STUDY OF THE STRUCTURAL BEHAVIOUR AND STABILITY OF DRILL STRINGS

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Abstract: This paper presents a systematic overview of drilling operations and a mechanical analysis of drill string behaviour under operational loads. Drilling activities are first classified according to their purpose, operating mode, energy used for rock disintegration, and the mechanism of action on the rock, with particular emphasis on rotary-hydraulic drilling systems. The main components of drilling installations, both surface and downhole, are also outlined. The second part of the study focuses on the stress and deformation state of the drill string, considering static loads generated by self-weight and torsion, as well as dynamic loads arising during handling operations, braking, and drilling disturbances. Analytical expressions are presented for normal stresses and elongations of the drill string under various working conditions. Finally, the buckling and stability of the compressed section of the drill string are analyzed using classical Euler buckling theory and the method of minimum potential energy. Critical loads and lengths leading to instability are derived, providing essential criteria for safe design and operation of drilling strings.

Keywords: Drill string, stress and deformation, buckling stability, rotary drilling.

1. INTRODUCTION

Drilling operations can be classified according to several criteria, with the purpose of drilling being one of the most important. From this standpoint, drilling activities include geological investigation drilling, production drilling, and special-purpose drilling. Geological investigation drilling comprises reference, prospecting, and exploration drilling, all aimed at acquiring progressively more detailed geological information for regional studies, deposit characterization, and reserve estimation. Production drilling is associated with resource exploitation and reservoir management, including wells used for pressure maintenance and monitoring during operation.

In addition, drilling operations may be distinguished by their application, operating mode, and the type of energy used for rock disintegration. Special-purpose drilling covers mining, hydrogeological, geotechnical, seismic, and large-diameter

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drilling works for engineering and industrial purposes. From an operational perspective, drilling may be manual or mechanical, while in terms of rock-breaking mechanisms, drilling methods are classified as mechanical, hydromechanical, or thermal, depending on the dominant energy source involved in the drilling process.

According to the mode of action exerted on the rock, drilling operations are mainly classified as percussive or rotary. In percussive drilling, rock fragmentation at the borehole bottom is achieved by repeated impacts of a chisel-shaped bit, or trepan, which may operate in dry conditions using rods or cables, with cuttings periodically removed by introducing water and extracting the resulting slurry, or in hydraulic conditions where the bit is actuated through hollow steel drill rods and fluid is continuously pumped to clean the borehole bottom.

Rotary drilling, by contrast, relies on the continuous rotation of the drilling bit, with rotary-hydraulic drilling being the most common method, in which the fragmented rock is continuously transported to the surface by circulating drilling fluid, ensuring uninterrupted drilling and efficient cuttings removal (Figure 1).

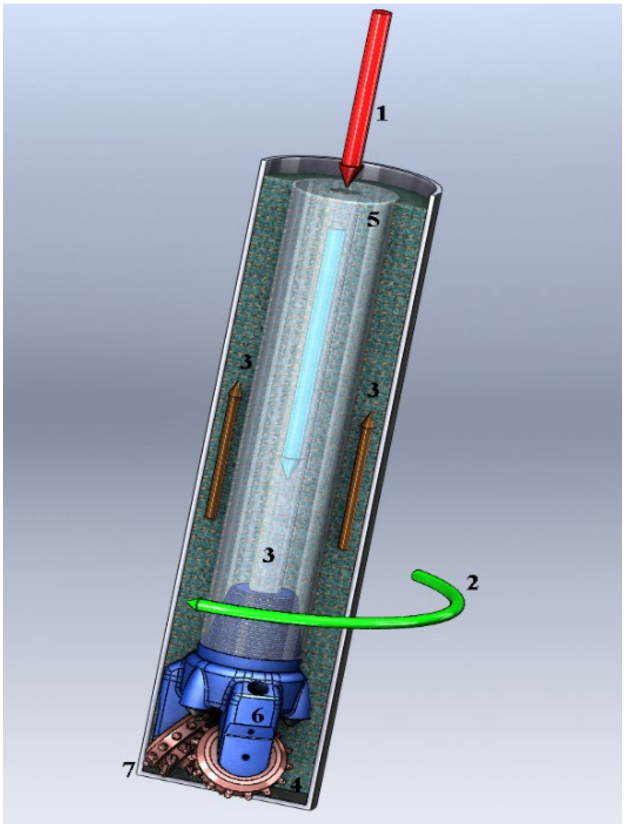

Fig. 1. Operating principle of the drilling bit
(1 – axial load applied to the bit; 2 – rotation of the bit; 3 – circulating fluid; 4 – fragmented rock; 5 – drill rod; 6 – drilling bit; 7 – borehole bottom)

Rock disintegration in rotary drilling is achieved by means of drilling bits that combine rotational motion with axial penetration into the rock formation. The rotation of the bit is provided by the drilling installation motors through the rotary table or top drive and transmitted along the drill string, while the axial load required for penetration is generated by the weight of the drill string itself. Drilling fluid is pumped through the interior of the drill rods to the borehole bottom, where it assists in cooling the bit and transporting fragmented rock to the surface. Depending on the configuration, rotary drilling can be carried out using surface-driven rotation systems, downhole motors, or combined rotary-percussive methods.

Regardless of the drilling method employed, all drilling installations are composed of common basic elements, which are conventionally divided into surface and downhole components. The surface equipment includes the power unit, derrick and substructure, hoisting system for handling the drill string, rotary system for transmitting rotational motion, and the circulation system responsible for drilling fluid flow. The downhole assembly comprises the casing string, the drill string, and the rock disintegration tools, which together ensure the structural integrity of the well and the effective execution of the drilling process.

2. STRESS AND DEFORMATION ANALYSIS OF THE DRILL STRING

During the drilling or mechanical coring process, the drill string is subjected to the action of a complex system of static and dynamic loads.

The static loads are generated by:

- the self-weight of the drill string;
- torsional moments.

The dynamic loads may occur in the following situations:

- during sticking or jamming of the drill string;
- during running-in operations (braking) and during pulling out of the drill string (release from slips or elevators);
- due to variations in the velocity of the drill string;
- during bending of the drill string while rotating, when it loses its straight configuration as a result of transverse oscillations caused by the operation of the cutting tools at the borehole bottom.

The self-weight of the drill pipes, in the upper portion of the drill string (which is subjected to tension), generates normal tensile stresses. Due to the axial load applied to the cutting tool by a portion of the self-weight of the drill pipes, the lower part of the drill string is subjected to compression. As a result, during drilling, normal compressive stresses occur in this region (Figure 2).

The following parameters are used:

- L - total length of the drill string [m];
- l - length of the heavy-weight drill pipes [m];
- σ_1 - normal tensile stress $\left[\mathrm{N/m^2}\right]$;

- A - cross-sectional area of the drill pipes $\left[m^2\right]$;
- A_1 - cross-sectional area of the heavy-weight drill pipes $\left[m^2\right]$;
- ρ_0 - density of steel $\left[kg/m^3\right]$;
- ρ_n - density of the drilling fluid $\left[kg/m^3\right]$;
- l_0 - compressed length of the heavy-weight drill pipes $[m]$;
- k - coefficient accounting for the increase in stress due to special tool joints;
- q_1 - mass per unit length of the drill pipe $[kg/m]$;
- q_2 - mass per unit length of the heavy-weight drill pipe $[kg/m]$;
- a - acceleration of motion $\left[m/s^2\right]$;
- g - gravitational acceleration $\left[m/s^2\right]$.

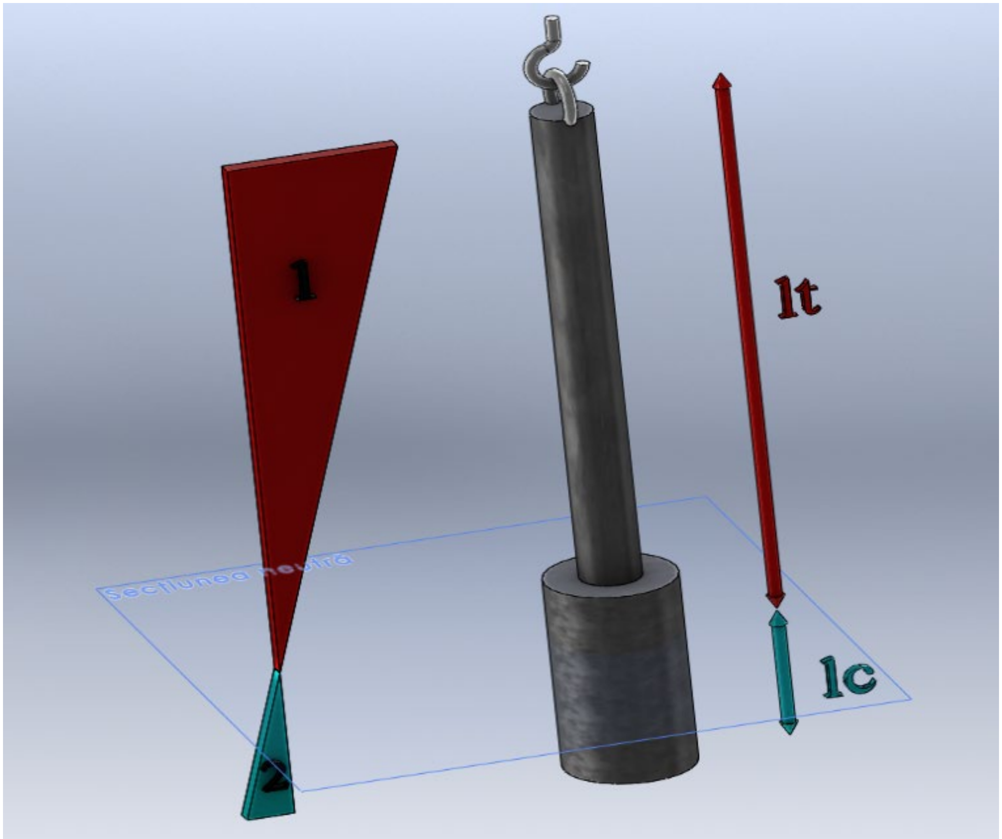

Fig. 2. Characteristic lengths of axial stresses in the drill string during drilling: (1 – tensile stresses; 2 – compressive stresses; l_t - length of the drill string subjected to tensile stresses; l_c - length of the drill string subjected to compressive stresses)

The maximum normal unit stress, in the uppermost cross-section of the drill string, when the drilling bit is lifted off the borehole bottom, assuming a constant cross-sectional area along the entire length of the drill string, is given by:

$$\sigma_1 = \frac{LA(\rho_0 - \rho_n)g}{A} = L \cdot g(\gamma_0 - \gamma_n), \frac{N}{m^2} \tag{1}$$

If the influence of the weight of the tool joints or couplings is also taken into account, then:

$$\sigma_1 = k \cdot L \cdot g(\rho_0 - \rho_n), \frac{N}{m^2}. \tag{2}$$

where $k = 1.05 - 11$, depending on the type of connection and the length of the couplings or tool joints.

If, however, the influence of special tool joints and the increased cross-sectional area of the heavy-weight drill pipes is considered, the normal unit stress is given by:

$$\sigma_1 = k \cdot g(L - l + l\frac{q_2}{q_1})(\rho_0 - \rho_n), \frac{N}{m^2} \tag{3}$$

During drilling with the bit on bottom, when a reduction of this stress is considered due to the fact that (only) a weight corresponding to the length of the heavy-weight drill pipes acting in compression is applied to the bit, the normal unit stress is:

$$\sigma_{1f} = k \cdot g[L - l + \frac{q_2}{q_1}(l - l_0)](\rho_0 - \rho_n), \frac{N}{m^2} \tag{4}$$

During handling operations, when variation in the velocity of moving masses occur, deceleration during running in of the drill pipes and sudden acceleration during pulling out while releasing from slips or elevators, the unit stresses are calculated using the relation:

$$\sigma_{1m} = \sigma_1(1 + \frac{a}{g} \pm \varphi), \frac{N}{m^2} \tag{5}$$

where σ_1 has the value given by relation (2) and φ represents a coefficient accounting for the friction of the drill string against the borehole walls (with the "+" sign during pulling out and the "-" sign during running in). The value of this coefficient depends on the inclination of the borehole and, for vertical boreholes, has a value of 0.2.

The dynamic loads that occur when lifting the drill string from the slips or from the elevators are relatively small, since the starting accelerations themselves are low $\left(a_p = 0.1 - 0.3 \quad m/s^2\right)$.

However, the dynamic loads that arise during braking while running in the drill string have significantly higher values, especially in the case of sudden braking ("short stop"), when the stopping acceleration a_0 is high. This acceleration is calculated using the relation:

$$a_0 = \frac{v_i}{t_f}, \frac{m}{s^2} \tag{6}$$

where v_i is the running-in velocity of the drill string, in m/s and t_f is the braking time, in s.

The absolute elongation of the drill string under the influence of its own weight is calculated using the relation:

$$\Delta l = \frac{G \cdot L}{2E \cdot A}, m \tag{7}$$

where G, in N, is the weight of the drill string, and E, in N/m^2, is the longitudinal modulus of elasticity. The weight G can be expressed as:

$$G = k \cdot A \cdot L \cdot g(\rho_0 - \rho_n), N \tag{8}$$

thus obtaining:

$$\Delta l = \frac{k \cdot L^2 \cdot g(\rho_0 - \rho_n)}{2E}, m. \tag{9}$$

During the transmission of a part of the drill pipe weight to the borehole bottom, the elongation is calculated using the relation:

$$\Delta l_1 = \frac{k \cdot g(L-l)^2 (\rho_0 - \rho_n)}{2E}, m \tag{10}$$

Taking into account the elongation of the drill string also under the action of its own weight, the following relation is obtained:

$$\Delta l_2 = k \cdot g \frac{\rho_0 - \rho_n}{2E}[(L-l)^2 + 2\frac{A}{A_1} l(L-l)], m \tag{11}$$

A simpler formula that can be used to calculate the elongation of drill strings due to their own weight, suspended in fluids with different densities, is:

$$\Delta l = \frac{L^2}{2E}[\gamma_0 - 2\gamma_n(1-\mu)], m \tag{12}$$

where μ is Poisson's ratio, which for steel has a value of 0.28. For steel, the longitudinal modulus of elasticity is $2.1 \cdot 10^5$ MPa.

3. BUCKLING AND STABILITY OF THE COMPRESSED SECTION OF THE DRILL STRING

3.1. For a section of the drill string

According to Figure 3, the AB segment of the compressed region of the drill string is subjected to the weight of the AM portion located above it, as well as to its own self-weight corresponding to the considered AB segment.

Assuming that during bending the ends of the AB segment remain on the borehole axis and considering them as simply supported, the problem reduces to the classical Euler buckling case, neglecting the forces resulting from the self-weight of the AB segment.

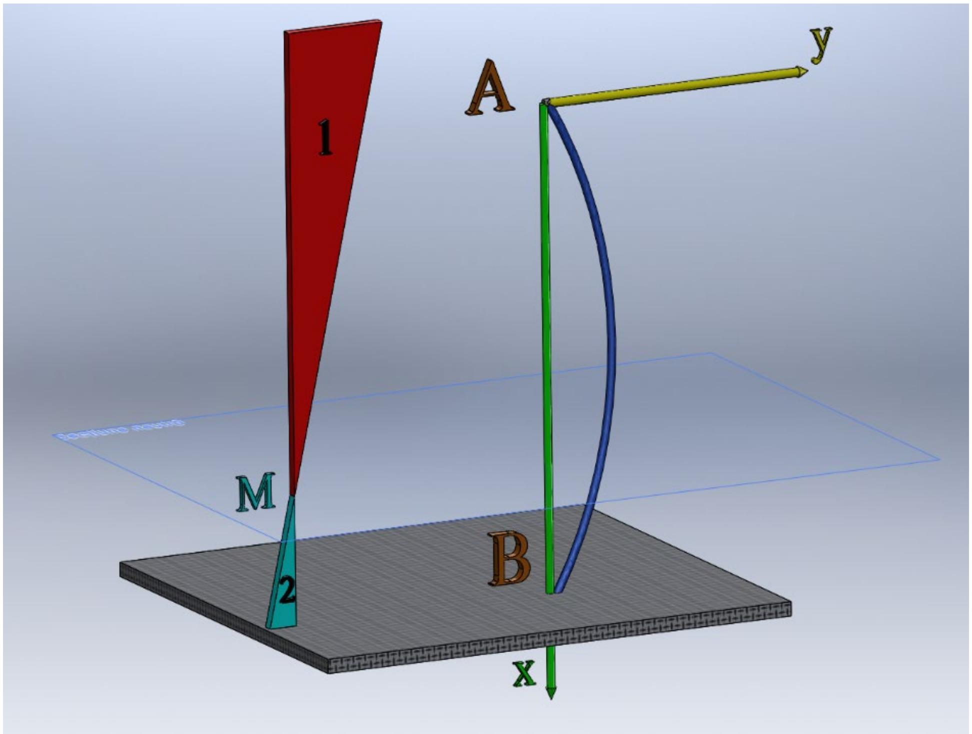

Fig. 3. Buckling of a compressed section of the drill string

By adopting the coordinate axes as shown, the equilibrium condition for this deviated position of the bar can be written, namely the differential equation of the bent axis of the bar, also referred to as the equation of the deformed neutral fibre:

$$\frac{d^2y}{dx^2} = -\frac{M}{EI_z}, \frac{1}{m} \tag{13}$$

where:

M - is the bending moment $[\mathrm{Nm}]$;

I_z - is the moment of inertia of the bar $[\mathrm{m}^4]$;

EI_z - represents the flexural rigidity of the bar.

For an arbitrary cross-section of the bar, defined by the coordinate x, the bending moment is given by:

$$M = P y \text{ , } Nm \tag{14}$$

where P, in N, is the axial force acting on the bar.

By substituting relation (14) into relation (13) and introducing the notation:

$$\alpha^2 = \frac{P}{EI_z} \tag{15}$$

the following differential equation is obtained:

$$\frac{d^2 y}{dx^2} + \alpha^2 y = 0 \tag{16}$$

The equation is linear and can be solved by assuming a solution of the form $y = D \cdot e^{kx}$ where k is a constant coefficient to be determined. Substituting this solution into equation (16) leads to the characteristic equation, whose roots are $k = \pm i\alpha$. Consequently, the general solution of the differential equation can be expressed as a combination of exponential functions.

Using the relations $e^{i\alpha x} = \cos \alpha x + i \sin \alpha x$, $e^{-i\alpha x} = \cos \alpha x - i \sin \alpha x$, the general solution becomes:

$$y = C_1 \cos\alpha x + C_2 \sin\alpha x \tag{17}$$

where C_1 and C_2 are two arbitrary constants to be determined from the boundary conditions.

However, the apparently larger number of boundary conditions compared to the number of unknowns is only apparent, since it can be easily observed that for the conditions at the both ends, namely for $x = 0$ and $x = 1$, the deflection y is equal to zero, leading to the same equations with the unknowns C_1 and C_2. These equations, obtained by imposing the boundary conditions on relation 17, take the form: $C_1=0$; $C_1 \cos\alpha l + C_2 \sin\alpha l = 0$.

3.2. For the entire length of the drill string

In this case, the drill string, shown in Figure 4, is considered as a bar subjected to the forces generated by its own weight, simply supported at the upper end on the section subjected to tension.

The method of variation of potential energy is employed. Under the action of compressive forces, compression energy is stored in the undeformed bar; therefore, the potential energy W_1 of the system with the bar still undeformed is equal to this compression energy, namely $W_1=W_{com}$.

As the compressive forces increase up to the critical value, the bar begins to bend, and the bending energy W_{inc} is added to the compression energy W_{com}. During this process, the compressive forces perform positive mechanical work L along the displacement paths of their points of application, which move downward during the bending of the bar. Consequently, in the deformed state, the potential energy W_2 of the bar becomes:

$$W_2 = W_{com} + W_{înc} - L \,, \; J \tag{18}$$

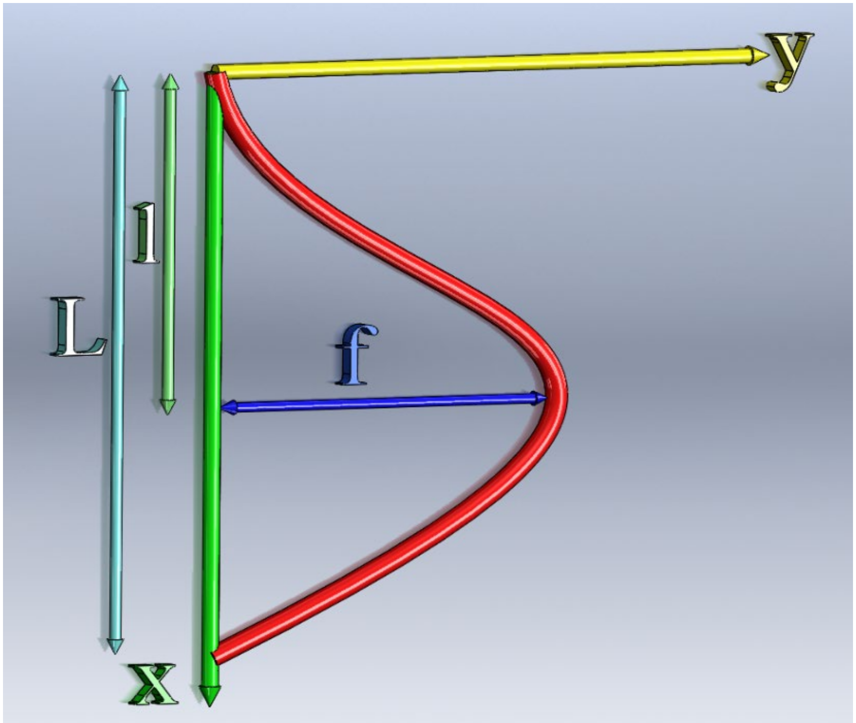

Fig. 4. Buckling of the drill string at the upper end

According to the principle of minimum potential energy, if $W_1 < W_2$, the stable configuration is the straight one, whereas if $W_1 > W_2$, the stable configuration is the deformed one.

The critical load at which the transition from the straight configuration to the deformed configuration becomes possible is determined from the condition $W_1 = W_2$, or, in the present case, $W_{inc} = L$. This means that the mechanical work performed by the external forces during the deformation of the bar, in absolute value, must be equal to the bending energy.

The potential energy method is approximate, because its application requires consideration of the equation of the deformed neutral fibre of the bar, while satisfying the boundary conditions. Indeed, in order to determine the bending energy W_{inc} using the relation:

$$W_{inc} = \frac{EI}{2}\int\left(\frac{d^2y}{dx^2}\right)^2 dx \text{ , } J \tag{19}$$

it is necessary to know the analytical expression of the deformed neutral fibre. The same information is required for the determination of the mechanical work L of the compressive forces, given by:

$$L = \frac{1}{2}p\int x\left(\frac{dy}{dx}\right)^2 dx \text{ , } J \tag{20}$$

In this relation, p, in N/m, represents the uniformly distributed forces per unit length.

Assuming that the compressive forces due to the self-weight reach their critical values and that the bar undergoes a slight deformation, the deformation shapes are selected by choosing the coordinate axes as shown in Figure 4.

By dividing the total length L of the bar into two portions of lengths l and $L-l$ the equations of the deformed neutral fibre are written for each portion. These equations must satisfy the following boundary conditions:

- for the portion of length l:
  - $x = 0; \quad y = 0$;
  - $x = l; \quad y = f; \quad y' = 0; \quad (y'' = 0)$;
- for the portion of length $L-l$:
  - $x = l; \quad y = f; \quad y' = 0; \quad (y'' = 0)$;
  - $x = L; \quad y = 0$

All these conditions are satisfied if the equation of the deformed neutral fibre (the deformed axis) has the following form:

- for the portion of the length l:

$$y_1 = \frac{f}{2}(1 - \cos\frac{\pi x}{l}) \tag{21}$$

- for the portion of length $L-l$:

$$y_2 = f \sin\frac{\pi(x - 2l + L)}{2(L - l)} \tag{22}$$

The bending energy W_{inc} and the mechanical work L are determined by successively performing the calculations for the first time and the second region.

- for the portion of length l, and considering that, after evaluating the integral, the following expression is obtained:

$$(W_{inc})_1 = \frac{EI}{2}\int_0^l (y_1'')^2\,dx = \frac{\pi^4 f^2 EI}{8l^4}\int_0^l \cos^2\frac{\pi x}{l}\,dx\,,$$

and taking into account that $\cos^2\frac{\pi x}{l} = \frac{1}{2}(1+\cos\frac{2\pi x}{l})$, after evaluating the integral, the following result is obtained:

$$(W_{inc})_1 = \frac{\pi^4 f^2 EI}{16l^3}\,, J \tag{23}$$

- for the portion of length $L\text{-}l$:

$$(W_{inc})_2 = \frac{EI}{2}\int_l^L (y_2'')^2\,dx = \frac{\pi^4 f^2 EI}{32(L-l)^4}\int_l^L \sin^2\frac{\pi(x-2l+L)}{2(L-l)}\,dx\,,$$

and taking account that $\sin^2\frac{\pi(x-2l+L)}{2(L-l)} = \frac{1}{2}[1-\cos\frac{\pi(x-2l+L)}{L-l}]$, after integration, the following results is obtained:

$$(W_{inc})_2 = \frac{\pi^4 f^2 EI}{64(L-l)^3}\,, J \tag{24}$$

The mechanical work L_1 of the compressive forces acting on the portion of length l is $L_1 = \frac{1}{2}p\int_0^l x(y_1')^2\,dx = \frac{\pi^2 f^2 p}{8l^2}\int_0^l x\sin^2\frac{\pi x}{l}\,dx$, from which, by substituting $\sin^2\frac{\pi x}{l} = \frac{1}{2}(1-\cos\frac{2\pi x}{l})$ and by evaluating the integral, the following results is obtained:

$$L_1 = \frac{\pi^2 f^2 p}{32}\,, J \tag{25}$$

Similarly, for the portion of the length $L\text{-}l$,

$$L_2 = \frac{1}{2}p\int_l^L x(y_2')^2\,dx = \frac{\pi^2 f^2 p}{8(L-l)^3}\int_l^L x\sin^2\frac{\pi(x-2l+L)}{2(L-l)}\,dx\,,$$

and finally,

$$L_2 = \frac{\pi^2 f^2 p}{16(L-l)^2}\left[\frac{L^2 - l^2}{2} + \frac{(L-l)^2}{\pi^2}\right], J \tag{26}$$

In accordance with relation (18), the loads at which the transition from the straight configuration to the deformed configuration becomes possible, namely the critical loads, are determined from the condition:

$$(W_{inc})_1 + (W_{inc})_2 = L_1 + L_2 \tag{27}$$

By substituting the values of W and L, the following equation is obtained:

$$\pi^2 EI\left[\frac{1}{l^3} + \frac{1}{4(L-l)^3}\right] = p\left[\frac{1}{2} + \frac{1}{(L-l)^2}\left[\frac{L^2 - l^2}{2} + \frac{2(L-l)}{2}\right]\right] \tag{28}$$

In order to determine the critical load $(pL)_{cr}$ or the critical length L_{cr}, it is necessary to eliminate the quantity l from equation (28). An approximate relationship between L and l is sought using the general Euler formula:

$$P_{cr} = \frac{\pi^2 EI}{(\mu L)^2}, N, \tag{29}$$

where the coefficient μ depends on the boundary conditions at the end of the bar and is equal to 1 for the case where both ends are simply supported, and equal to $\sqrt{0.5} = 0,7$ for the case where one end is simply supported and the other is fixed.

The coefficient μ expressed numerically, is equal to the ratio of the length of this portion to the total length of the bar. In the present case, this portion (or, more precisely, half of this portion), which deforms as a bar simply supported at both ends, has the length L-l. Assuming $\mu = 0.7$ the relation $2(L-l) = 0.7 - L$ is obtained, from which it follows that $l = 0.65L$.

By substituting this value of l into equation (28), the calculation expression for the critical weight of the compressed portion of the drill string is obtained:

$$(pL)_{cr} = \frac{3,2\pi^2 EI}{L^2}, N. \tag{30}$$

CONCLUSIONS

This paper presents an analytical approach to the calculation and dimensioning of the drill string, with emphasis on the stress state, deformation behaviour and stability conditions occurring during drilling operations. The analysis highlights that the drill string is subjected to a complex combination of static and dynamic loads, generated by its own weight, axial forces, torsional moments and centrifugal effects associated with rotary motion.

The conditions leading to the appearance of compressive stresses and buckling in the lower section of the drill string were established, and critical loads were determined using both classical stability theory and the potential energy method. The results show that the stability of the compressed section strongly depends on the length of the drill string, boundary conditions and elastic properties of the drill pipes.

Dynamic stability under the action of centrifugal forces was analysed by modelling the drill string as a slender rotating rod. Critical lengths, angular velocities and rotational speeds were obtained, defining the conditions under which dynamic buckling may occur. These results underline the importance of controlling rotational speed, especially for long drill strings.

The determination of bending stresses demonstrated that maximum stresses occur at the midpoint of the deformed half-sine wave and depend on pipe diameter, borehole clearance and deformation amplitude. The use of heavy-weight drill pipes in the lower section of the drill string, as well as protective elements, was shown to be effective in reducing buckling deflection and bending stresses.

Overall, the analytical relations derived in this study provide a useful basis for the proper design and operation of drill strings, contributing to improved stability, reduced mechanical failures and safer drilling conditions.

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