Theoretical grounding and controlling optimal parameters for water flooding tests in field pipelines

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Abstract. Emergencies and breaks of field pipelines result in both gas, oil, and refined products waste and high costs for maintenance and repair evaluated in billiard rubles annually. The purpose of this work is calculate test pressure in field pipelines with the account of pipes ovality and curvature, stress pattern, surface damage areas, plasticity of pipe steels and optimal parameters control. To determine maximum test pressure it is advisable to divide the pipeline into parts for hydraulic tests, denote sources of water feeding and points of water discharge. Minimum elastic bend radius (EBR) and walls thickness (WT) should be found for every part. Unfailing service can be provided by periodical hydraulic tests, e.g. once in three years (in case when the test pressure is \( \sigma_{tp} = \sigma_{op} \) or by periodic decrease of the operational pressure. The other way to provide unfailing service of the pipeline is decreasing cyclic load.

1. Introduction
Oilfield pipelines are important engineering constructions. Their emergencies and breaks result in gas, oil, refined products waste and require high repair and maintenance costs evaluated in billiard rubles annually. Pipelines emergencies are accompanied by bursts, fires, water, soil and air pollution, which destroy flora, fauna, and economics. All these necessitate quality and reliability increase of pipelines by hydraulic tests. Pipeline analysis and safety takes important place in providing their reliability. Oil pipelines are systems of consequently connected elements (pipes, fittings, tubular parts), so failure of any of them results in oil transportation stop and economic wastes.

2. Results and discussions
At the factory every pipe undergoes short (less than 30 s) hydraulic tests according to GOST 31443-2012. At domestic factories test pressure is lower than 0.90-0.95\( \sigma_{tp} \) (\( \sigma_{tp} \) is normal flow pressure). Some foreign companies use test pressure 1.1\( \sigma_{tp} \). But at factories short run-tests means for crack detection and non-destructive tests do not allow determining all the failures in pipes. To determine metallurgical and constructional failures after rolling and welding in pipe production, and constructional failures effective preoperational tests should be conducted [1].

Hydraulic testing of pipelines with increased pressure during operation determines defects in pipes walls metal; durability of pipes with big diameter after tests is determined by residual defectiveness. Pipe defect area depends on hydraulic tests parameters (pressure, soaking time, cycle number).

Hydraulic test parameters are: test pressure \( p_h = (1.1\ldots1.5) \ p_{oper} \), this pressure soaking time is \( t_i = 6\ldots24 \) h and load cycles number \( N_i = 1\ldots3 \) are different in regulation documents (CS 6.13330.2012, CS...
86.13330.2014 and others).

Test norms for active pipelines are regulated by Regulations of testing linear parts of active pipelines (GOST R 53580-2009) which determine test pressure by Marriott’s formula \( p_m = 2 \cdot 0,95 \sigma_f \delta / (D_o - 2 \delta) \), where \( \delta \) is pipe wall thickness, mm; \( D_o \) is outside pipe diameter, mm.

Marriott’s formula gives rough estimation (error \( \pm 25 \% \)), as it does not take into account type of stress, changes in pipe size to the moment of failure, strain hardening coefficient \( n \), plastic properties of pipe steels and surface defects dimensions, pipeline caverns and fractures which lead to failures. Test pressure usually determined by difference in high points of the tested part is found with the account of hydrostatic pressure.

Hydraulic tests of the operating pipelines are usually done after their reconstruction, technological re-equipment and total overhaul. Pure and chemically neutral water is used in tests. Test time depends on the part diameter and length and is 48-165 hours for pipes 500-1200 mm in diameter and length 10–40 km [1].

The purpose of our work is to calculate test pressure for oil pipelines with the account of ovality and curve of pipes, stress pattern, surface failure areas, and plasticity properties of tube steels. In foreign practice there is no unified approach for choosing test pressure level [1]. Regulatory documents on designing, constructing and operation of oil pipelines (CS 6.13330.2012, CS 86.13330.2014) do not provide precise data about test pressure values, causing metal strain equal to EBR [2]. For example, regulations in CS 6.13330.2012 are met by conditions

\[
\begin{align*}
\sigma_{rs}^i &= 0.9 \sigma_{fp}; \\
\sigma_{lons}^i &= 0.9 \sigma_{fp}; \\
\sigma_{equiv}^i &= 0.9 \sigma_{fp};
\end{align*}
\]

and regulations CS 86.13330.2014 are met by conditions

\[
\begin{align*}
0.9 \sigma_{fp} &\leq \sigma_{rs}^i \leq \sigma_{fp}; \\
0.9 \sigma_{fp} &\leq \sigma_{lons}^i \leq \sigma_{fp}; \\
0.9 \sigma_{fp} &\leq \sigma_{equiv}^i \leq \sigma_{fp};
\end{align*}
\]

where \( \sigma_{rs}^i, \sigma_{lons}^i \) are radial and longitudinal stresses correspondingly occurring at maximal test pressure, MPa; \( \sigma_{fp} \) is steel flow pressure, MPa; \( \sigma_{equiv}^i \) are equivalent stresses (stress intensity), calculated by energy theory and corresponding to \( \sigma_{rs}^i, \sigma_{lons}^i \), MPa.

3. Experimental part

During the test conditions (1) and (2) cannot be fulfilled as for example at \( \sigma_{rs}^i = 0.9 \sigma_{fp}, \sigma_{lons}^i = -0.9 \sigma_{fp} \) equivalent stress reaches \( 1.56 \sigma_{fp} \); for steel X60 it exceeds standard value for ultimate resistance stress \( \sigma_u \).

With the necessity to determine maximal number of defects of different origin and tendency to increase test pressure work [3] gives conditions conforming this to maximum:

\[
\begin{align*}
\sigma_{rs}^i &= u \sigma_{fp} \leq \sigma_{man}; \\
|\sigma_{lons}^i| &\leq \sigma_{fp}; \\
\sigma_{equiv}^i &\leq \sigma_{fp};
\end{align*}
\]

where \( u = \sigma_i / \sigma_{fp} = 0.9 \) (is taken according to CS 86.13330.2014); \( \sigma_{man} \) is radial pressure caused by manufacture test pressure, MPa.

Nearly all oil pipelines have ovality defects. S.P. Timoshenko solved this problem [4]. Critical radial pressure \( p_{rs}^c \), resulting in pipe yield can be written neglecting residual stress with the account of operation. Analyzing the influence of pipe form changes and load biaxiality coefficient for pipeline \( m \) on failure pressure, having developed and specified operational methods [4], we receive new formulas for the pipe [5]. Analysis of these formulas show that destructing internal pressure, which results in loss of carrying capacity is greater (by 5 % at tensile loads, 9 % at compressing loads) then the pressure
received by the methods not taking into account coefficient changeability \(m\). On the other side, in work [6] basing on the comparative analysis of the cross section decay coefficient by a crack conducted by Battelle Memorial Institute in the USA with determined experimental data the formulas were specified for failure pressure \(p_{\text{fail}}\). Combining these results we receive new formulas for pressure \(p^l_{rs}\) for the pipeline with the surface defect:

\[
    p^l_{rs} \leq (u\delta/ND)\left[B - (B^2 - 2n\sigma_p p_E N D_{\text{out}}/\delta)\right],
\]

where \(N = 1/K_p K_f p\); \(B = n\sigma_p + 0.5p_E D_{\text{out}} N/\delta + 0.75p_E (D_{\text{out}}/\delta)^2\); \(p_E = 2E/(1 - \mu^2)(D_{\text{out}}/\delta)^3\); \(\Delta = (D_{\text{out}} - D_{\text{in}})/D_{\text{middle}}\); \(K_p = (\delta + l)/[\delta - l/M]\); \(M = 1 + 1.61(L/cos\omega)^{1.65}/2D_{\text{out}}\delta\); \(K_f p = (2/3)^n\xi^{n-1}[1 + (2m - 1)^2(2 - m)n/6\xi^2]\); \(\xi = \sqrt{1 - m + m^2}\); \(\omega\) is fabrication weld inclination angle; \(D_{\text{out}}, D_{\text{in}}, D_{\text{middle}}\) are outside, internal, and middle pipe diameters, mm; \(L, l\) are length and depth of the surface defect; \(m\) is relation of longitudinal stress to radial one.

Test pressure \(p^l_{\text{long}}\) causing longitudinal stresses \(\sigma^l_{\text{long}}\) in metal of curve pipeline, with the account of works [3, 6] can be received by formula

\[
    p^l_{\text{long}} = 2\delta(\sigma_p - C)/qD_{\text{in}}N,
\]

Where \(q = \mu + \beta(0.5 - \mu); \mu = 0.3\) is Poisson’s ratio; \(\beta = (1 + 5\gamma)/(1 + 15/16)(\gamma^2/\eta^2)\); \(C\) is a part of total longitudinal stresses, determined for curvilinear pipes and elastically bent pipe parts with the account of work [6] by formula \(C = -(1 - \beta)\alpha\Delta t/E (E D_{\text{out}}/2\rho)\); \(\alpha\) is a coefficient of linear expansion equal \(1.2 \cdot 10^{-5}\ K^{-1}\); \(E\) is elasticity modulus equal \(2.1 \cdot 10^5\ MPa; \Delta t\) is temperature difference during the test \(K; \rho\) is minimal elastic bend radius of the pipe axis on the tested part, m; \(\gamma = D_{\text{out}}/2l\); \(\eta = f/l_{\text{pipe}}\) is pipeline curvature; \(l_{\text{pipe}}\) is length of curvilinear pipeline part, m.

Studying approaches in [7–12] we receive test pressure \(p^l_{\text{equiv}}\) along equivalent stresses \(\sigma^l_{\text{equiv}}\) for a more complex case (curvilinear pipeline with longitudinal defects):

\[
    p^l_{\text{equiv}} \leq \frac{2\delta C(N - 2\mu q) + \sqrt{4(N^2 - N\mu q + \mu^2 q^2)R_t^2 - 3N^2C^2}}{nN D_{\text{in}}(N - 2\mu q + \mu^2 q^2)}.
\]

where \(R_t < \sigma_p\) is calculated pressure.

From pressures received by formulas (4)–(6) we determine the minimal and compare it to manufacture test pressure. Less of these pressures will be maximal test pressure. Further, according to new specified formulas we receive with the account of [3, 6]:

\[
    \sigma^l_{rs} = \frac{N p^l_{rs} D}{2\delta} + \frac{3}{4}\left[D_{\text{middle}}(D_{\text{middle}}/\delta)\right]^2 \frac{dp^l_{rs}}{1 - (p^l_{rs}/p_E)},
\]

\[
    \sigma^l_{\text{long}} = \left[p^l_{rs} D_{\text{middle}}(D_{\text{middle}}/\delta)\right]\left[\mu + \beta(0.5 - \mu)\right] + C; \quad \sigma^l_{\text{equiv}} = \sqrt{(\sigma^l_{rs})^2 - \sigma^l_{rs} \sigma^l_{\text{long}} + (\sigma^l_{\text{long}})^2}
\]

and check corresponding stresses which must meet the conditions (3).

For the studied case value \(\Delta t\) must be equal to temperature differences of pipe walls, during the moments of pressure rise end till test and closing calculating scheme of the tested pipeline. With the sufficient precision the pipeline wall temperature at calculation scheme closing moment can be taken (with spare) equal to temperature of the air outside. Pipeline temperature at the end of the pressure rise can be considered equal to the temperature of the tested medium [2].

The first addend in equation (7) is average membrane stresses in the pipe and the second one is curve stress which depends on pipe ovality.

But the first addend is received by Marriotte’s formula and gives a crude error (error \(\pm 25\%) of pipe strength assessment as it does not take into account plasticity qualities of pipe steels and pipelines defects parameters which result in failures. Load biaxiality, diameter changes and pipe thickness are not taken into account to the moment of failure. Besides surface defects (scratch marks, caverns, cracks, mechanically unsmooth radial welds, parts of localized strength and plasticity with decrease in pipe walls thickness) which can result in pipe failure of this part.

When there is no ovality in this pipe at first approximation it is possible to apply works [2] and our more precise formulas (4), (5), (7), (8).

Basing on the said above to determine maximal test pressure it is rational according to paragraph
11.27 CS 86.13330.2014 to divide the pipeline into parts for hydraulic tests, and determine sources for receiving water and places of damping water. For every part it is necessary to find minimum elastic bend radius and walls thickness.

The calculations showed that choosing pressure for hydraulic tests should meet the conditions (3) and recommendations [2], and supplement to paragraph 11.27 CS 86.13330.2014 and parts 2 and 3 of GOST 31443-2012 of the Instruction on hydraulic test of pipelines with increased pressure (stress-test method) should be edited as follows: “Herewith longitudinal and equivalent stresses should not exceed normal (guaranteed) flow pressure”.

4. Conclusion
Failure-free operation of the pipeline can be provided by regular hydraulic tests about once in three years (in case when the test pressure \( \sigma = \sigma_0 \) or by regular lowering operational pressure. Other method of providing failure free operation of the pipeline is decrease in load cyclicity.

For example, to provide failure-free operation during 15 years (at \( \sigma = \sigma_0 \) number of load cycles should not exceed 90. Reinforcing the pipeline can be recommended as overhaul for failure-free operation of pipelines. For this experimental prototype technologies and machines for local and continuous reinforcement of the pipelines have been developed. Regular hydraulic tests with increased pressure can provide decrease in pipelines failure and improve their operational reliability.

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