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Research on Energy-saving Technology of Crank Balanced Pumping Unit

Feng Ziming, Zhang Jindong, Gu Huibin and Luan Yunwei
School of Mechanical Science and Engineering, Northeast Petroleum University, Daqing, Heilongjiang, 163318, China

Abstract: In order to saving energy and emission reduction, a secondary balance technology was used to reform the conventional beam pumping unit. The output torque from the reduction gearbox of conventional beam pumping units is usually characterized by its periodic drastic changes. Based on the idea of "cutting peak and filling valley" and the theory of the Fourier series expansion for torque curve, the second balance device is designed in order to slow down the fluctuations in the torque curve. The second balance device, with the similar balance principle to the crank balance, is connected to the output shaft of the reduction gearbox and can further reduce torque fluctuation rate and peak torque. Field test result of the secondary balance device shows that the Root-Mean-Square (RMS) torque is decreased by 22.7% and the energy-saving rate of motor reaches 6.54%.

Keywords: Beam pumping unit, fourier series, net torque of gearbox, secondary balance

INTRODUCTION

Characterized by their simple structures, reliable and durable performances, beam pumping units have been leading the mechanical oil-production equipment although its disadvantages include low efficiency and high energy consumption. According to the related statistics, power consumption makes up about one-third of the total costs in oilfield production. Power consumed by beam pumping unit accounts for nearly 80% of the total. Low motor working efficiency resulted by severe fluctuations of net gearbox torque and over-high installed power often lead to high energy consumption. Therefore, efforts at reducing net gearbox torque fluctuations can play an important role in energy-saving and consumption-reducing in oilfield production. In order to reduce the costs and increase the benefits in oilfield development, all kinds of energy-saving pumping units have been developed in major oilfields. Many design plans (Guo *et al*., 2002; Firu and Che, 2003; Takacs, 1997; Gibbs, 1993; Li *et al*., 1995; Sun 2003; Chen *et al*., 2004) are provided about improving energy-saving of beam pumping units. These plans are mainly related to changing the structure to reduce the torque fluctuation and the peak torque, thus reducing motor rated power and increasing equipment life (Zhu *et al*., 2005). Such as the partial composite balance and barbell composite balance pumping unit all have good energy-saving effect, but its reconstruction and cost are high (Ma *et al*., 2006).

In order to saving energy and emission reduction, it is necessary to design a device to decrease the fluctuations of torque and energy consumption. Based on the principle of Fourier series to establish the precise balance principle of pumping units, this study provides the design plan for the secondary balance of pumping units including the design of secondary institutions and the programs for calculating the precise balance of the pumping units, as well as the optimized design of the related parameters of the secondary balance.

Result of the field tests shows that the secondary balancing device can effectively reduce the fluctuations of net gearbox torque and the working efficiency of the motor is improved. The RMS torque reduction rate reaches 23.8%, while motor efficiency is increased by 13% on average. At the same time, data from the field tests is basically consistent with the theoretical calculation. This shows that a wide application prospect is waiting for this secondary balance device. Three wells are equipped with the secondary balancing devices in No. 1 Oil Production Plant, Daqing oilfield. The secondary balance parameters are calculated according to the principle of Fourier series and field tests are conducted.

BALANCE THEORY OF PUMPING UNITS

Figure 1 shows the four-bar mechanism of pumping units. In crank rocker mechanism of beam pumping units, crank is the driving part while the suspension center is the driven part in upstroke. In downstroke, suspension center is the driving part while crank is the driven part. Even with reverse flow of energy, both active and passive conditions exist in the

Corresponding Author: Feng Ziming, School of Mechanical Science and Engineering, Northeast Petroleum University, Daqing, Heilongjiang, 163318, China

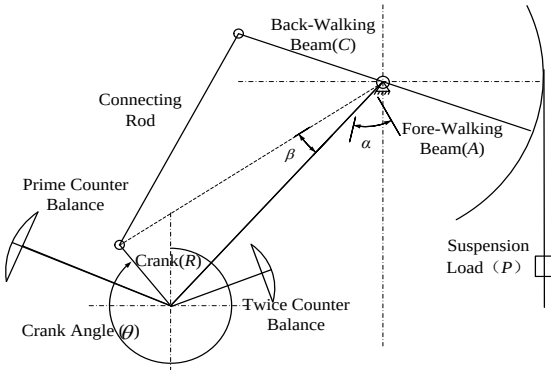

Fig. 1: Four-bar linkage of a pumping unit

crank-rocker mechanism. The reverse flow can lead to mechanical loss. Considering the reverse flow of energy, net gearbox torque is calculated by formula (1).

$$T_c(\theta) = \frac{AR\sin\alpha}{C\sin\beta}(P - B)\eta_b^k \tag{1}$$

$T_c(\theta)$ is a periodic function that its independent variable is θ and its period is 2π. It satisfy Dirichlet condition;

- T_c (θ) is continuous in the interval [0, 2π] and no discontinuous point
- The extreme point number of $T_c(\theta)$ is limited in the interval [0, 2π]

Therefore function $T_c(\theta)$ can be transformed into Fourier series:

$$T_c(\theta) = a_0 + \sum_{k=1}^{\infty}(a_k \cos k\theta + b_k \sin k\theta) \quad (k = 1, 2, 3\cdots) \tag{2}$$

where, α_0, α_k, b_k are all Fourier coefficient, its calculation formula are as follows:

$$a_0 = \frac{1}{2\pi}\int_0^{2\pi} T_c(\theta)\, d\theta \tag{3}$$

$$a_k = \frac{1}{\pi}\int_0^{2\pi} T_c(\theta)\cos(k\theta)\, d\theta \quad (k = 1, 2, 3\cdots) \tag{4}$$

$$b_k = \frac{1}{\pi}\int_0^{2\pi} T_c(\theta)\sin(k\theta)\, d\theta \quad (k = 1, 2, 3\cdots) \tag{5}$$

The results of Triangle transformation of formula (1) as follows:

Table 1: Fourier coefficient and torque characteristic value

	k = 1	k = 2	k = 3	k = 4
A_k	1.95	-5.55	-4.17	-2.72
B_k	40.76	8.56	2.31	-0.74
C_k (kN/m)	40.81	10.20	4.77	2.82
T_k (°)	2.74	-32.94	-60.96	-74.75
Average torque (kN/m)	8.66	8.67	8.68	8.69
RMS torque (kN/m)	12.08	9.70	9.10	8.93
Cycle load coefficient	1.39	1.12	1.05	1.03
Peak Torque (kN/m)	29.22	22.21	18.53	15.83
Minimum torque (kN/m)	-4.18	0.80	4.04	5.56
Degree of balance (%)	60.60	57.47	71.87	78.63
Motor Power (kW)	8.91	7.16	6.71	6.59

$$T_c(\theta) = c_0 + \sum_{k=1}^{\infty} c_k \sin(k\theta + \tau_k) \quad (k = 1, 2, 3\cdots) \tag{6}$$

That is; $c_0 = \alpha_0$; $c_k = \sqrt{a_k^2 + b_k^2}$; $\tau_k = \arctan\left(\frac{a_k}{b_k}\right)$

Since the integrand in formula (3), (4), (5) are not elementary function; they are represented by data forms in the actual calculation. Therefore it is difficult to calculate the exact integral value of the original function. In addition, the polished rod load P mutation appears in the upper and lower dead points, so we have to use the numerical methods to solve the above function. So long as the numerical integrator precision is controlled within a proper range, it can meet the requirements of engineering applications. Therefore, Simpson numerical integration formula is often employed to calculate all above Fourier coefficients.

With the exchange of Fourier series and trigonometric function of Polish rod torque, we get the trigonometric function shown by Eq. (6). The coefficient ck and τk in Table 1 are substituted into the formula (6), it is the Fourier series deformation expression of polish rod torque. The phase angle τkin formula (6) is generated in order to eliminate the cosine terms of series expression (2); its physical function is to eliminate the cosine component of the polish rod torque. With sinusoidal components balanced manner, a phase angle of τk (a generalized offset angle) exists. Only when τk is considered, the sine and cosine torque components can be balanced out.

This shows the energy-saving principle of beam pumping units: adjust the structural parameters of pumping units to make the sine and cosine components of every torque harmonic change within a proper range, use appropriate phase angle and counterbalance weight to eliminate all the sine and cosine components of polish rod torque so that the peak torque and RMS torque of the crank shaft can be reduced and energy-saving can be realized.

Torque spectrum analysis indicates that the polish rod torque can be developed into series form madding of constant term, fundamental harmonic and harmonic component. Their physical senses are as follows:

- Constant term c_1 is useful work completed by pumping units when elevating oil. It represents the algebraic sum of area surrounded by the torque curve and horizontal axis (θ axis) in the torque diagram of a pumping unit. Theoretically, c_1 is the net gearbox torque value by ideal balance, which is the researchers' target in their optimized designs of pumping units and energy-saving? Fundamental harmonic is the balanced torque curve of pumping units, with its maximum balancing torque of amplitude $c_1$and the phase angle of τ_1. This is a general theoretical basis of pumping unit balance calculation.
- The second harmonic is net gearbox torque after a pumping unit is balanced for the second time with two stages of counterbalance weight. Its amplitude c_2 is the maximum balancing torque of the second balance, while τ_2 is the balancing phase angle. It provides this study with an oretical basis for the optimization and modification of the second balance of pumping units.
- K-order harmonic is net gearbox torque after pumping unit is k-th balanced with k stages of counterbalance weight, with the maximum balancing torque of k-th balance of amplitude c_k, while the balancing phase angle of k-th balance of τ_k.

When the pumping unit is multi-balanced with counterbalance, the net gearbox torque tends to a straight line and the pumping unit is working in an ideal balanced condition. We can get the real motor power consumption with the following formula:

$$N_k = \frac{T_{e,k} n}{9.549\eta} \tag{7}$$

CASE CALCULATION

Based on the above formulas, calculation is conducted on CYJ10-3-37HB pumping unit from Daqing General Machinery Plant. Table 2 shows the well conditions parameters from the calculation.

Figure 2 shows a multi-balance design based on calculated results with formula (6). Polish rod torque is an unbalance gearbox torque. The primary balance torque is a net gearbox torque. The primary balance torque and secondary counterbalance torque is the secondary balance torque. So the ideal line torque is the results of multi-balance to polish rod torque.

Table 1 shows various counterbalance weights, phase angles, net gearbox torques, specific values of torque and magnitude of motor power after counterbalance is conducted for four times. When we put the coefficients of c_k and τ_k from Table 1 into

Table 2: Working parameters of the pumping units well system

Pumping unit type	CYJ10-3-37HB
Depth of plunger (m)	1000
Pump diameter (mm)	57
Water ratio (%)	91
Stroke (m)	3
Frequency of stroke (min^{-1})	6
Working fluid level (m)	600
Oil tube diameter (mm)	76
22 mm rod length (m)	400
19 mm rod length (m)	600

Table 3: Parameters of the test torque and motor active power

	Prime balance	Secondary balance	Change (%)
Active power (kW)	4.20	3.84	8.57↓
Power factor	0.319	0.355	11.29↑
System efficient	26.01%	36.03%	10.02↑
Motor efficient	53%	66%	13.00↑
Peak torque (kN/m)	19.55	18.01	7.88↓
RMS torque (kN/m)	9.68	7.38	23.76↓

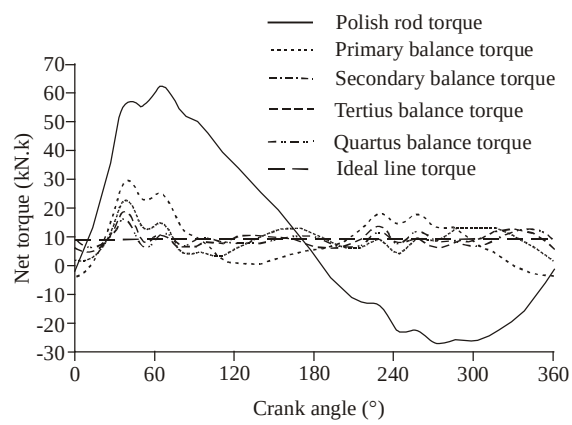

Fig. 2: Multi-balance to polish rod torque

the formula (6), we can get the Fourier series expansion deformation of pumping unit working torque:

$$T_c(\theta) = c_0 + c_1 \sin(\theta + \tau_1) + c_2 \sin(2\theta + \tau_2) + c_3 \sin(3\theta + \tau_3) + c_4 \sin(4\theta + \tau_4) \cdots$$

That is:

$$T_c(\theta) = 8.69 + 40.81 \sin(\theta + 2.74) + 10.2 \sin(2\theta - 32.94) + 4.77 \sin(3\theta - 60.96) + 1.06 \sin(4\theta - 72.39) \cdots$$

Based on Fourier series convergence, we know from Fig. 2 and Table 1:

- The net gearbox torque is converged to a line torque by multi-balance (the line torque was 8.69 kN/m in this case). After primary balance, many torqueses showing the balance conditions indicate that the fluctuation ratio of net gearbox torque has much room to be reduced. Therefore, a secondary balance device for better balance effects of the

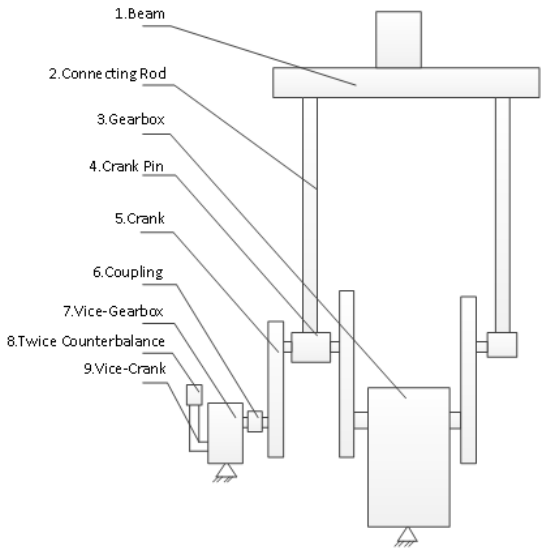

Fig. 3: Position and structure of secondary balance device

pumping units is shown in the following part of this study.

- When secondary balance completed, motor power consumption is reduced by 19.64%. The rated power of motor can be changed from 11kW to 7.5 kW. So secondary balance device is an effective approach of energy-saving.

DESIGN AND APPLICATION OF SECONDARY BALANCE DEVICE

Secondary balance device is designed based on crank balance theory of pumping unit. Figure 3 shows the design.

Figure 3 shows that without any change of the pumping unit structure, crank of the secondary balance device only after simple modification is connected to crank (5); crank (5) is connected with gearbox (7) with coupling (6); the secondary counterbalance (8) is placed in the secondary crank (9). The primary crank of gearbox (3) revolves one cycle while the secondary crank of vice-gearbox revolves two cycles. In this way, secondary balance is realized. Figure 4 shows the application of a secondary balance device on a conventional beam pumping unit.

The pumping unit for test is Model CYJ10-3-37HB, with its rated motor power 18.5 kW, frequency of stroke 5.5 min^{-1}, stroke 3 m, pump diameter 38mm, depth of plunger 853.15 m, working fluid level 560 m, submergence depth 293.09 m, primary counterbalance torque 65 kN/m, phase angle 0°, the secondary balancing torque 15.4 kN/m and phase angle 40°. The test includes two parts: test on output power parameters of the pumping unit and test on the same parameters when the pumping unit is equipped with a secondary balance device. Figure 5 to 7 and Table 3 show the

Fig. 4: Secondary balance device

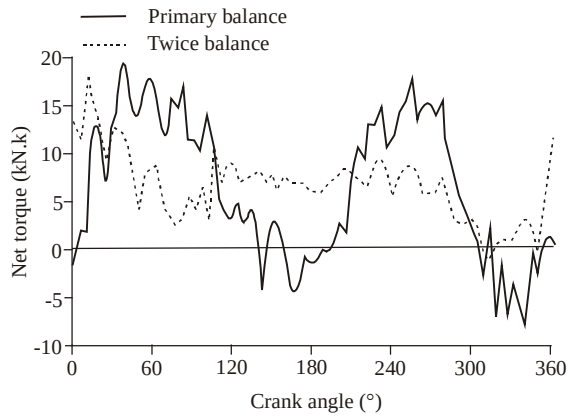

Fig. 5: Comparison between the two net torque curves of prime balance and of secondary balance

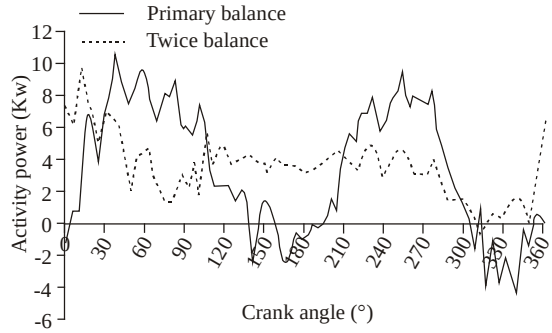

Fig. 6: Comparison between the two active power curves of prime balance and of secondary balance

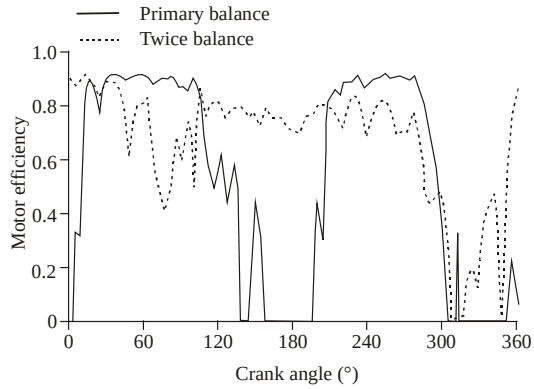

Fig. 7: Comparison between the two motor efficiency curves of prime balance and of secondary balance

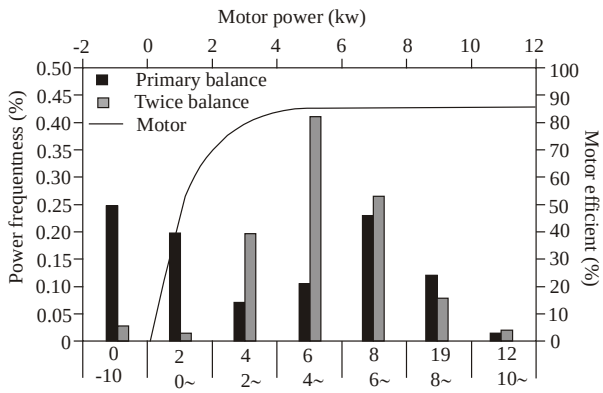

Fig. 8: Energy-saving mechanism

comparison between the old and new test data and research results.

From Fig. 5 to 7 and Table 3 we can see that with secondary balance device, RMS torque, Maximum torque, active power are significantly reduced, at the time the motor efficient increased. The theoretical curve and the test curve are basically consistent.

DISCUSSION OF ENERGY-SAVING MECHANISM

The purpose of placing the secondary balance device is to realize the "cutting peak and filling valley" effect on the periodic gearbox output torque and make the torque curve as gentle as possible so that negative torque can be eliminated. Meanwhile, by reducing torque fluctuation the cyclic loading impact on gearbox can be reduced and gearbox service life can be lengthened. Test and analysis on motor output power show that equipped with the secondary balance device, the motor works very efficiently in the entire cycle, so the motor works smoothly. This increases the motor efficiency and service life.

The orange curve in Fig. 8 represents the efficiency curve of the test motor (11 kW); the red histograms represents frequency distribution of the test input power data in primary balancing condition, while the blue histograms represents frequency distribution of the test input power data in secondary balancing condition. The motor efficiency curve in the study indicates that when the motor works in negative power range, its efficiency is zero. When the motor works in positive power range, its efficiency is rapidly increased with the power increase. When motor works in 0~2kW power range, the slope of motor efficiency curve is larger. That is, the transient efficiency is particularly sensitive to power change; little changes of power will lead to significant change in efficiency. Working in 3~11kW power range, the slope of motor efficient curve shows a gentle trend. That is to say the transient efficiency is not sensitive to power change. We often think that if the load rate of motor is less than 25%, the motor is working with low efficiency. To place the secondary balancing device is to keep the motor from working in the low efficient range.

Comparison between the two histograms of the motor active power frequency distribution with and without secondary balance shows that without secondary balance, the motor works in low efficient condition for about 45% of its working time. While equipped with secondary balance, the motor works in low efficient condition for only 4% of the total working hours. This tells us that the secondary balancing device makes the motor work efficiently in most of cycle period, which improves the efficiency and service time of the motor.

RESULTS

- According to expansion results of the polished rod torque Fourier, primary and secondary balances are conducted on pumping units. This can obviously reduce the net torque oscillation of gearbox and effectively eliminate the most exceeded-torque and unbalance-torque phenomenon which generally exist in pumping units. This shows that the Fourier coefficient balance method can be appropriately applied to pumping unit system.
- The peak and the RMS values of torque can be considered as the main factors in selecting motors. The secondary balancing device can greatly reduce the peak and RMS torque values. Therefore it can improve the working condition and service life of gearbox and motor. Besides, the secondary balance can also improve the working conditions of motors.
- Results from field tests show that with secondary balancing device, the RMS value of net gearbox torque decreased by 21.92% and fluctuations in the torque curve is significantly reduced. It proves that balance theory of pumping units put forward in this study is correct.

NOMENCLATURE

T_c = Polished rod torque, N/m
P = Polished rod load, N
B = Unbalancing weight of pumping unit structure, t
A = Walking beam forearm, m
C = Distance between center of gravity of walking beam to fulcra of walking beam, m
R = Crank length, m
η_b = Mechanical drive efficiency form crank to polished rod, (0.85~0.91)
k = Coefficient. During upstroke, k = -1; during down stroke, k = 1
α = Angle between connecting rod and crank, Deg
β = Angle between walking beam aft-arm and connecting rod, Deg
θ = Crank angle, Deg
N_k = Motor power, kW

$T_{e,k}$ = After n times balance, the RMS net torque value of reducer gearbox, kN/m
N = Frequency of stroke, min^{-1}
η = Transmission efficiency from reducer gearbox to motor

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