Interaction Between the Pipeline and Additional Equipment for Trenchless Technologies

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Abstract. In this article the authors defined the limits of applicability of the pipeline pusher as additional equipment for pipeline construction when using trenchless methods. In this case, the pushing force is applied to the free end of the pipeline section located on the day surface. The authors obtained analytical dependences for determining the stress-strain state of the pushed pipeline when using the pusher in a particular case. In addition, they identified values of axial pushing force that are considered dangerous, because they can cause pipeline dropping down from the roller supports during pipeline pullback.

1. Introduction

Such pipeline construction scheme that provides pipe displacement into the borehole or horizontal well by means of a drilling rig or other power unit is relevant for trenchless technologies in general. It involves the application of forces to the pipeline for its movement within the wellbore with the aim of placing it in its operating position. Note that according to the classical scheme the force applied to the pipeline is pulling [2]. Nevertheless, as additional equipment, pipeline pushers may be used to set pipeline in motion by a wide range of forces [5]. Moreover, the force of the pusher can be applied to different points of the pipe. This dramatically affects the general picture of stress distribution in the pipe material [1]. The article attempts at studying interaction between the pushed pipeline and the pusher in some special cases.

Research on the interacting pipeline and pushing equipment implies finding correlation between the force of a pusher and the stress appearing in a spatially curved pipeline. It is also necessary to take into account the primary stress in the wall of the pipeline from elastic bending, so the problem has a specific solution only in a particular case that implies a certain disposition of roller supports, a range of pipe diameters and radiuses of elastic bending, as well as a ratio of different pipeline sections lengths.

Due to the fact that the pipeline section already situated in the well at the moment is limited in its displacements by the well walls, we are most interested in analyzing the behavior of the pipeline section that is laid on roller supports at the entrance to the well. It is this interaction between the pipeline section and the pusher that will limit the possible force that may be applied to the pipe to intensify the pullback process.

The principal possibility to push the pipe by the drilling rig using a reverse thrust can base on specially developed instructions or technological charts. These instructions must be based on calculations of stability of drilling rods of different types when pushing them into wells of different diameters using a variety of drilling rigs. For steel pipelines of large diameters, it is necessary also to keep in mind limits of load on launch way supports when pushing the pipeline out of the borehole in the opposite direction [6]. These limits must be based on appropriate calculations, too.
2. Materials and methods

At the entrance of the well, the pipeline is subject to elastic deformation with the bend radius \( R = 1000 \cdot D \), where \( D \) – the outside diameter of the pipeline (figure 1). The left part of the pipeline section profile on the day surface is illustrated below. The pipeline profile is shown as it should look in real life.

![Figure 1](image.png)

**Figure 1.** The design scheme of the aboveground section of the pipeline during pullback into the well using a trenchless method (left part): \( O \) – the entrance to the well; \( x_{p1}, x_{p2}, x_{pn} \) – coordinates of the supports, \( R \) – the radius of curvature of bent sections; \( L_{z1}, L_{z2}, L_{z3}, L_{z4} \) – lengths of different sections horizontally.

This scheme of elastic bending is most suitable for real steel pipelines. The right part of the design scheme is shown in figure 2. The \( AD \) section will be straight only for plastic pipelines [8].

![Figure 2](image.png)

**Figure 2.** The design scheme of the aboveground section of the pipeline during pullback into the well using a trenchless method (right part): \( \Delta H \) – the level difference between the maximum height and the height of the horizontal section; \( \beta \) - the central angle of the arch of the free section.

The pipeline profile is supported by roller supports \( P_1, P_2 \ldots P_n \) ensuring free movement of the pipeline at a tangent to the pipe axis. We consider that distances between supports \( \Delta P \) are equal. It is desirable that \( H_{max} \) was as small as possible for practical reasons. For this purpose, the bending radius
$R$ should be minimally permissible, that is it should be constant throughout the curved $OC$ section (figures 1, 2).

The height will be measured from the zero level. We will assume that the zero level is the point of the well entrance that begins in the foundation pit. It is recommended to choose the entry angle into the well within $5\text{°} - 20\text{°}$ and it should be kept in mind that an increase in $\alpha$ causes a significant increase in the value of $H_{\text{max}}$. The pipeline is curved convex up in the $OB$ section, it is curved convex down in the $BC$ section and the $CD$ section is straight. In addition, we designate $H_h$ as the height of the horizontal straight section.

Let us determinate the height of supports above the zero level that provides the circular arc shape in sections $OB$ and $BC$. The maximum height would be:

$$H_{\text{max}} = R \cdot (1 - \cos \alpha)$$

As the maximum height should be $H_{\text{max}} \geq H_h$, the following equation must be fulfilled:

$$\alpha \geq \arccos \left(1 - \frac{H_h}{R}\right)$$

In case $H_{\text{max}} = H_h$, the $AD$ section of the pipeline would be straight and horizontal. Probably, it is the best variant but it is possible only for polyethylene pipelines of small diameters and it does not make sense to consider this variant.

The scheme of the pipeline elastic bending that is most suitable for real steel pipelines must be accepted for practical use. This scheme is described in figure 1. Let us find the angle $\beta$ available on the scheme. In the picture, we can see that:

$$\Delta H = 2R \cdot (1 - \cos \beta) = 4R \cdot \sin^2 \frac{\beta}{2}$$

This implies:

$$
\sin^2 \frac{\beta}{2} = \frac{\Delta H}{4R}
$$

$$
\cos^2 \frac{\beta}{2} = 1 - \sin^2 \frac{\beta}{2} = 1 - \frac{\Delta H}{4R}
$$

Therefore:

$$
\sin \beta = 2 \cdot \sin \frac{\beta}{2} \cdot \cos \frac{\beta}{2} = 2 \sqrt{\frac{\Delta H}{4R}} \cdot \sqrt{1 - \frac{\Delta H}{4R}}
$$

$$
\cos \beta = \cos^2 \frac{\beta}{2} - \sin^2 \frac{\beta}{2} = 1 - \frac{\Delta H}{4R} - \frac{\Delta H}{4R} = 1 - \frac{\Delta H}{2R}
$$

Now let us define the length of the pipeline sections:

$$L_{Z1} = R \cdot \sin \alpha$$

$$L_{Z2} = L_{Z3} = R \cdot \sin \beta$$

The equation of the pipeline axis in the $OB$ section in the $xOy$ coordinate system would look like:

$$Y_{OB} = \sqrt{R^2 - \left(x - L_{Z1}\right)^2} - \left(R - H_{\text{max}}\right)$$

For the $BC$ section:

$$Y_{BC} = -\sqrt{R^2 - \left(x - L_z\right)^2} + \left(R + H_h\right)$$

where $L_z$ –the length of the aboveground section of the pipeline without a straight horizontal section. It may be calculated as:

$$L_z = L_{Z1} + L_{Z2} + L_{Z3}$$

Then the height of supports from the zero level will have the following values.

For the $OB$ section:
For the BC section:

\[ H_{Pi} = Y_{OB}(x_i) \]  \hspace{1cm} (13)

For the CD section:

\[ H_{Pi} = Y_{BC}(x_i) \]  \hspace{1cm} (14)

If the shape of the curved axis exactly coincides with the described one, the maximum stress from bending with the bend radius \( R \) would be:

\[ \sigma_R = E \cdot \frac{D}{2R} \]  \hspace{1cm} (16)

where \( E \) – Young’s modulus.

If we have conditions that \( E = 2 \cdot 10^{11} \text{ Pa} \) and \( R = 1000 \cdot D \), we will have:

\[ \sigma_R = \frac{2 \cdot 10^{11} \cdot D}{2 \cdot 1000 \cdot D} = 100 \text{ MPa} \]  \hspace{1cm} (17)

However, pipeline sections sag between supports under the gravitation force. Hence, it appears that stresses can fundamentally differ from the theoretical ones. Obviously, the bigger the distance between supports, the greater the difference. [4].

Let us define the relationship of additional stresses to the distance between supports. If we place roller supports at regular intervals, then we can consider the \( P_iP_{i+1} \) section of the pipeline as a clamped rod loaded with an evenly distributed load \( q \) – its own weight (figure 3).

\[ q \]

\[ \sigma \]

\[ \sigma_R \]

\[ \sigma_{q} \]

\[ \sigma_{cur} \]

**Figure 3.** The design scheme of the section of the pipeline laying on supports under the action of its own weight: \( L_s \) – the distance between supports.

Then maximum bending moment in the pipeline section laid on supports will appear on its ends and will equal [3]:

\[ M_q = \frac{q \cdot L_s^2}{12} \]  \hspace{1cm} (18)

The maximal stress from this moment caused by loading action \( q \), will be:

\[ \sigma_q = \frac{M_q D}{2 \cdot I} = \frac{q \cdot L_s^2 D}{24 \cdot I} \]  \hspace{1cm} (19)

where \( I \) – the moment of inertia of the pipeline cross-section.

At the \( OB \) section, the direction of the moment \( M_{stat} \) coincides with the direction of the bending moment of the pipeline, therefore stress from bending over the radius \( R \) and stress from the load \( q \) will be summarized:

\[ \sigma_{cur} = \sigma_R + \sigma_q \]  \hspace{1cm} (20)

It means that the maximum stress will exceed the stress \( \sigma_R \) determined according to (16). Now let us find the distance \( L_s \) between the supports from:
For this purpose, we express $L_s$ from (19):

$$L_s = \frac{24 \cdot I \cdot \sigma_q}{q \cdot D} = \frac{24 \cdot I \cdot \sigma_R \cdot J}{q \cdot D}$$  \hspace{1cm} (22)$$

As a result, we will obtain the value of bending stresses for steel 09G2S (A 516-55, A 516-55, A 516-65, A 561 Gr 70 - USA; SM41B, SB49 - Japan) that equals 400 MPa. Therefore, we can set the distance between the supports $\sim 40$ m taking into account the safety factor $\sim 2$. At the same time, boundary conditions at $O$ is not exactly a fixed-end condition and even more so freely supporting, $M_O = 0$, because the diameter of the well is much bigger than the diameter of the pipeline. Consequently, stresses at $O$ can grow stronger than in the middle part of the pipeline ground section. Namely, the distance between the supports must be $\sim 20$ m.

Next, let us consider the possibility of using equipment for pushing a pipe by means of the axial force applied at $D$ (figure 2) to intensify the pipeline pullback process. Application of the axial force $T$ at $D$ causes the appearance of compressive stresses:

$$\sigma_T = \frac{T}{A}$$  \hspace{1cm} (23)$$

where $A$ – is the cross-sectional area.

This stress is not very big. For example, application of the axial force $T = 10^6$ H to a pipe with the diameter $D = 530$ mm and the wall thickness $\delta = 10$ mm increases the stress by about 61 MPa. On the other hand, the axial force will cause the appearance of the excess bending moment in the curved $OC$ section and the stress from this bending moment may be significant. [7].

We will consider a pipeline in a deformed condition to determine the stress-strain state under the action of the axial force. This pipeline is described in figures 1 and 2. The supports under the pipeline are unilateral constraints, since they do not prevent the pipe from tearing off them. Accounting of the unilateral constraints considerably complicates solving the problem. In this work, the following method is applied. At first, the problem is solved without the constraints. The constraints are restored in the points with a negative bent $W$ and the problem is solved again. Further, we check the deflections. And if at all points the deflections are positive, then the obtained results are accepted for the solution of the problem. Note that the direction of the deflection $W$ and hence the connections placement depends on the value of the force $T$ for a given value of the pipeline weight. Therefore, we have to repeat all the calculations for each value of the force $T$.

Thereby we found the general solution of the differential equation of equilibrium for every section between the supports.

3. Results and Discussion

The solutions of these equations were obtained. Analyzing the obtained solutions, we can make a conclusion. Let us take as an example a special case where the axial force of the pusher is applied to the elastically bent pipeline that has a diameter $D = 0.53$ m, a wall thickness $\delta = 0.01$ m and a radius of curvature $R = 1000 \cdot D$ (figure 4).

![Figure 4](image)

Figure 4. A special case of determining the stress-strain state of a pipeline when using a pusher. A complete absence of connections at intermediate supports is shown in a scheme in figure 4.
The calculation results show that deflections at all points are less than zero, under the action of the axial force \( T = 6 \cdot 10^5 \) N or less. This means that for this value of the force \( T \) the pipeline should remain lying on the supports.

When increasing the axial pushing force to \( T = 7 \cdot 10^5 \) N, the pipeline loses contact with the support at \( B \), but at other intermediate points the deflections are still less than zero. That is, while \( T = 7 \cdot 10^5 \) N, the pipeline will lay on all supports except point \( B \).

In case of increasing the axial force to \( T = 8 \cdot 10^5 \) N, the pipeline loses contact with all supports at intermediate points. Wherein the lifting will be up to 0.4 m at \( B \). This pipeline lifting may be dangerous and it could cause pipeline falling during the pullback process. If at the same time we put a connection at \( B \), the pipeline will rise above the support at \( l \) (figure 4), but it will be a very small value, about 1.5 mm.

If the axial force reaches \( T = 10^6 \) N, it will cause pipeline rising above all supports up to 0.3 meters. Herewith at \( l \) the pipeline will rise above the support up to 0.13 meters, and the stress in the pipeline will reach a value of up to 340 MPa. These stresses will be caused mainly by bending, whose value can be reduced noticeably by adding retaining connections in intermediate points of the bent pipeline.

4. Conclusion

Thereby, we can apply to the pipeline the axial force of a pusher up to \( 6 \cdot 10^5 \) N without using the additional upper support devices. These dependences were obtained for the pipeline entrance angle into the borehole \( \alpha = 5^\circ \). Larger values of angles can be investigated as well, but it is obvious that an increase in the entrance angle will cause a decrease in the permissible value of the axial pushing force applied to the pipeline.

Applying the force of a pusher to the free end of the pipeline located on the day surface is the most easily implemented way in the practice of pipeline construction. Significant results can be achieved using this most economic and universal method of pullback.

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