- Submergence depth: 200 m; Fluid density: 0.95 t/m³; Maximum liquid column load: 23 KN.
- Overall dimensions of downhole drive mechanism: 72mm*9000mm.
3. Downhole drive mechanism scheme of reciprocating pump production system

As shown in Figure 2, the downhole drive drive mechanism scheme of downhole reciprocating pump production system mainly includes: submersible motor, reducer, changement, rack and pinion mechanism and reciprocating pump etc. The use of submersible motor drives reciprocating pump to complete artificial lift, with emphasis on the transmission system in the rack and pinion mechanism.

![Figure 2: Downhole drive mechanism scheme of reciprocating pump for oil production.](image)

Figure 2. Downhole drive mechanism scheme of reciprocating pump for oil production.

Rack and pinion mechanism schematic diagram, as shown in Figure 3, the rack in the pinion and rack mechanism is directly connected with the piston rod of reciprocating pump, because the unilateral bearing of rack may cause load imbalance when system work normally, a balance strut is installed at the position of the rack relative to the gear shaft to share the load on the rack, the balance strut and the rack are connected with the piston rod of reciprocating pump through a connecting piece.

![Figure 3: Rack and pinion mechanism schematic diagram.](image)

Figure 3. Rack and pinion mechanism schematic diagram.

The load on the rack can be approximately regarded as the liquid column load on the full plunger at the moving liquid depth:

$$W'_L = \rho_L g L_f A_p$$  \hspace{1cm} (1)

In the formula: $W'_L$—Liquid column load on full plunger at moving depth, kN; $\rho_L$—Density of liquid, t/m³; $g$—gravity acceleration, (=9.81 m/s²); $L_f$—Moving level depth, m; $A_p$—Piston cross-sectional area, m².

4. Dynamic analysis of transmission system

4.1. Rack and pinion dynamics model

For the pinion and rack mechanism, it can be regarded as an elastic system, and the simplified dynamic model is shown in Figure 4.

![Figure 4: Simplified diagram of rack and pinion mechanism model](image)

Figure 4. Simplified diagram of rack and pinion mechanism model.
In the figure: \( I_p \) - moment of inertia of gear, kg/m²; \( I_g \) - moment of inertia of rack, kg/m²; 
\( m_p \) - pinion weight, kg; \( m_g \) - rack weight, kg; \( \omega_p \) - rotation rate of gear, r/min; 
\( \omega_g \) - rotation rate of rack winding the speed instantaneous center, r/min; \( e \) - error of gear tooth; 
\( \alpha \) - rack and pinion engagement angle, °; \( k_m \) - mesh composite stiffness; 
\( c_m \) - meshing comprehensive damping; \( R_p \) - gear base circle radius, mm.

The meshing force of gear and rack \( F_n \) is mainly composed of elastic meshing force \( F_k \) and viscous meshing force \( F_c \):

\[
F_n = k_m \delta(t) + c_m \dot{\delta}(t) \quad (2)
\]

The relative displacement of the gear and rack along the engagement line is calculated by formula (3), formulas (2) and (3) be united can obtain formula (4),

\[
\delta(t) = \omega_p R_p - \omega_g R_g - e(t) \quad (3)
\]

\[
F_n = k_m \left( \omega_p R_p - \omega_g R_g - e(t) \right) + c_m \left( \omega_p R_p - \omega_g R_g - e(t) \right) \dot{\delta}(t) \quad (4)
\]

Comprehensive mesh stiffness \( k_m \) can be obtained by formula (5):

\[
k_m = \frac{1}{\delta_p + \delta_g} \quad (5)
\]

In the formula: \( \delta_p \) - single tooth deformation of gear, mm; \( \delta_g \) - single tooth deformation of rack, mm.

Gear mesh damping \( c_m \) can be calculated by formula (6):

\[
c_m = 2 \xi_g \frac{k_m R_p^2 R_g^2 I_p I_g}{R_p^2 I_p + R_g^2 I_g} \quad (6)
\]

In the formula: \( R_g \) - Rotation radius of rack, m; \( \xi_g \) - Damping ratio of tooth mesh.

4.2. Dynamics simulation

ADAMS software is used to simulate the gear rack and bevel gear mechanism, and different driving functions are applied to it to observe the rack output velocity, acceleration, and displacement curve. Firstly, the drive function started by uniform acceleration be applied to it, and the drive function is:

\[
\text{if}(\text{time} > 3: 0.94 \times \text{time}, 2.81, \text{if}(\text{time} > 12: 2.81, 2.81, \text{if}(\text{time} > 18: 14.09 - 0.94 \times \text{time}, -2.81, \text{if}(\text{time} > 27: -2.81, -2.81, \text{if}(\text{time} > 30: 28.19 + 0.94 \times \text{time}, 0, 0)))))
\]

Figure 5 shows the simulated rack output acceleration curve driven by uniform acceleration, and the acceleration has obvious amplitude change at the time nodes of 0s, 3s, 12s, 18s and 27s, this is because the elastic deformation of the gear transmission system and the instantaneous acceleration caused by inertia torsion is too large, but its influence on the speed is not very big. However, the excessive acceleration caused by the inertia torsion will produce a large vibration impact, which will cause great damage to the gear transmission system and shorten its service life.

In order to improve the vibration impact created by the inertial load caused by the inertial acceleration on the system, different accelerations are applied to the system to start, and different accelerations are used to buffer the inertial load. The driving function of variable acceleration starting is applied to the input shaft as follows:

\[
\text{if}(\text{time} > 3: -0.94 \times \text{time}, -2.81, \text{if}(\text{time} > 12: -2.81, 2.81, \text{if}(\text{time} > 13: 2.81 + 1.405 \times \text{time}, -2.81, \text{if}(\text{time} > 27: 2.81, 2.81, \text{if}(\text{time} > 30: 2.81 - 0.94 \times \text{time}, 0, 0))))))
\]
Figure 6 shows the rack acceleration curve under the starting of variable acceleration drive function. In the curve, the graph is neat and there is no obvious instantaneous amplitude increase, this indicates that under the variable acceleration starting, the inertial acceleration of the gear tooth system can be reduced to some extent, and the buffering effect can be played. Reducing the vibration shock caused by the inertial acceleration of the gear tooth system plays a certain role in improving the system life. Of course, this kind of situation occurs in the simulation. In the actual operation, certain deviations may occur due to various reasons and the interference of various operating conditions. The specific situation depends on the actual operating conditions.

![Figure 5. Rack acceleration output curve driven by uniform acceleration](image1)

![Figure 6. Rack acceleration output curve driven by variable acceleration](image2)

4.3. Kinematic analysis

According to the parameters in 1.2 and formula (1), the load-bearing load can be calculated. Downhole drive reciprocating pump production system is powered by submersible motor, its frequency can be adjusted by ground frequency control cabinet. During normal operation, the submersible motor rotates in uniform speed. Meanwhile, the time of upper stroke and lower stroke is equal, and the time of acceleration and deceleration is also equal[8]. The oil submersible motor is connected with the protector, and the power is transferred to the changement through the reducer, and then connected with the intermediate transmission mechanism to drive the reciprocating pump to complete the oil extraction work, its movement law is similar to that of the submersible motor.

Take a certain point at the end of piston rod of reciprocating pump as the research object, according to the reciprocating pump's stroke of 0.8m and jig frequency of 2, the system cycle time $T=30s$ can be calculated. Take the acceleration time of the submersible motor is $t_0$, the acceleration is $a$, and the
maximum speed of the submersible motor is $n_{\text{max}}=3000\text{r/min}$, then, the speed and acceleration of the submersible motor are respectively:

$$n(t)=\begin{cases} 
  a t, & 0 \leq t \leq t_1 \\
  a, & t_1 \leq t \leq t_2 \\
  a(t_0 - t + t_2), & t_2 \leq t \leq t_4 \\
  -a, & t_4 \leq t \leq t_5 \\
  -a(t_0 - t + t_5), & t_5 \leq t \leq t_6
\end{cases}$$  \hspace{1cm} (7)

$$a(t)=\begin{cases} 
  a, & 0 \leq t \leq t_1 \\
  0, & t_1 \leq t \leq t_2 \\
  -a, & t_2 \leq t \leq t_4 \\
  0, & t_4 \leq t \leq t_5 \\
  a, & t_5 \leq t \leq t_6
\end{cases}$$  \hspace{1cm} (8)

In the rack and pinion, the relationship between the speed of the gear and the motor speed is as follows:

$$\omega = \frac{2\pi n}{60i_1i_2i_3}$$  \hspace{1cm} (9)

In the formula: $n$ is the speed of the motor, $\text{r/min}$, and $i_1, i_2, i_3$ is respectively the transmission ratio of planetary gear reducer, changement and bevel gear.

For the convenience of the following analysis, let,

$$i = i_1i_2i_3$$  \hspace{1cm} (10)

The piston rod of reciprocating pump is directly connected with the rack, and its power comes from motor through a series of deceleration and the transformation of motion model[9]. Thus, the velocity of the piston rod is shown in formula (11):

$$v = \frac{2\pi n}{i_1i_2i_3}$$  \hspace{1cm} (11)

Therefore, the displacement, velocity and acceleration of the suspension center is:

$$s(t)=\begin{cases} 
  -\frac{\pi R}{i}, & 0 \leq t \leq t_1 \\
  -\frac{\pi Ra}{i} - \frac{2\pi Ra}{i}(t-t_1), & t_1 \leq t \leq t_2 \\
  -\frac{\pi Ra}{i} - \frac{2\pi Ra}{i}(t-t_1) - \frac{2\pi Ra}{i}(t-t_2) + \frac{1}{2} \frac{2\pi Ra}{i}(t-t_2)^2, & t_2 \leq t \leq t_4 \\
  -\frac{\pi Ra}{i} - \frac{2\pi Ra}{i}(t-t_1) - \frac{2\pi Ra}{i}(t-t_4), & t_4 \leq t \leq t_5 \\
  -\frac{\pi Ra}{i} - \frac{2\pi Ra}{i}(t-t_1) - \frac{2\pi Ra}{i}(t-t_5) + \frac{\pi Ra}{i}(t-t_5)^2, & t_5 \leq t \leq t_6
\end{cases}$$  \hspace{1cm} (12)

$$v(t)=\begin{cases} 
  -2\pi Ra/i, & 0 \leq t \leq t_1 \\
  -2\pi Ra/i, & t_1 \leq t \leq t_2 \\
  -2\pi Ra(1-t+t_2), & t_2 \leq t \leq t_4 \\
  2\pi Ra/i, & t_4 \leq t \leq t_5 \\
  2\pi Ra(1-t+t_5), & t_5 \leq t \leq t_6
\end{cases}$$  \hspace{1cm} (13)
In one of the strokes, $0 \sim t_1$ is acceleration stage, $t_1 \sim t_2$ is uniform speed stage, $t_2 \sim t_3$ is deceleration stage, $t_3 \sim t_4$ is reverse acceleration stage, $t_4 \sim t_5$ is reverse uniform speed stage and $t_5 \sim t_6$ is reverse deceleration stage. When the system works normally, in the process of upper stroke, the piston rod end first does uniform acceleration movement, then does linear movement, and finally does uniform deceleration movement. At this time, the upper stroke is completed; Then the reversing device changes the direction of motion and enters the downstroke stage. In this process, first do uniform acceleration to the maximum speed, then uniform speed for a period of time, and then do uniform deceleration until the speed is zero, complete a stroke.

5. Conclusion

The following conclusions are drawn after analyzing the drive mechanism of downhole drive reciprocating pump production system based on pinion and rack:

1) Downhole drive reciprocating pump production system, in the downhole drive reciprocating pump with submersible motor to achieve artificial lift, because the elimination of transfer force and movement of the sucker rod, can avoid the rod and tube eccentric wear and improve system efficiency.

2) The transmission mechanism scheme of downhole drive reciprocating pump production based on rack and pinion mechanism provides a new method for artificial lift of rodless pump.

3) If the submersible motor starts at uniform acceleration, it will produce a large shock vibration load to the system due to the inertia load; If the submersible motor is started with variable acceleration, it can cushion the vibration shock caused by inertia load and inertia torque of gear teeth, and improve the service life of the whole oil production system.

Acknowledgement

This paper is supported by “Shaanxi Province Key Discipline Of Mechanical and Electronic Engineering” and “Petroleum drill and exploitation equipment” engineering research center, Shaanxi province. Thanks for the support of colleagues from “Petroleum Machinery Research Institute”, Xi’an Shiyou University.

References

[1] Wei Q W, Zhang M, Yang B, Zhang B J. The Problem Existing in Linear Motor Mining Pumps and Its Improvement Measure [J] 2007 Chin. J. Oil Field Equipment 36(6):10-13

[2] Bai L P, Ma W Z, Yang Y, et al. Discussion on Energy Saving of Motor for Beam pumping Unit [J] 1999 Chin. J. China Petroleum Machinery 27(3):41-44

[3] Chai X J, Liu Y P, Li H S, et al. Theory Research Intodouble Stepless Pa-rameter Modulation Top Drive In-Phase Well Pumping Unit [J] 2005 Chin. J. Oil Drilling & Production Technology 27:50-53

[4] Huang X D, Yao M C, Lei D R, et al. The Development and Field Test of the Power-driven Electric Submersible Reciprocating Pump [J] 2018 Chin. J. Drilling & Production Technology 41(2):82-84

[5] Wang L. Downhole driving oil extraction device of linear motor 2017 (Chinese patent: ZL201621276786.0.)

[6] Zhao L, Yang X Y, Qu Z Q, Zhou Q, Wang J R, Liu J M. Design of Lifting System of a New Rodless Downhole Pumping System [J] 2008 Chin. J. Oil Field Equipment 37(2):37-39

[7] Zhang Q, Hu Q H, Zhang M L, Zhou Z H, Xie L F. Research Progress of pump driven by
Downhole Linear Motor [J] 2015 Chin. J. Science & Technology Vision (10):46+69
[8] Zhang C. 2008 Dynamics of Machinery (Beijing: Higher Education Press) pp 1-4
[9] Li R F, Wang J J. 1997 Gear system dynamics (Beijing: Science Publishing House)