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Research Article

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Passing capacity calculation method of screw pump in directional wellbore

Minzheng Jiang, Daxing Sun and Lingyu Zhang

Northeast Petroleum University, Daqing, Heilongjiang, China

ABSTRACT

The influencing factors to the passing capacity issue of screw pump in directional wells are pump length, casing size and dogleg angle, wherein casing size and dogleg angle is immutable. On the premise of no change in pump length, an axial force is applied to an end of pump, forcing the pump flexural deformation to increase pump passing capacity. The theoretical analysis to this squeeze pump method has been done, and then the calculation model of the pump through the dogleg is built, under specific conditions the maximum allowed value of pump setting depth is derived, in the calculation model, considering the impact of bending of pump on stator rubber structure. Case studies show that, some dogleg cannot be through merely under the weight of the pump can be through by adopt the method of squeeze pump, according to the calculation results can determined the maximum allowed value of pump depth, while avoiding stator rubber being snapped and degumming in the squeeze process.

Keywords: Directional well, screw pump, passing capacity, passing by squeeze, stator rubber degumming.

INTRODUCTION

As thin and poor reservoir started mining, the advantages, efficiency and energy saving, of screw pump in directional well system can be played, but the issue about passing capacity of pump remains difficult. Directional wellbore trajectory is a space curve, in actual drilling will inevitably produce several larger dogleg. In this case, designers' usual disposal method is set the pump in vertical section or replaces the pump with a shorter one. Those two approaches are safe and without technical difficulty, but often cannot meet the requirements of production.

According to the features of complex structure of electric submersible screw pump, the mechanical analysis of any part of unit has been made in the literature [1]. General method for calculating a variety of lift pump passing capacity is put forward in the literature [2], in which researchers calculate the allowed maximum pump length corresponds to expected pump setting depth. The literature [3] describing the passing capacity of drill of horizontal wells, focusing on drilling safety. In this article, in view of the screw pump with simple structure, an axial force is applied to an end of pump, forcing the pump flexural deformation, using the deformation of the pump to make its passing. Do the theoretical analysis of this process and built a corresponding calculation model, according to the calculation results can determined the maximum allowed value of pump depth, in this process, stator rubber structure is weakness. To ensure the pump works, strength check the pump's stator rubber. Case studies show that, the method of squeeze pump enables the pump increase passing capacity, as far as possible to increase the pump depth, and to avoid the rubber stator structure being snapped and degumming phenomenon.

PASSING CAPACITY ISSUE DESCRIPTION

As shown in Fig.1 is a scale diagram for the screw pump at kickoff section in the directional well. During the installation process for a screw pump well, screw pump stator down into the well following the end of tubing. The curvature radius of the section in the diagram is 172m, the dogleg is 10°/30m. The joint between the pump stator and the tubing as shown in the Fig.1(I), you can see that wellbore geometry like a straight line within the scope of the pump length, and that outer diameter of tubing diameter is less than the pump's stator. Tubing has sufficient space to bending deformation with the wellbore geometry, has little impact on the passing capacity of pump. In order to highlight the key issues, present relationship between pump stator and casing, we simplified the problem as shown in Fig.2 passing capacity simplified model.

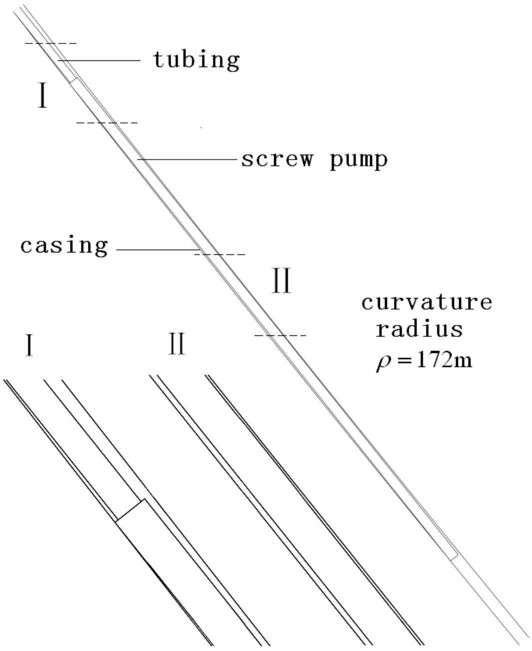

Fig.1 Scale diagram of screw pump in directional well

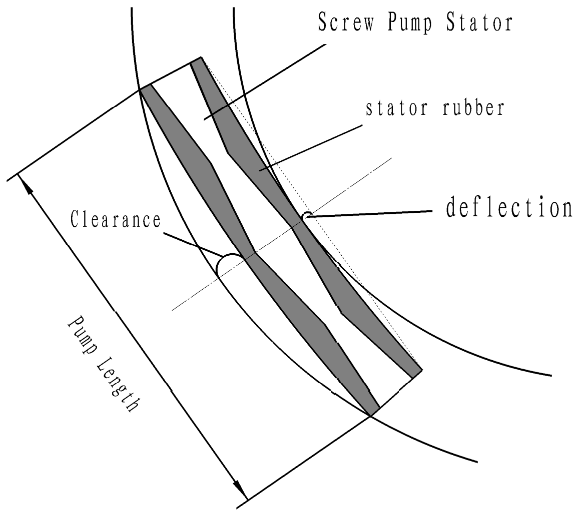

Fig.2 Passing capacity simplified model

Assuming the wellbore trajectory is a smooth space curve. Handling the wellbore geometry data by cubic spline interpolation method, we can obtain the deviation angle at any depth. Selecting the appropriate section length, depending on each section deviation angle and well depth of the beginning and end, we can achieve the curvature radius of the section as

$$R = \frac{360\Delta H}{2\pi\Delta\alpha}. \quad (1)$$

The clearance between pump and casing

When the pump go through the kickoff section, if omit the deformation of pump effects, the maximum clearance between pump and casing is located in the middle of pump. The maximum allowable value of this clearance is the diameter difference between the pump and casing. The model is further simplified to a new one as shown in Fig.3, in this model pump edge sliding on the casing cavity edge, we can work out the maximum clearance between the pump and the casing wall based on geometric relation. Wherein ρ for the curvature radius of the casing cavity edge, τ for the maximum clearance between pump and casing, L for pump length.

If the curvature radius of the section and the length of the selected pump are already known, the maximum clearance between the pump and casing could be calculated by the Pythagorean Theorem as

$$\tau = \rho - \sqrt{\rho^2 - \frac{L^2}{4}}, \quad (2)$$

Where

$$\rho = R + \frac{d_c}{2}.$$

The simple beam model of pump stator

Normally，the dogleg of kickoff section in around 10°/30m. By the results of some calculation can be concluded that even without considering the bending deformation of the pump, pump can also be passed in this case. However, in the actual drilling, wellbore trajectory inevitably would appear several larger doglegs, sometimes deviated from preset paths. For such cases, we attempt to draw support from deformation of the pump to make the pump successfully passed. So a simply supported beam model is built for the pump stator as shown in Fig.4. Wherein q for the uniform load from pump stator weight, α for the angle between pump and the vertical direction.

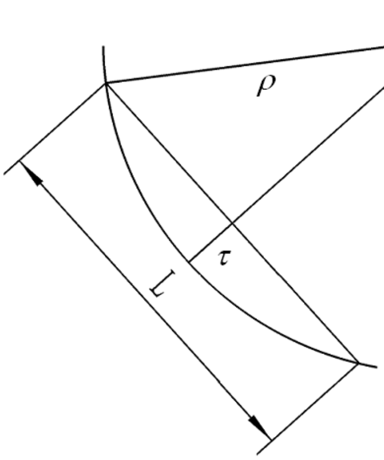

Fig.3 Clearance between pump and casing

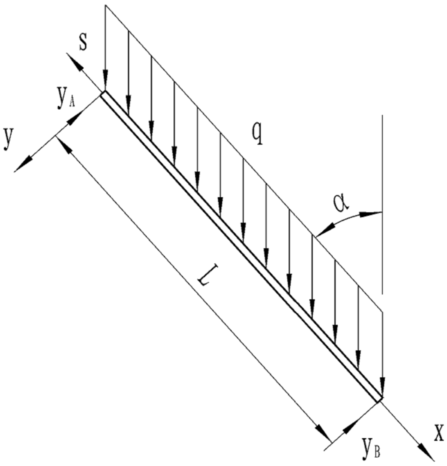

Fig.4 Simple beam model of pump stator

In the simply supported beam model, the deflections of pump stator under gravity could be calculated,

$$\omega_G(x) = \frac{qx\sin\alpha}{24EI_Z}\left(L^3 - 2Lx^2 + x^3\right). \quad (3)$$

Maximum clearance between pump and casing is

$$\tau_1 = \tau - \omega_G\left(\frac{L}{2}\right).$$

Set casing inner diameter is d_c , pump outer diameter is Q. While $\tau_1 < d_c - D_p$, pump can through the current section. While $\tau_1 \geq d_c - D_p$, pump cannot through easily.

When dogleg reaches a certain value, or the deflection of pump which under the influence of gravity is small, pump still cannot through the section. Due to the length of the pump has a great influence on the passing capacity, so when larger dogleg wells are encountered in actual production, generally use short pump. But the shorter the pump length, the less the series of pump, lifting capacity goes down at the same time, so the method, instead of the pump with a shorter one, does not meet the actual needs of production. In view of this problem, this article put forward imposing an axial load on the pump end, attempts extrusion pump to make it go into the large dogleg section until the through on the premise of do not damage the structure of the pump stator. The specific methods are discussed in the next section.

THEORETICAL ANALYSIS AND MECHANICS MODEL OF SQUEEZE PUMP METHOD

When the pump is stuck in the casing, as the tubing continues to decline, tubing weight is gradually loaded on the end A of pump, at that time the counterforce from casing will be produced and loaded on the pump.

In the front of calculation, the curvature radius of the section being researched is assumed as a constant, while the actual curvature radius of the section is changing. As shown in Fig.5, when the curvature radius from ρ_B reduced to ρ_L , if the pump to continue down, the pump will be subject to concentrated load N_c from casing, so that the pump further flexure deformation occurred. When the deflection reaches a certain range, the pump can continue down.

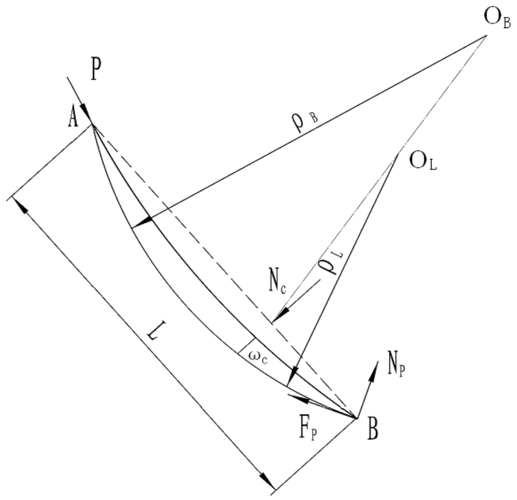

Fig.5 Mechanics model of squeeze pump method

Because of the contour of pump and casing is tangent, and the action point is the tangent point, thus the equation that describes the relation between required deflection at action point and action point location as

$$\omega_n\left(x_B\right)=\rho_L-\sqrt{\rho_L^2-x_B^2}-d_c+D_p .$$

Depending on the required deflection at action point and the action point location, combined with Eq.3, the concentrated load N_c of casing exerted on the pump could be deduced,

$$N_c\left(x_B\right)=\frac{qLx_B\left(x_B^3-2Lx_B^2+L^3\right)\sin\alpha-24EI_ZL\left(\rho_L-\sqrt{\rho_L^2-x_B^2}-d_c+D_p\right)}{8x_B^4-32Lx_B^3+44L^2x_B^2-24L^3x_B+4L^4} \tag{4}$$

When the distance of action point and pump end B is L_B, the bending moment of any point on pump is

$$M(x)=\begin{cases}\dfrac{N_cL_B}{L}\left(L-x\right)+\dfrac{qx\left(L-x\right)\sin\alpha}{2} & (0\le x\le L_B)\\ \dfrac{N_c\left(L-L_B\right)x}{L}+\dfrac{qx\left(L-x\right)\sin\alpha}{2} & (L_B\le x\le L)\end{cases}, \tag{5}$$

wherein $M(x)$ consists of two parts of bending moment, the one comes from gravity, another is caused by the concentrated force from casing. Wherein x is the distance of a point on the pump and the pump end B .

In a certain position, the maximum value of axial force being loaded on the pump end A is represented as

$$P_{\max}(H)=\rho_t gA_t\int_0^H\cos\alpha(h)\mathrm{d}h-F_S , \tag{6}$$

In which the axial force P comes from tubing gravity. While the pump due to the larger dogleg stuck in the casing, as the tubing down, the axial fore on pump end A increased gradually, when the deflection at the action point increases to a certain value, pump can get through current section.

In the calculation should confirm whether self-lock behavior of pump in the casing occurs. According to the selected material of casing and pump, the friction factor f can be determined, the condition of the self-lock occur for

$$\tan\varphi\le f$$

$$\varphi = \arccos\frac{x_B}{\rho_L} + \theta_B$$

When the self-lock condition is satisfied, the pump cannot through the current section by the squeeze pump method.

THE STRENGTH CONDITION OF PUMP STATOR

The stator cylinder strength

Due to the pump in the process of passing produced a larger bending deformation, so build strength condition of the pump cylinder as

$$\sigma_{\max} = \frac{M_{\max}}{W} \le [\sigma],$$

In which the parameter $M_{\max}$ can be evaluated according to the Eq.5.

The pump stator rubber strength

Should notice is that the structure of the screw pump stator is vulcanization bonding chemigum bushing within the cylinder jacket, so pump bends must have an impact on the structure of rubber. Snap and degumming is main failure form of rubber, while the tensile strength, tear strength and the bonding strength of rubber to be much less than the pump cylinder strength, so established the strength condition of the stator rubber as

$$\sigma_{r\max} = \frac{M_{\max} d_p}{I_Z} \le [\sigma_r], \tag{7}$$

Different manufacturers and different pump types corresponding with different technical parameters, the value of σ_r should choose the smaller of bonding strength and tensile strength, in addition to selecting the appropriate safety factor.

Comparing two strength conditions, in the process of pump through the kickoff section, the strength condition of the pump stator rubber is stricter than the pump cylinder strength conditions, and that stator rubber intact or not determines the screw pump works correctly, so pump stator rubber strength condition is the important gist in strength check.

PRACTICAL CASE STUDIES

Du66-54-Ping40 well as a case, its wellbore geometry data is shown in TABLE 1. In this case, the inner diameter of casing d_c =123.7mm, the length of the pump L =7m, the outer diameter of the pump D_p =89mm, the inner diameter of the pump cylinder d_p =79mm, the weight of the pump G_p =76kg, and the elastic modulus of the pump cylinder $E = 1.4\times10^{11}$ Pa.

TABLE 1 Wellbore geometry data of Du66-54-Ping40 well

depth(m)	deviation angle (°)	azimuth (°)
0	0	0
530	0.54	89.7
550	3.49	34.35
575	12.71	42.73
600	21.1	46.6
625	27.79	47.76
650	31.62	51.06
675	36.33	49.95
700	39.44	48.63
725	45.29	48.15

First of all, analyze the wellbore geometry data, the minimum curvature radius of the wellbore trajectory is 155.36m in the depth of 366.7 m, and calculated that

$$\tau_1 = 0.039122\text{m} \ge d_c - D_p,$$

which implies that merely under gravity the pump cannot through this section.

Then calculations this case by squeeze pump method, achieved the maximum bending moment

$M_{\max}$ =186.437N.m and,

The contact angle between the pump end B and casing

φ =88.711395°,

which implies that the self-lock does not occur and pump could through this section by squeeze method. the maximum tensile stress on the pump stator $\sigma_{r\max}$ =12.61MPa.

Pump stator rubber parameters as shown in TABLE 2. Because of technical conditions and the production process is different, the parameters of different pump also each are not identical. So in this case, some pump with lower Related parameters whose stator rubber snap and degumming phenomenon may occur.

TABLE 2 the reference range of pump stator rubber main parameters

tensile strength (MPa)	tear strength (MPa)	bonding strength (MPa)	Bonding interface adherence ratio (%)
≥ 12	35-50	≥ 12	≥ 70

RESULTS AND DISCUSSION

Using the method of squeeze pump can effectively improve the passing capacity of screw pump in directional well. During the design process, according to the squeeze pump model results can determined that the pump whether can down to the expected pump setting depth.

Bending has larger effect on the structure of the screw pump rubber stator, through the calculation of squeeze pump model to ensure that the rubber structure without being snapped and degumming, But the effect of local bending on the rubber stator working life still needs further study.

NOMENCLATURE

A_t = Tubing cross sectional area, m^2
d_c = Casing inner diameter, m
d_p = Inner diameter of the pump cylinder, m
E = Elastic modulus, GPa
F_S = The friction between the tubing and casing in the process of tubing falling, N
g = Acceleration of gravity, m/s^2
H = Pump current depth, the tubing length, m
I_Z = Second polar moment of area, m^4
L = Screw pump length, m
$M_{\max}$ = Maximum bending moment on the pump, N.m
q = The uniform load which derived from the gravity of pump stator, N/m
R = The curvature radius of the section, m
W = Section modulus, dimensionless
x = The distance between a particular point and the end B on pump stator, m
x_B = Distance between action point and pump end B, $x'_B \leq x_B \leq \frac{L}{2}$, m
α = The angle between pump and the vertical direction, °
$\alpha(h)$ = The deviation angle of the h depth in the wellbore, °
$\Delta\alpha$ = The absolute value of the difference between two ends' deviation angle of the section,°

ΔH = The difference value between two ends' depth of the section, m
θ_B = The angle caused by bending deformation at pump end B, °
ρ = The curvature radius of the edge for casing cavity, m
ρ_t = Tubing density, kg/m^3
$[\sigma]$ = Stress of allowance of pump cylinder, MPa
$[\sigma_r]$ = Allowable stress of the pump stator rubber, MPa
$\sigma_{r\max}$ = Maximum tensile stress of the pump cylinder inside surface , MPa
τ = Maximum clearance between the pump and casing, m
φ = Contact angle between the pump end B and casing, °
$\omega_G(x)$ = Deflections of x position on pump stator, m

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