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Review article

Review of variable speed drive technology in beam pumping units for energy-saving

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ABSTRACT

Beam pumping units have been widely used in oilfields worldwide due to its simple structure, strong field adaptability, and convenient maintenance. Different energy-saving technologies have also been broadly applied in various beam pumping units. Among these energy-saving methods, the variable speed drive is one of the most acceptable techniques in the oil and gas industry. In this paper, the energy-saving technology of variable speed drives is discussed in detail for beam pumping units pointing out existing difficulties and current research status in kinematics. Three application examples of a variable speed drive in Daqing oilfield, the largest oilfield in China, is shown.

Contents

1. Introduction

Beam pumping units have been applied in oilfields for more than 150 years because of its simple structure, durability, and convenient maintenance (Beckwith, 2014). At present, there are over 920,000 oil wells worldwide where oil wells using beam pumping units contribute over 21% of the total oil production (Takacs, 2015). In China, there are nearly 200,000 oil wells and more than 80% of them are using beam pumping units. In some Chinese oilfields, over 30% of the total electric energy is consumed by beam pumping operations (Xing and Dong, 2015a).

Due to the structural characteristics and the crank balance mode of beam pumping units, its gearbox must be subjected to

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severe alternating shock load. Also, this load is relatively large when the pumping-system is started from a stationary state. To start the pumping unit smoothly, only high-power drive motor and/or large-capacity transformer can be used. Although the high-power motor or the large-capacity transformer solves the starting issues, the average load rate of the motor is low, which makes the power coefficient of the motor minor.

Variable speed drive is one of the fast-developing energy-saving technologies for pumping units in recent years. Generally, its optimization goal is to reduce the polished rod peak load, decrease the alternating impact load, decrease the cyclic loading coefficients, and cut down the installed power of the prime motor. At present, this technique has been widely applied in Canada, America, and China. However, there are some remaining issues in the application process of variable speed drives, such as over-torque, over-load, and low pump efficiency. The reason is that after variable-speed driving, the moment of inertia of each moving part of the pumping unit is no longer a negligible influence factor. The traditional crank balance adjustment standard is not suitable for variable-speed drive operation. The suspension speed and the suspension acceleration are inversely related. Therefore, the impact of the rod vibration load and the inertial load on the suspension load is opposite. It is difficult to judge the influence of the variable speed curve on the suspension load. The pumping unit is not always in the best operating state under the existing variable speed curve.

First, this paper introduces the variable speed drive technology of the beam pumping unit, and reviews and discusses the development status of the dynamic research of the beam pumping unit. Secondly, this paper summarizes and analyzes the application effect of the variable speed operation optimization technology of the beam pumping unit. Finally, this paper proposed the existing problems of variable speed operation technology and future research and development directions.

2. Variable speed drive

In a beam pumping unit that is driven with variable speed, a speed controller is installed in front of the motor without replacing the original equipment in the pumping system. Based on the required motor power, the preferred variable speed drive curve is performed based on the Fourier series expansion, where the calculation of parameters, such as comprehensive power saving rate and periodic load coefficient, are performed. Under the goal of maximizing the overall power saving rate, the variable speed driving curve is optimized. The optimized speed curve is converted into frequency signals and input to the speed controller for optimizing the pumping units. The operating steps are as follows:

1. Input the structural parameters and the production parameters of the pumping units to calculate or measure the motor input power.

2. The balance adjustment of the pumping units is chosen based on the current balance method; the best current balance is selected as 85%.

3. Fourier coefficient expansion is performed on the input power curve. The variable speed drive curve is optimized under certain constraint conditions: the maximum stress of the rod is less than the permissible stress; the maximum output torque of the gearbox is less than the permissible torque; the cyclic load factor of the variable speed drive is less than the load factor of the constant speed drive; the oil output of the pumping system under variable speed conditions is less than the oil output of the pumping system under constant speed conditions.

4. Convert the optimized variable speed drive curve into a discretized frequency signal and input it to a variable frequency controller. In this way, the pumping system runs in an energy-saving and stable manner.

5. After achieved the variable speed curve, a comprehensive performance evaluation can be carried out in thinking about the four-bar motion functions and the rod wave equations. The following parameters can be calculated: angular acceleration of the motor, output shaft torque, output power, suspension point displacement, velocity, acceleration, load, output net torque of the gearbox, balance torque, polished rod torque, inertia torque, overall power saving rate, periodic load factor, degree of balance, etc. After that, the performance parameters before and after the variable speed optimization operation are compared. According to the analysis results, the variable speed optimization operation curve is further improved.

The variable speed drive optimization process is shown in Fig. 1. The variable speed drive is suitable for synchronous motors. However, for three-phase asynchronous motors, the slip rate is generally less than 3%. This method also can be applied to three-phase asynchronous motors regardless of the influence of the slip rate on the variation speed optimization.

3. Development of kinematics in pumping units

3.1. One-dimensional modeling of the rod pumping system

In the 1950s, the non-profit organization Rod Pump Research Institute established an API method for calculating the load of rod pumping system. This method applied a semi-empirical formula with a relatively low accuracy (Halderson, 1953). In 1963, Gibbs (1963) created a one-dimensional damping wave model as shown in Eq. (1), which is used to describe the longitudinal vibration of the rod, and to predict the indicator diagram of optional cross-sections on the polished and the sucker rod. Fig. 2 presents a suspension point indicator diagram based on the damping wave equation (1). From then on, using the wave equation to solve the vibration load issues of rods is widely applied in various oilfields. Later, Gibbs (1963) and Gibbs and Neely (1966) further extended the application range of the one-dimensional wave equation to present downhole pump card, rod stress, and pump inlet pressure. Based on the comparisons between the wave equation and the API method (Gibbs, 1977, 1982), Gibbs pointed out that the one-dimensional wave equation method is more applicable to pumping units, sucker-rods, and electric motors in optional subsurface working conditions.

$$\frac{\partial^2 u(x,t)}{\partial t^2} = a^2 \frac{\partial^2 u(x,t)}{\partial x^2} - \frac{\pi a v}{2L} \cdot \frac{\partial u(x,t)}{\partial x} \tag{1}$$

In 1990, Svinos (1990) presented a multi-composite rod wave model based on the application of equivalent stiffness and equivalent density on the rod cross-section with the diameter and material variations. However, this model does not consider the continuous conditions at the changing rod cross-section. In 1988, Lea (1988) established a multi-composite rod wave model based on the continuous conditions of force and displacement at the interface between two adjacent rods, which highly improved the simulation accuracy.

The above studies only considered the one-dimensional axis vibration of the sucker rod and have not considered the vibration of the tubing and the corresponding liquid column. This type of model is often called a one-dimensional simulation model of the rod pumping system.

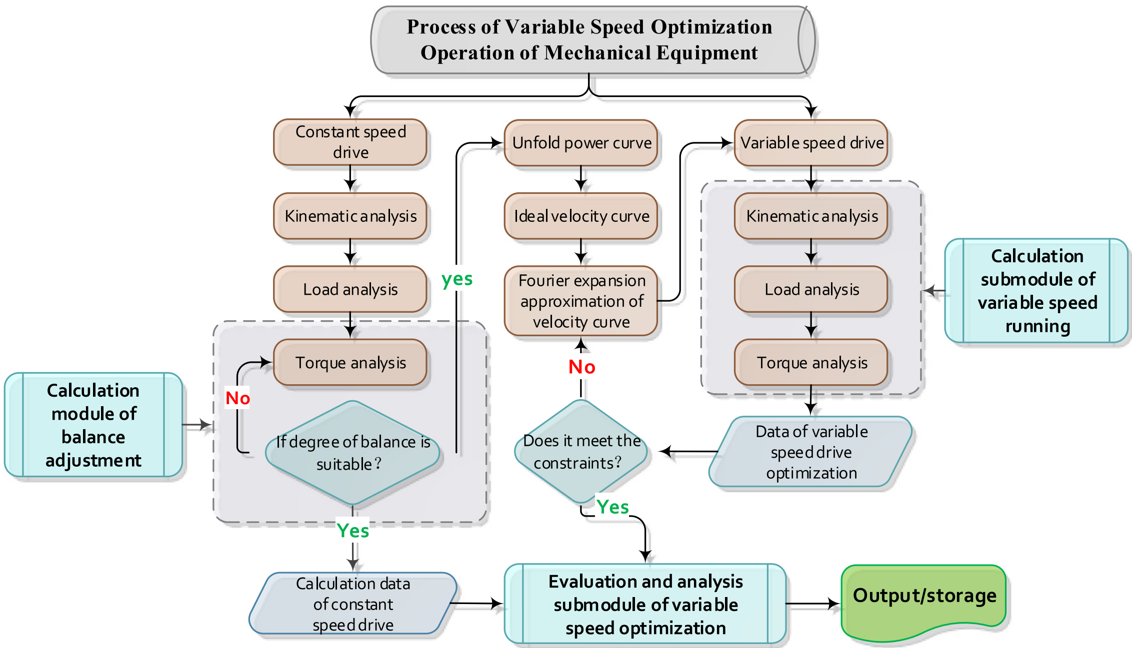

Fig. 1. Process of variable speed optimization operation.

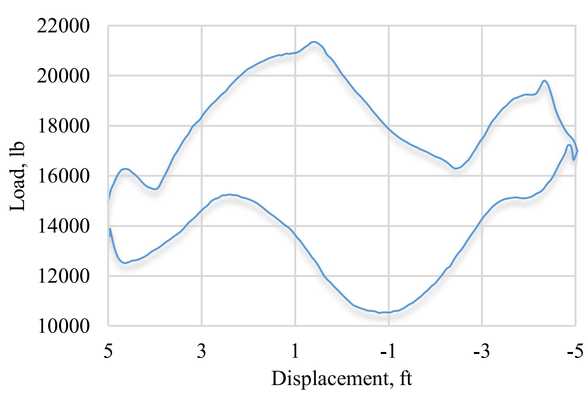

Fig. 2. Suspension point indicator diagram based on the damping wave equation (1) (Beckwith, 2014).

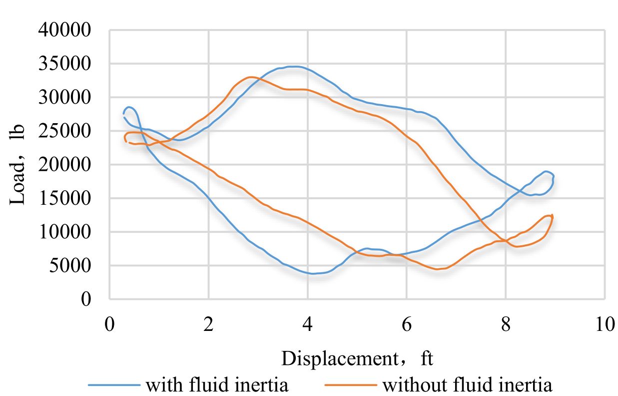

Fig. 3. Effect of fluid inertia on polished-rod dynagraph (Doty and Schmidt, 1983).

3.2. Two- and three-dimensional modeling

In 1987, Chaci and Purcupile (1987) discretized the sucker-rod into a damped spring–mass system and established a system of ordinary differential equations to describe the vibration of each mass unit. This method is different from the partial differential equations of the wave equation. Some scholars are still using this method to investigate the dynamics of the rod. In Gibbs' study (Gibbs and Neely, 1966), the established model based on wave equations can reflect the influence of liquid inertia on the dynamic parameters of the pumping system, ignoring the elasticity effect from the liquid column.

The vibrational coupling of the sucker rod and the liquid column in the tubing was first studied by Doty and Schmidt (1983) in 1983 as displayed in Eq. (2). Fig. 3 presents a suspension point indicator diagram based on the damping wave equation (2).

$$\begin{cases} \rho_r A_r \frac{\partial v_r}{\partial t} = \frac{\partial f_r}{\partial x} + F_{rf} + F_{cf} + F_{rt} - \rho_r g A_r \\ EA_r \frac{\partial v_r}{\partial x} = \frac{\partial f_r}{\partial t} \\ \rho_f (A_t - A_r) \left(\frac{\partial v_f}{\partial t} + v_f \frac{\partial v_f}{\partial x} \right) \\ \quad = -(A_t - A_r) \frac{\partial p_f}{\partial x} - \rho_f (A_t - A_r) g - F_{rf} - F_{cf} + F_{ft} \\ \frac{\partial \rho_f}{\partial t} + \frac{\partial}{\partial x} (\rho_f v_f) = 0 \end{cases} \quad (2)$$

The axial force and the axial velocity at any section of the rod were taken as two parameters to describe the longitudinal vibration. The liquid column pressure and velocity at any section of the rod were used to describe the vibration of the liquid column. In 1991, Csaszar et al. (1991) applied a two-dimensional wave model to simulate shallow wells. The results showed the liquid column inertial load has a large impact on the pumping system that was working in shallow wells with large pump and low compressibility well fluids. In some cases, large static loads were observed, indicating that liquid inertia cannot be ignored in certain situations. Tripp, Jennings, and Gibbs et al. extended the wave equations to fiberglass rod string and achieved the expected results (Tripp, 1988; Jennings and Laine, 1991; Gibbs, 1991). In 1992, Everitt and Jennings (1992) established a wave equation based on viscous resistance and finite difference method, which was applied to the diagnostic model of the sucker rod system. DaCunha and Gibbs (2009) studied the propagation of the semi-infinite-spatial-domain wave equation in a slender homogeneous elastic finite-length rod and improved the downhole pump card by improving the accuracy of the damping term in the wave equation.

Lelia and Evans (1995a) further studied the mathematical model of dynamic prediction of the coupled motion of the sucker rod and the liquid column. In this study, two different assumptions were made on the liquid column: the first is that the liquid column only contains single-phase incompressible liquids; the second is that the liquid column is a single-phase micro-compressible liquid. Tripp and Kilgor (1991) applied the rod–liquid column coupled vibration model to predict the dynamic parameters of 92 wells. The prediction results were matched well with measured results. In 1995, Lelia and Evans (1995b) proposed the Mac Cormack explicit numerical algorithm for the rod–liquid column coupled vibration modeling.

Between 1988 and 1991, Yu et al. (1989) established a numerical model that comprehensively considered the three-dimensional coupled vibration of sucker rods, liquid columns, and tubing as presented in Eq. (3). Fig. 4 displays a suspension point indicator diagram based on the three-dimensional damping wave equation (3). The assumptions made for the three-dimensional rod–liquid column–tubing vibration models are conventional beam pumping unit, low slip motor, constant crank speed, no bubble in the liquid column, constant pump inlet pressure, and ignorance of valve resistance. However, these assumptions limited the application range of this model to a certain degree. The extension of this three-dimensional wave equation model requires more verification. In 1997, Lollback et al. (1997) applied the three-dimensional wave equation to directional wells and analyzed the factors affecting the load.

$$\begin{cases} \rho_r A_r \frac{\partial v_r}{\partial t} = \frac{\partial f_r}{\partial x} + \rho_r A_r g \left(1 - \frac{\rho_f}{\rho_r}\right) - F_r \\ E_r A_r \frac{\partial v_r}{\partial x} = \frac{\partial f_r}{\partial t} \\ \rho_t (A_h - A_t) \frac{\partial v_t}{\partial t} = \frac{\partial f_t}{\partial x} + \rho_t g (A_h - A_t) - F_t \\ E_t (A_h - A_t) \frac{\partial v_t}{\partial x} = \frac{\partial f_t}{\partial t} \\ \rho_f (A_t - A_r) \frac{\partial v_f}{\partial t} = -(A_t - A_r) \frac{\partial P_f}{\partial x} - F_f + \rho_f (A_t - A_r) g \\ E_f \frac{\partial v_f}{\partial x} = -\frac{\partial P_f}{\partial t} \end{cases} \tag{3}$$

The two-dimensional and three-dimensional wave equations established in the above literatures do not consider the influence of motor variation speed on the pumping system.

3.3. Other modeling

In 1964, J.W. Hart proposed to use high-slip motors in beam pumping units. Years of experimental research and field observations have proved that the high slip drive motor is fit for beam pumping units under asymmetric cyclic load, which gives great energy-saving effects. The application of high slip motor has prompted engineers to develop the coupling of kinematics and dynamics of pumping units under variable speed drive. The coupling modeling under variable speed was first proposed by Gibbs in 1975 (Gibbs, 1975). He established the kinematics and the dynamics functions of pumping units during variable speed operations and gave the solutions for the coupling modeling. If only the kinematics need to be solved, the speed of the motor needs to be measured by a tachometer and then converted into the speed of the gearbox. The angular acceleration of the gearbox is obtained by Fourier expansion of the crank angular velocity and then derivation. If the coupling of kinematics and dynamics needs to be solved, the differential equation of the motor is required to be established, and the differential equation of the crankshaft motion is also required to be obtained through the gear ratio. The motion characteristics of the motor can be determined by its external characteristics. The load variation of the motor can be gained by the indicator diagram of suspension point to solve the crank motion differential equation. In Gibbs's work, the precondition of the solution to the coupling modeling was to test the motor speed, the motor's external characteristics, and the indicator diagram of suspension point. These data were then used as the boundary conditions to solve the coupling behavior of pumping system. However, the test is relatively difficult. Therefore, the universality of coupling model needs to be expanded.

In 1996, Dong et al. proposed a model for motor speed fluctuation prediction (Dong et al., 1996), where the motor motion differential equation and the sucker rod wave equation were involved. The indicator diagram of suspension point and the change of motor speed (or crank speed) were also presented in their paper. However, there is no comparison curve between gearbox net torque and motor output power. In 2010, Shardakov and Wasserman (2010) established a longitudinal vibration mathematical model for the sucker rod. This model considered the strong nonlinearity of the pump valve force and the Coulomb friction, which can handle the monotonicity and non-monotony association between force and speed. In 2013, Jiang et al. (2013) established a full coupling model of beam pumping units and the driving motors. Through the external characteristics of the high slip motor obtained by test, a cycle iterative method was used to solve the motor-pumping coupling equations. This method can be used to analyze the dynamic characteristics of the beam pumping unit. As shown in Fig. 5, the prediction results are in good agreement with the test data. The variable speeds in the high slip motor are passive control technology. The variable speed drive technology is different from high-slip motors and belongs to active control technology. However, the high slip motor technology provides a basic theory for the active variable speed drive technology.

In the optimizing process of beam pumping systems, engineers often treat the transmission efficiency of the belt as a fixed value. To further accurately calculate the transmission efficiency of the pumping system, Xing and Dong (2015b) established a belt transmission efficiency slip model, which is combined with a one-dimensional wave equation, to optimize the pumping system. In 2016, Xing (2016) used ANSYS software in the calculation of rod mechanics and carried out the calculation of longitudinal vibration and flexural coupling of pumping rods. The authors also presented the correlation between the equivalent stiffness and the bending of the rod. However, Xing did not consider the coupling between the liquid column and the rod. In 2016, Lao and Zhou (2016) used an inclined well as the research object to establish a coupling model of the buoyancy and the lateral force on the tubing, revealing how the buoyancy affects the dynamic performance of the sucker-rod. Again, this model does not consider the coupling between the tubing and the rod. These recent literatures have analyzed the vibration characteristics of the sucker rods in beam pumping units, such as the load at the multi-stage joint, the equivalent stiffness of the belt, the buoyancy in the tube annular, and the lateral forces of inclined wells.

Through the summarization of researches in the coupled dynamic model of motor–rod–pump, it can be seen that the relevant theoretical models are not ideal, lacking detailed qualitative analysis, quantitative research, and reliable practice. Establishing a coupled mathematical model of kinematics and dynamics of the motor–rod–pump are highly desired by the oil-fields.

4. Investigation of variable speed optimization in oil pumping

The research on the variable speed optimize operation of pumping units started quite early, beginning from intermittent oil production and changing of the pumping speed. As early as

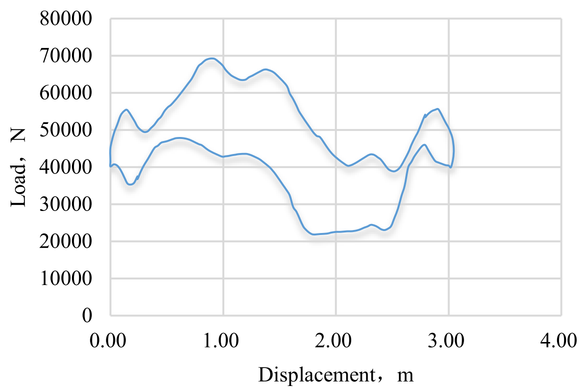

Fig. 4. Suspension point indicator diagram based on the damping wave equation (3) (Yu et al., 1989).

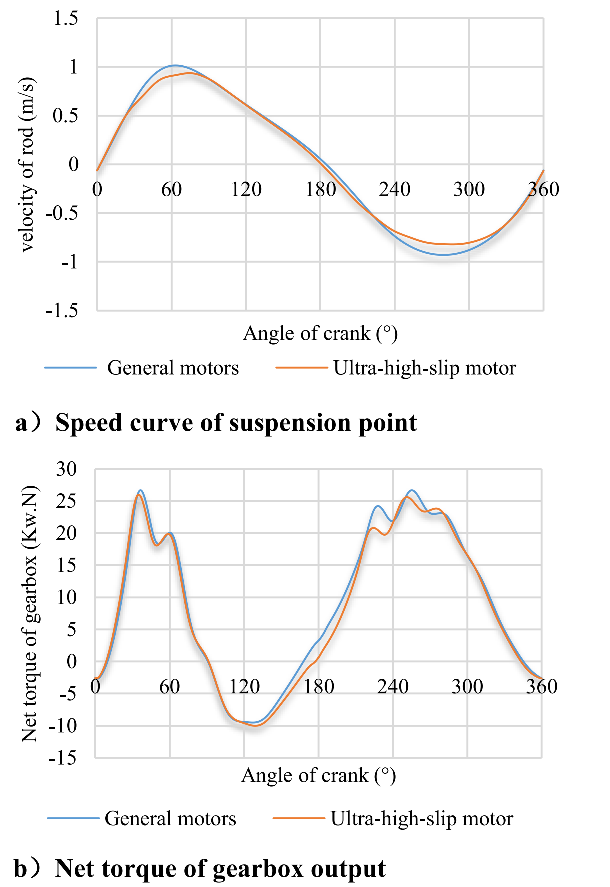

Fig. 5. Comparisons of working curves between Ultra-high-slip and general motor (Jiang et al., 2013).

1932, Coberly and Harris (1932) invented a method of changing the pump speed based on the bottom-hole oil pressure in his Patent US1957320, but the accuracy of this method is relatively low. Grable and Laney (1972) changed the pump speed based on the amount of subsurface oil production in the Patent US3807902. Gibbs (1976) used the indicator diagram to detect changes in input power in the Patent US3951209. When the power was reduced to a certain value, it was considered that pumping off occurred. At this moment, the pump was stopped for some time and the pump speed was intermittently changed. Womack and Jahns (1977) improved the variation speed control method of the Patent US3951209, which is no longer based on the power of one stroke. Using the power of the last quarter of the upper stroke and the first quarter of the downstroke to decide whether to stop pumping. In the Patent US4034808, Patterson (1977) proposed to control whether to stop pumping based solely on the power of the first quarter of the downstroke, according to the variation speed control method of the patent US3951209. In the Patent US4145161, the variation speed control method invented by Skinner (1979) is based on the proportional relationship between pump speed and electric power. But in fact, the proportional coefficient between the two is not a constant, which will cause a large error.

In 1987, Gibbs (1987) analyzed the relationship of the motor load, output torque of the motor, balance degree, polished rod load, and energy consumption of the gearbox, which provided theoretical model for the later application of variable speed drive technology. In 1989, in the Patent US5044888, Lawrence and Hester (1989) invented a variable speed control system by changing the pump speed through transmitting parameters during the first half of the downstroke to maximize the output of pump. In 1995, Gibbs et al. (1995) used a one-dimensional wave equation to calculate the downhole pump card and used the card data to determine the fullness of the pump. The fullness data will also be used to determine the pumping speed in the variable frequency drive to prevent pumping off.

Through Patent US4490094 (Gibbs, 1984), US5252031 (Gibbs, 1993) (based on the oil pump performance algorithm in the Patent US3343409 (Gibbs, 1967)), and US4973226 (Mckee, 1990), the pumping parameters are optimized by the changing of rod string, pump diameter, stroke, and speed (using variable frequency drives), or an ultra-high slip motor was used to change the motor speed for matching the polished rod load. As presented in patent US4102394 (Botts, 1978), patent US5246076 (Watson, 1993), and patent US20060095140A1 (Broren, 2005), it is proposed to limit surface facilities and reciprocating pumps to reduce the polished rod load. In the patent US6890156b2, Watson (2005) gave three-motor drive velocity curves as shown in Fig. 6. Case 1 in Fig. 6 increased the upstroke and the downstroke speed. Case 2 and 3 in Fig. 6 are to make the suspension point run on the trapezoidal velocity curve. According to the calculation of the velocity curve in Fig. 6, case 1 increased the peak load of the polished rod. Case 2 and 3 reduced the peak load of the polished rod, but there is no guarantee that the power saving rate will increase. Also, the degree of balance of the pumping unit varies greatly.

Beginning in January 2004, in the PDVSA (a Venezuela field) of San Tome Field Orinoco Belt, around 100 oil wells were combined with Rod Pump Controller and Variable Speed Drives to ensure that the fullness of the pump is high by maintaining a stable hydrodynamic surface. The yield increase ranged from 10% to 160%. But, the optimized velocity drive curve results in the net torque of gearbox more than the rating torque of the gearbox. By limiting the gearbox torque to the gearbox rating, the VSD controller reduces the downstroke speed during incomplete pump fill to protect the rod and pump (Peterson et al., 2006). Since 2007, an oilfield, operated by Santos, has installed variable speed drives in over 400 oil wells with the remote monitoring. After four years of application for the beam pump wells, Santos' automation program decreased the failure frequency and a 75% reduction in operating cost and a 3% reduction in downtime. The yield increase was between 18%–30%. If there is no clear and comprehensive automation strategy, the VSD technology will not be able to realize its full potential, such as standard hardware using best in class OEM, remote monitoring software platform, a depth training program (Radzevicius and Clarke, 2012).

Palka and Czyz (2008) proposed a downhole fluid production optimization method for beam pumping units to maximize fluid

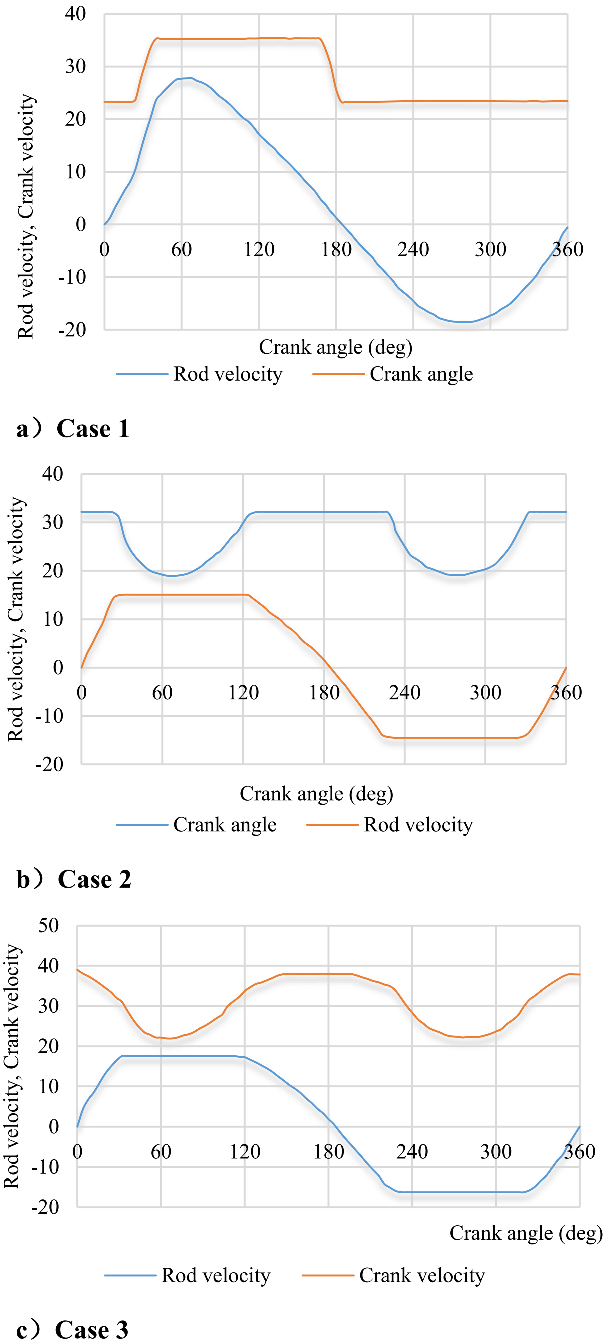

Fig. 6. Variable speed curve.

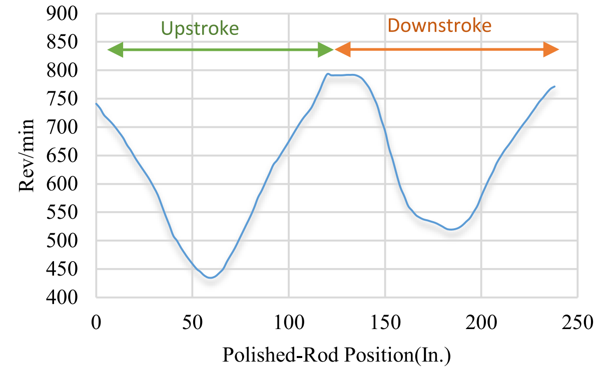

Fig. 7. Optimum motor speed (Palka and Czyz, 2008).

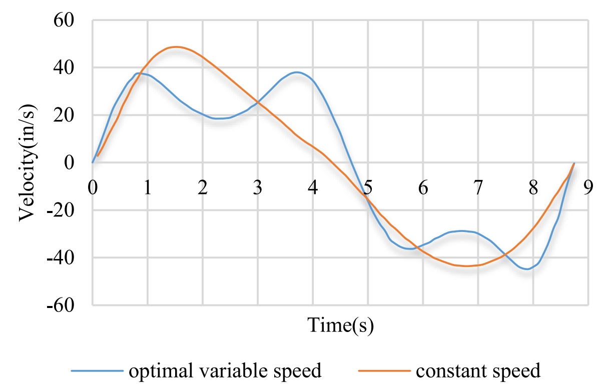

Fig. 8. Speed of polished rod (Palka and Czyz, 2008).

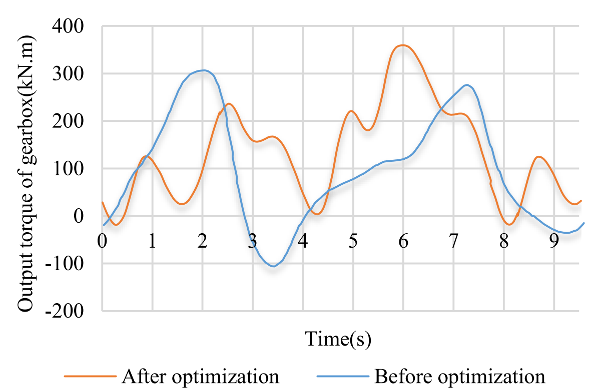

Fig. 9. Net torque curve of gearbox output (Palka and Czyz, 2008).

production (Palka and Czyz, 2009). The method increases oil production by improving the motor drive speed during pumping cycles, while also reducing the rod stress and motor energy consumption. The motor optimal drive speed curve is determined by the Fourier series expansion of the motor speed (Fig. 7). The Fourier coefficient is determined based on the production maximization target which meets the given rod stress and power consumption. The pump performance parameters that need to be obtained during the optimization process, such as the amount of produced liquid, the rod stress, and the motor torque caused by the variation speed, are calculated through predictive analysis. The optimized variable speed curve presented by Pumpwell Solution Ltd has been used to drive many pumping wells (Fig. 7 and Fig. 8). The measured results (Palka and Czyz, 2008) indicated that the variable speed drive technology can certainly increase pump stroke and production. However, as shown in Fig. 9, from the given torque curve, the peak torque increased even more.

In 2010, Jiang et al. (2010) optimized motor variation speed to reduce the polished rod load. The polished rod movement curve is shown in Fig. 10, where the rod load and its fluctuation can be reduced based on this speed model. The speed optimization curve proposed by Jiang et al. (2010) is similar to a sinusoid and the starting point has a certain phase angle. A disadvantage of this speed profile is that the negative torque increases at a position near 360° of stroke. However, in the working of the pumping well, it is not desirable to have a negative torque. In 2011, Sam et al. (2011) used six new measurement tools for SRP (an automated fluid level measurement tool provided by RAG Austria) to monitor the depth of downhole fluids and used variable speed drive technique to control the pump speed to avoid bottom hole pressure below the bubble point pressure. But this new measurement tool has no function of pro-active maintenance before production failure occurs for fewer systems downtime.

In 2015, Elmer (2015) applied the pump stroke optimization technique to oil wells of the Eagle Ford field. The technical feature is to reduce the downstroke speed and to keep the upper stroke speed constant, where pump leakage is decreased and air anchor obtains more time to separate the gas. The application indicated that ten wells were very successful, five were barely successful, and five were unsuccessful. In the same year, the variable speed drive technology was used in the Cooper Basin in central Australia. This technology has the advantages of self-feedback yield maximization and equipment protection, which is a preferred device for beam pumping units. VSD wells have decreased a 64% failure frequency compared with wells with no automation, and 76% of pump-failure incidents on VSD wells occurred within 2 years of pump installation, which indicated the VSD technology need to be improved (Carpenter, 2015).

In 2016, to achieve more accurate fluid level information, Rohoel-Aufsuchungs AG (RAG) has developed an electronically controlled liquid level tester (without a gas gun like in a conventional hydrodynamic test instrument) to record the depth of the working fluid level. Passing the input signal to the variable speed drive controller achieves the goal of preventing pumping off and maximizing production. Furthermore, they developed a transient multiphase flow model for production performance monitoring which shows a very good agreement to the data of downhole gauges. RAG has used fluid level measurements in many cases in the past two years, and saved approximately half a million EUR, and provided an important contribution to the mature oilfields (Burgstaller, 2016).

Also, sucker rod pump systems with VSD have been applied to 550 wells in Murphy Oil's Eagle Ford unconventional shale fields. The authors discussed the application of variable speed drive technology and the reasons for failures. This VSD has real-time monitoring and optimizes control functions in the unconventional oil well, and is connected to desktop and enterprise monitoring software through the SCADA system for management on the variability of inflow, gas interference, and natural decline (Clarke and Malone, 2016).

In 2016, Dong et al. (2016b) used the real-time variation of frequency to establish a model for the variable frequency control system in beam pumping units. In this study, the minimum power of the motor is the optimization target. The real-time frequency optimization model is launched under the constraints of constant pumping speed, equality of up & down stroke time, and limited frequency range. This method used the constrained nonlinear optimization system and the penalty function to find out the optimal solution. Dong et al. introduced (Dong et al., 2016a) the method of frequency conversion in the optimal operations of beam pumping units. In their point, the inertia and balance adjustment of moving parts under variable speed operations cannot be ignored, and the constraints should include multiple nodes of the pumping unit system.

To evaluate the influence of the pumping, balance device, frequency function, motor output shaft, and their coupling relationship, Dong et al. (2016b) designed three cases (see Fig. 11):

- Case one: Optimization of the pumping parameters (such as stroke, down pump depth and rod string) and balance parameters;
- Case two: Optimization of pumping parameters, balance device parameters, and real-time frequency function;
- Case three: Optimization of pumping parameters, balance device parameters, real-time frequency function, and moment of inertia of belt on the motor output shaft.

Fig. 11 shows the power curve for these optimization calculations. The curve envelope area represents energy consumption.

In 2017, as to Cyclic Steam Stimulation, Alwazeer et al. (2017) develop an automation algorithm for variable speed drive to improve well management with flexibility. This algorithm maximizes inflow potential by keeping the working liquid level slightly above the level when it was pumped off. This method has improved recovery and significantly decreased manpower demand by 20% of total surveillance time. In 2018, with horizontal wells, during a gas slug, the VSD will slow pump speed for increasing the fullness of the pump, which will reduce wear and stress on the rods and downhole pump (Allison et al., 2018). After the average SPM dropped from seven to four, the average fullness of the pump increase, and the production increase from 70 to 100 bbl/D.

To reduce pump leakage and provide enough time for the gas and liquid separation in the gas anchor, the literature (Elmer and Elmer, 2018) used rod-pump controllers with a variable-frequency drive (RPC-VFDs) to control the pumping speed in 20 Eagle Ford wells in March of 2015. For instance, the upper stroke speed was reduced, and the downstroke speed remained unchanged. This fast upstroke/slow downstroke solution has many benefits, such as decreased pump leakage, increased fullness of the pump, decreased power reduction, increased production. But the test indicated that not all the RPC-VFDs wells have good results.

Ferrigno et al. (2018) comprehensively studied 50 wells using Intelligent Well Controller and Telemetry Systems. Both techniques used wave equations to calculate indicator diagram and analyze ground production data to optimize the production of wells, including controlling motor speed and pump strokes. This VSD technology can increase the pump fullness with gas compression or fluid pound, but they did not explain the causes of premature breakages of rods.

5. Cases study

The optimization control technology of variable speed of beam pumping unit has a long history and has undergone many changes and improvements. The variable speed has a great impact on the power, torque, power saving rate of the motor, the net torque of the gearbox, the periodic load factor, the suspension point load, the pump load, the actual stroke of the pump.

Different variable speed curves will bring different effects on the working performance of the pumping unit system. It is a difficult thing to improve all the above performances. Therefore, this paper chooses three variable speed curve in the literatures (Palka and Czyz, 2008; Watson, 2005; Jiang et al., 2010) to calculate, and draws the performance curves of the motor, gearbox, suspension point and pump after the variable speed operation of the pumping unit, and the comprehensive performance is compared and analyzed as following.

5.1. Case one

The variable speed optimization proposed in Palka and Czyz (2008) is one of the more advanced energy-saving technologies for pumping wells and has been widely used in many fields. In China, this technology has been applied to 121 wells in the Daqing oilfield, of which 18 wells had a negative power saving rate, 48 wells had reduced system efficiency, 52 wells had reduced pump efficiency, and rods were overloaded with 69 wells. Fig. 10 shows the motor speed curve of the variable speed drive with an average speed of 750 r/min. When performing the variable speed operation evaluation, all parameters remain the same except the speed curve.

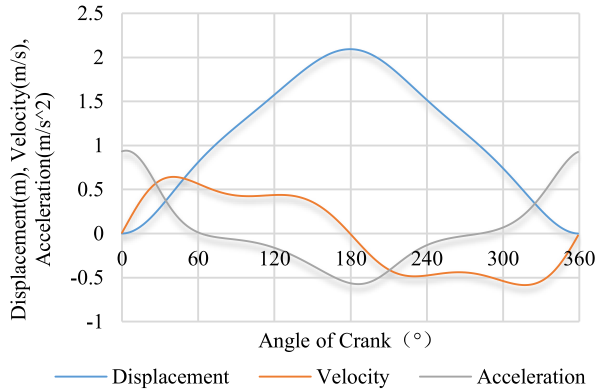

Fig. 10. Operating parameter curves of variable speed driven pumping unit (Jiang et al., 2010).

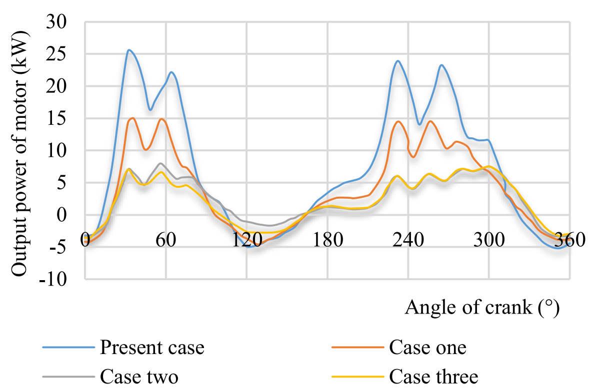

Fig. 11. Comparisons of motor output power curves (Dong et al., 2016a).

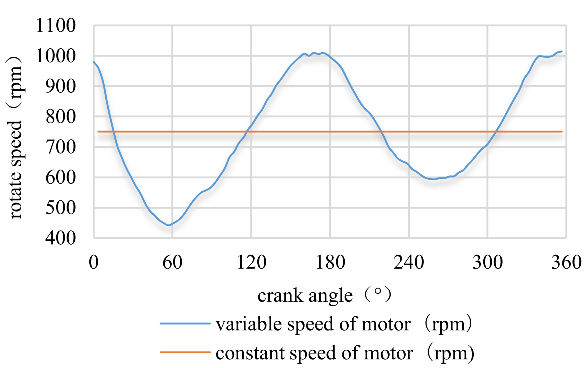

Fig. 12. Speed curve of motor in variable speed operation.

The variable-speed drive optimizing simulate calculation was programmed by Visual Basic 6.0. The simulation procedure has been shown in Fig. 1. The program can run on WinXP®, Win7® and Win10® operating systems. The simulator is used for an oil well in Daqing Oilfield (China) for the optimization calculation and analysis of variable speed operation. The optimized speed curve of the prime motor is shown in Fig. 12. The input parameters of variable speed optimization operation has been displayed in Tables 1 and 2, including structure parameters and production parameters. Pumping units CYJ10-3-37HB was selected as an object to analyze its performance. The rated power of the prime motor was 22 kW.

The motor speed varies between 40%∼137%. The periodic velocity distribution is: speed is high near the upper and the lower dead spots, and reduces at other positions with the decreasing of the load amplitude. The basic energy-saving idea is "heavy-load with slow-drive, and light-load with fast-drive" to maintain the smoothest input power, improve the operating efficiency of the motor, and increase the overall performance. Due to the rotational inertia of rotating parts of the system, the transmission characteristics of the four-bar mechanism, and the three-dimensional coupling dynamic characteristics of the rod–tubing–liquid, it is difficult to evaluate the variable speed operation theory with a single performance index of pumping system.

Table 1
Structure parameters of sucker-rod pump system.

Structure parameters	Value
Fore-arm length (mm)	3000
Aft-arm length (mm)	2400
Linkage rod length (mm)	3350
Rotational inertia of motor (kg m^2)	1.39
Rotational inertia of walking beam (kg m^2)	2140
Horizontal throw (mm)	2300
Crank radius (mm)	1150
Center-height	5290
Rotational inertia of crankshaft (kg m^2)	5330
Rotational inertia of gearbox (kg m^2)	2.54

Table 2
Oil well condition and swabbing parameters.

Swabbing parameters	Value
Frequency of stroke (min^{-1})	6
Stroke (m)	3
Depth of plunger (m)	1000
Working fluid Level (m)	600
Oil pressure (MPa)	0.5
Pump diameter (mm)	57
Rod diameter (mm)	22
Tube diameter (mm)	76
Water cut (%)	90
Tube pressure (MPa)	0.4

Figs. 13–16 are the comparisons of the variable speed and the constant speed operation curves. The peak of the suspension point curve is shaved and the acceleration of the upper stroke is extended to reduce the inertial load on the polished rod. The net torque of the gearbox is reduced by 6.22%, and the most negative torque is eliminated. When the horsehead is near the bottom dead spot, some of the negative torque is not eliminated. Compared with the torque curve, the motor power curve has a very large amplitude dropping. The power reduction benefits from the combined effect of the inertia load and the low-speed curve section, which is very effective for improving the efficiency of the motor. Based on the polished rod indicator diagram, it was found that the polished rod load was reduced by 2.75% after the variable speed operation, which is about 1.5 kN.

As can be seen in Fig. 17, the motor torque, power, power saving rate, cyclic load factor, net gearbox torque, polished rod load, and pump displacement have been improved to varying degrees. The maximum torque of the motor shaft is reduced by 14.71%; the maximum output power of the motor is reduced by 34.96%; the motor power saving rate is 2.91%; the cyclic load factor is reduced by 22.03%, and the maximum output net torque reduction rate of the gearbox is 6.22. %, the maximum load reduction rate of the polished rod is 2.75%, the over-stroke of the pump is 0.28%, and the maximum pump load is increased by 1.73%. Among all the 8 performance items, 7 performance items have been improved, indicating that the variable speed drive technology has the effect of improving the performance of the conventional pumping system. The variable speed drive technology can make fully using of the inertia energy of moving

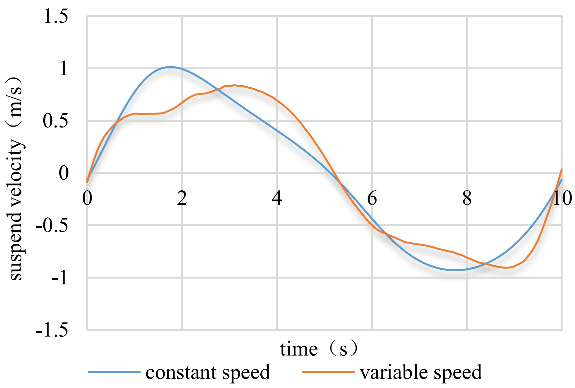

Fig. 13. Velocity curve of suspension point.

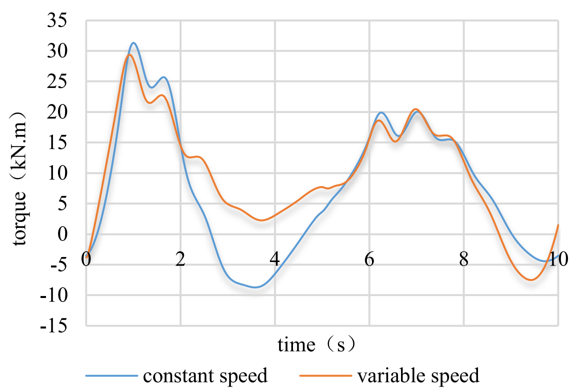

Fig. 14. Net torque curve of gearbox.

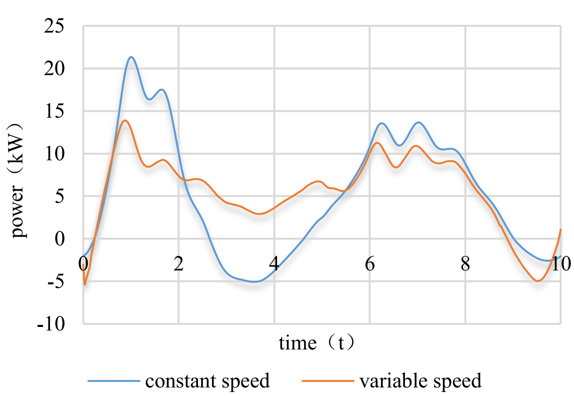

Fig. 15. Power curve of motor.

Fig. 16. Dynagraph card.

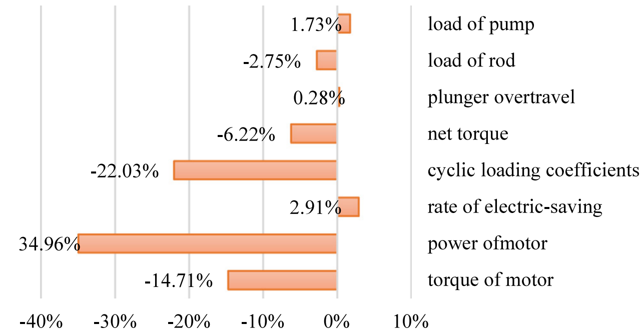

Fig. 17. Comparison of comprehensive performance before and after variable speed optimization.

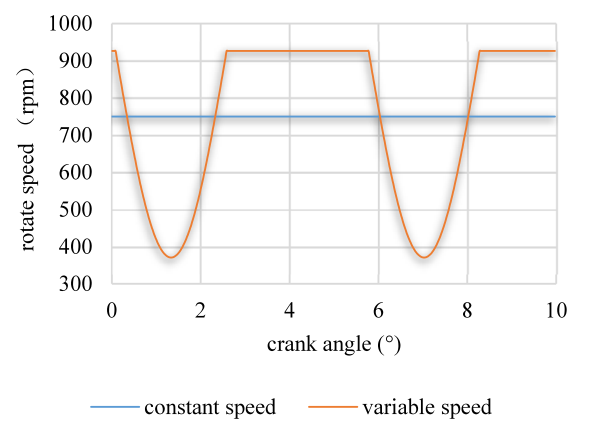

Fig. 18. Double concave speed curve of motor in variable speed operation.

parts in the pumping unit so that it makes the periodic fluctuation load curve smoother than before. There are different degrees of reduction of the peak load of transmission parts such as motor, belt, reducer, four-link bar and polished rod, which is conducive to the safety of the system operation. Over-stroke of pump can increase production, but in our case, the over-stroke of pump is very weak. The peak load of pump is increased by 1.73%. The relationship between pump stroke and peak load needs further study.

5.2. Case two

In this case analysis, the optimization speed curve in reference (Watson, 2005) is selected. Fig. 18 shows the motor speed curve with a maximum of 926.4r/min, a minimum of 370.6r/min, and an average speed of 750r/min. In the evaluation of the variable-speed operation, the system structure parameters and production parameters are the same as **Case One** except for the speed curve of the motor.

The speed of the motor varies from 49% to 123.5% of average speed 750r/min. The principle of periodic speed distribution: according to the conventional beam pumping unit with a balanced degree of 1, the speed curve shape of the motor is reversed as to the net torque curve of the gearbox. The speed curve is composed of a half part of a sinusoidal curve and a straight line. It is to verify the energy-saving effect of the principle of “heavy-load with slow-drive, and light-load with fast-drive”.

Figs. 19–22 are the comparison of the working characteristic curves of the variable speed drive and the conventional constant speed drive. In Fig. 19, the suspension point speed curve by variable speed drive is not smooth, and it may cause a serious

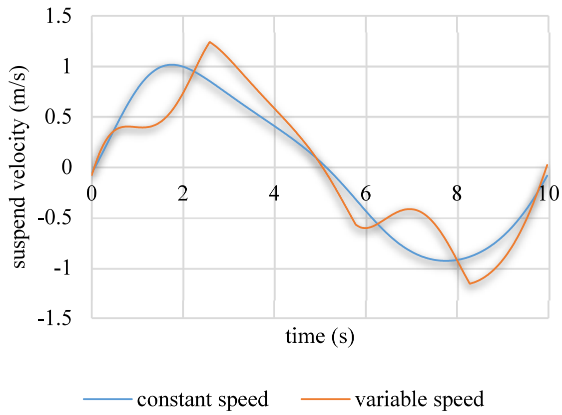

Fig. 19. Velocity curve of suspension point.

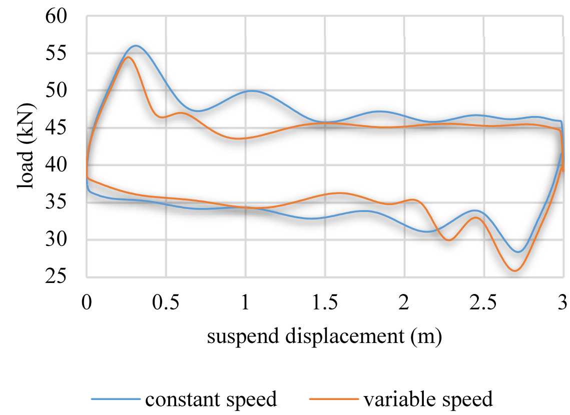

Fig. 22. Dynagraph card.

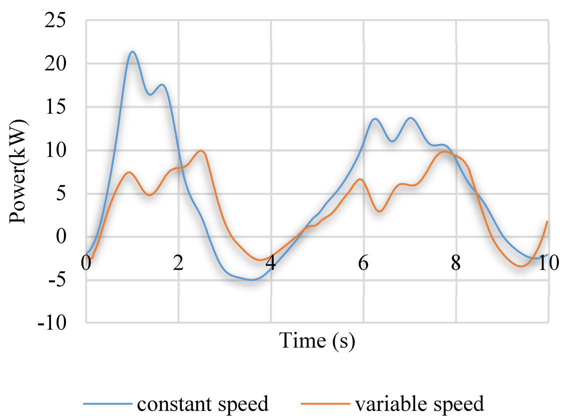

Fig. 20. Net torque curve of gearbox.

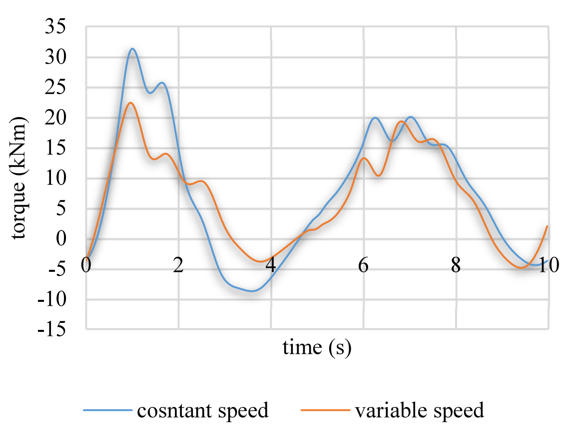

Fig. 21. Power curve of motor.

impact on the pumping well system, which is not conducive to safe operation.

From the comparison of the curves in Figs. 20–22, it can be seen that the peak values of the net gearbox torque, the motor power, and rod load have been reduced with different degrees, and the negative torque of the gearbox and the negative power of the motor are also reduced. The reduction of negative power and negative torque is beneficial to improve the efficiency and the safety of the pumping system.

As can be seen from Fig. 23, several performance indicators of the pumping system have been improved with different degrees. The maximum torque of the motor is reduced by 34.56%. The maximum output power of the motor is reduced by 53.76%. The power-saving rate of the motor is 32.87%. The periodic load factor is reduced by 8.93%. The maximum net torque of the gearbox is reduced by 28.57%. The maximum load reduction rate of the polished rod is 2.82%. The over-stroke of the pump is 0.38%. The maximum pump load is increased by 1.38%. Among them, seven of the eight performance indicators have been improved, only the pump load is increased, which is an unfavorable factor. It is indicated that the variable speed drive curve in this case can improve the overall working performance of the conventional beam pumping units.

5.3. Case three

The optimization curve of Case three is selected in reference (Jiang et al., 2010). In Fig. 24, the variable speed drive curve of the motor is sinusoidal, with a maximum of 975r/min, a minimum of 525r/min, and an average speed of 750r/min. In the calculation of variable-speed operation, except for the speed curve, the units' structure parameters and production parameters are the same as Case one.

The speed of the motor varies from 70% to 130% of 750r/min, and it has a symmetrical structure as a sine curve. This case assumes that the net output torque of the gearbox is an ideal sinusoid with no distortion. The variable speed drive curve of the motor is also distributed according to the principle of "heavy-load with slow-drive, and light-load with fast-drive".

Figs. 25–28 is the comparison of operating characteristic curves between the variable speed operation and the conventional constant speed operation. In Fig. 25, by the variable speed drive, the peak suspension point speed decreases and moves backward, which may reduce the impact of the load on the pumping unit. From the comparison of the curves in Figs. 26–28, it indicated that the peak net torque of the gearbox, the motor power, and the suspension load at a variable speed are all lower than at a constant speed. The negative torque of the gearbox and the negative power of the motor are also reduced, but in the last two seconds of the cycle, the negative torque and negative power increase, which is not conducive to energy-saving. It indicated that this speed curve needs to be further optimized.

As can be seen from Fig. 29, motor torque, motor power, power saving rate, periodic load factor, net gearbox torque, polished rod load, pump displacement all have been improved with different degrees. The maximum torque of the motor shaft is reduced by 16.91%. The maximum output power of the motor is reduced by 32.43%. The motor power saving rate is 12.56%. The periodic load factor is reduced by 5.59%. The reduction rate of the maximum

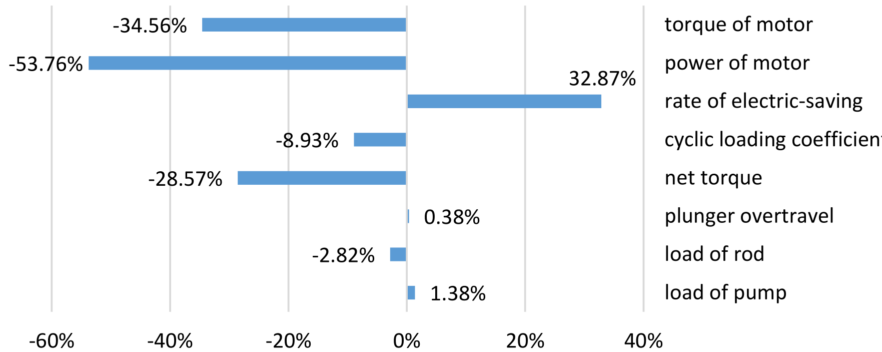

Fig. 23. Comparison of comprehensive performance before and after variable speed optimization.

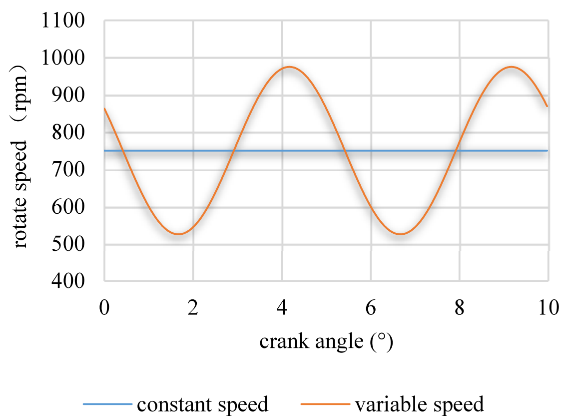

Fig. 24. Speed curve of motor in variable speed operation.

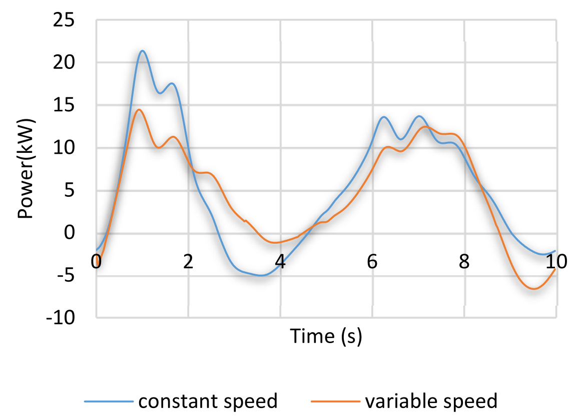

Fig. 26. Net torque curve of gearbox.

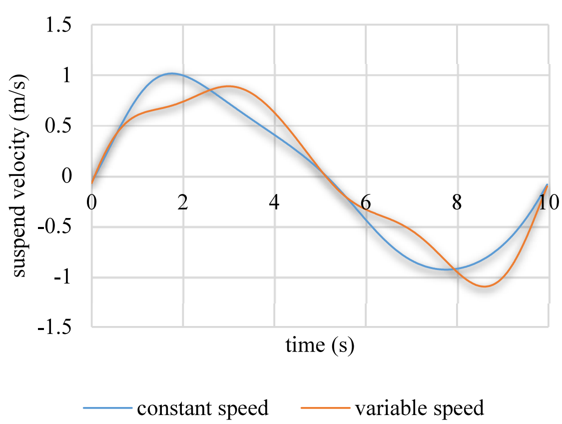

Fig. 25. Velocity curve of suspension point.

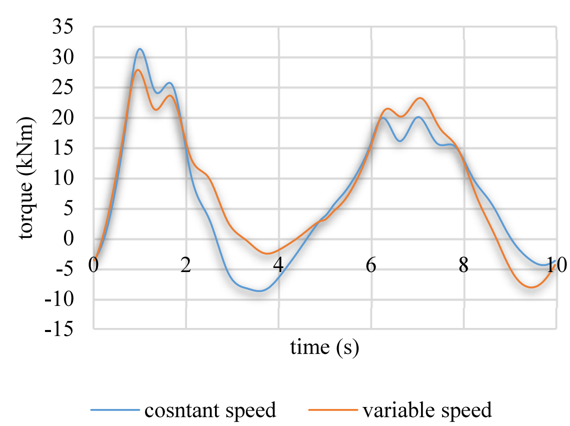

Fig. 27. Power curve of motor.

output net torque of the gearbox is 11.16%. The maximum load reduction rate of the polished rod is 0.78%. The over-stroke of the pump is zero. The maximum pump load was increased by 3.9%. Among them, five of the eight performance indicators have been improved. The suspension point load changes very little, and the pump has no over-stroke. The increased pump load is an unfavorable factor.

5.4. Variable speed optimization case analysis results

(1) Variable speed operation technology can reduce the peak torque and peak power of the motor, improve the power saving rate of the motor, reduce the alternating impact load on the motor, and reduce the rated power of the motor;

(2) Variable speed operation technology can reduce the peak torque of the gearbox and reduce the cyclic load factor of the pumping unit, reduce the alternating impact torque on the pumping unit, increase the running stability of the pumping unit, and make the pumping unit more reliable;

(3) Variable speed operation technology can reduce the peak load of the polished rod and increase the pump stroke, that is, reduce the alternating tensile and compressive stress of the sucker rod, improve the life of the rod and increase the pump efficiency;

(4) The variable speed operation technology can adapt to the changes of the pump stroke, speed, and submergence depth, but the phenomenon of "over-torque" may occur when the system is over-balanced, especially under the influence of the initial phase angle of the variation speed curve is obvious.

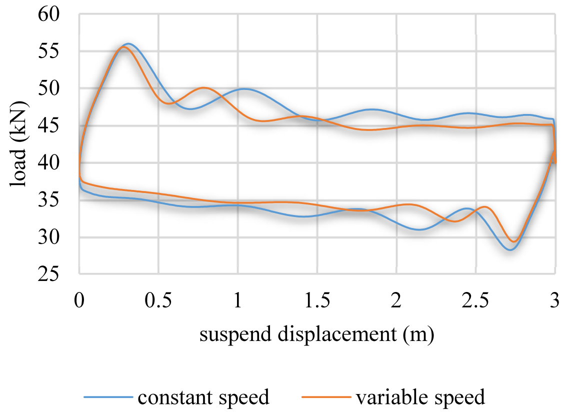

Fig. 28. Dynagraph card.

6. Conclusions and recommendations

The variable-speed operation technology of pumping units is one of the most advanced energy-saving technologies that can be applied in oilfields. From the initial pumping speed operation to the current full-cycle variable speed control operation, the optimization goal is to overcome the inherent defects of the four-bar mechanism. In the case of constant speed drive pumping unit, kinematics and dynamics models have been well documented. One-, two-, and three-dimensional wave equations for calculating rod load and corresponding numerical solutions have been established. At the same time, by testing the high slip motor speed & the polished rod dynamometer, and using this data as the boundary condition, the variable speed coupling model between the high slip motor and the pumping unit has also been solved. However, models of the fully coupled dynamic behavior of the motor–rod–liquid–pump still need more investigations. Also, solutions to a series of issues caused by variable speed operation still need to be research, such as pump leakage, opening & closing of pump valves, energy-saving mechanism, the influence of inertial load caused by moving parts, balance adjustment and comprehensive performance evaluation.

From the patents worldwide, the proposed patents for variable speed control technology in China mainly focused on hardware implementation, and there are not many patents that focus on optimization strategies or methods for specific variable speed operation. From the patent represented by many countries, most of them show the specific variable speed optimization curve, but it do not specify which optimization method of the speed curve was used for, and there is no clear mathematical model.

Regarding the current study of the dynamic performance of the pumping system, most researchers are more accustomed to building models on the motor, rod liquids, and pump, respectively. To thoroughly understand the energy distribution of the pumping system and find out the potential of energy saving, it is necessary to use the motor–rod–liquid–pump as a research object to carry out the following research:

(1) Research on the establishment of a coupled motor-rod-liquid-pump dynamic model of beam pumping units under variable speed drive;

(2) Research on the energy-saving mechanism and its comprehensive performance;

(3) Carry out the experimental analysis on the comprehensive performance of the beam pumping unit when the variable speed drive is applied.

Nomenclature

v_r—the rod velocity, m/s;
v_t—the tube velocity, m/s;
v_f—the liquid column velocity, m/s;
f_r—the inner force of rod, N;
f_t—the inner force of tube, N;
ρ_r—the density of rod, kg/m^3;
ρ_f—the liquid density, kg/m^3;
ρ_t—the density of tube, kg/m^3;
A_r—the sectional area of rod, m^2;
A_t—the inner area of tube, m^2;
A_h—the outer area of tube, m^2;
E_r—the rigidity of rod, N/m^2;
E_t—the rigidity of tube, N/m^2;
E_f—the rigidity of liquid column, N/m^2;
F_r—the resistance met with rod, N/m;
F_f—the resistance acted on the liquid, N/m;
F_t—the resistance met with tube, N/m^2;
P_f—the pressure of liquid, N/m^2;
a—the velocity of force propagation, ft/s;
υ—the damping factor, dimensionless;
x—the distance along unstrained sucker rod measured from the polished rod, ft;
t—the time, seconds
L—length of sucker-rod string, ft;
F_{rf}—the force per foot of rod arising from the viscous forces of the fluid acting on the rod surface, N/ft;
F_{cf}—the viscous force of the liquid on the rod couplings, N/ft;
F_{rt}—the force per foot of rod arising from the friction between the tubing and rods, N/ft;
F_{ft}—the viscous force associated with the tubing, N/ft.

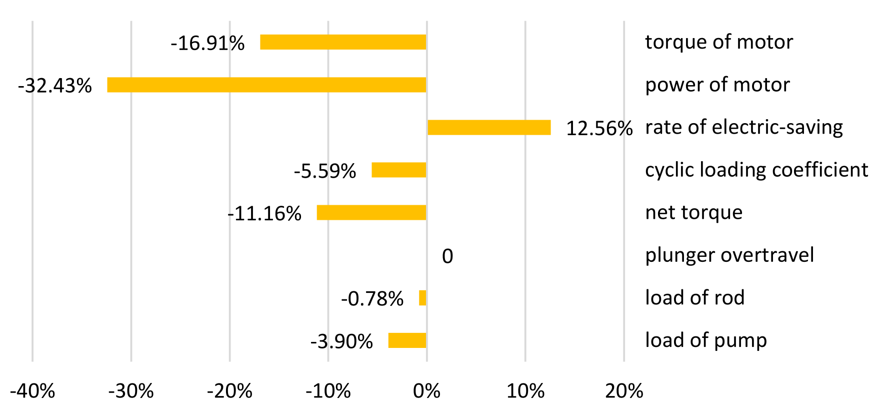

Fig. 29. Comparison of comprehensive performance before and after variable speed optimization.

Declaration of competing interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

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