Chair of Petroleum and Geothermal Energy Recovery

Master's Thesis

Investigation of methods to assess the operability of trunk pipelines with flaws in a combined stress state.

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September 2022

MONTANUNIVERSITÄT LEOBEN
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AFFIDAVIT

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Master Thesis 2022
Petroleum Engineering

Investigation of methods to assess the operability of trunk pipelines with flaws in a combined stress state

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Chair of Petroleum and Geothermal Energy Recovery

I dedicate my master's thesis to my family,

who has always supported and believed in me during my life?

Acknowledgements

My master's thesis was helped by my supervisor from Ufa State Technical University, Associate Professor of the Department of Transport and Storage of Oil and Gas, Candidate of Technical Sciences Gumerov Aidar Kabirovich. Also, my master's thesis was helped by my supervisor from Montanuniversität Leoben Prof. Herbert Hofstätter.

Abstract

Despite the fact that many studies have been carried out in the oil and gas sector and there are many standard technical documents regulating the assessment of the operability of trunk pipelines (TP) with flaws, the problem of assessing the durability and operability of main pipelines is still present.

At the moment, the international standard of the company DNV [8], which serves as the basis for Russian analog [5] is the most comprehensive method for determining operability and residual strength of trunk pipelines with defects. However, they do not consider taking into account axial and bending loads when evaluating pipeline sections with interacting defects.

The aim of the scientific work is to improve the standard technical documentation of trunk pipelines with defects subjected to combined loading.

As a result of the work, a comprehensive analysis of technical documentation and theoretical methods was performed. Considering different flaw options numerical model for the assessment of residual strength of trunk pipeline in a combined stress state was created and the importance of taking into account bending stresses was proven using the Ansys software package. Improved methodology for assessing the operability of trunk pipelines with interacting flaws taking into account longitudinal stresses was obtained. In addition, the ability to take into account the actual location of the defect relative to the applied loads was added to the methodology.

Zusammenfassung

Trotz der Tatsache, dass im Öl- und Gassektor viele Studien durchgeführt wurden und es viele technische Standarddokumente gibt, die die Bewertung der Betriebsfähigkeit von Hauptleitungen mit Mängeln regeln, bleibt das Problem der Bewertung der Haltbarkeit und Betriebsfähigkeit von Hauptleitungen bestehen.

Derzeit ist der internationale Standard der Firma DNV [8], der als Grundlage für das russische Analogon [5] dient, die umfassendste Methode zur Bestimmung der Funktionsfähigkeit und Restfestigkeit von Hauptleitungen mit Mängeln. Sie berücksichtigen jedoch keine Axial- und Biegebelastungen bei der Bewertung von Rohrleitungsabschnitten mit Wechselwirkungsfehlern.

Ziel der wissenschaftlichen Arbeiten ist die Verbesserung der einheitlichen technischen Dokumentation von Sammelleitungen mit Mängeln bei kombinierter Belastung.

Als Ergebnis der Arbeit wurde eine umfassende Analyse der technischen Dokumentation und der theoretischen Methoden durchgeführt. Unter Berücksichtigung verschiedener Fehleroptionen wurde ein numerisches Modell zur Bewertung der Restfestigkeit der Hauptrohrleitung im kombinierten Spannungszustand erstellt und die Bedeutung der Berücksichtigung von Biegespannungen unter Verwendung des Ansys-Softwarepakets nachgewiesen. Es wurde eine verbesserte Methodik zur Bewertung der Betriebsfähigkeit von Hauptrohrleitungen mit interagierenden Fehlern unter Berücksichtigung von Längsspannungen erhalten. Darüber hinaus wurde die Methodik um die Möglichkeit erweitert, den tatsächlichen Ort des Defekts relativ zu den aufgebrachten Lasten zu berücksichtigen.

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Chapter 1

Introduction

1.1 Background and Context

With the rapidly increasing demand for oil and gas supplies, the pipeline industry is driven to develop large diameter, thin wall thickness, high operating pressure and high grade steel pipeline systems. Moreover, there have been increasing pipeline activities in the harsh geographic environments, such as the Arctic and Siberia regions, where significant soil movements are encountered.

During service, the remaining strength of pipelines depends on a number of factors, including the operational and environmental conditions, combined stress state and defects introduced by construction, third-party damage, corrosion and ground movement, etc.

Despite the fact that many studies have been carried out in the oil and gas sector and there are many codes and standards regulating the assessment of the operability of trunk pipelines with flaws, the problem of assessing the durability and operability of trunk pipelines is still present. All work on the design, construction, operation, repair, as well as periodic inspections and assessment of the technical condition of oil and gas pipelines, taking into account accumulated flaws, are regulated by a system of codes and regulatory standards. This implies regulatory framework's decisive role in ensuring pipelines' operability and safety. The achievement of the limiting state of the pipeline occurs due to a complex of factors, including the presence of defects, operating conditions, cyclic load, environmental conditions, as well as the complex state of the pipeline, including longitudinal stresses from pressure, temperature and elastic bending. If some phenomena and mechanisms of wear are not provided in the regulatory framework, then measures to counteract these phenomena will not be provided either; wear and destruction by these mechanisms will occur on all pipelines.

At the moment, there are various approaches and calculation methods for solving the problem of assessing the risk of pipe defects. The accuracy of calculations is necessary in order to know the pipeline's condition as close to reality as possible. This will also reduce the cost of subsequent repairs and unforeseen pipeline failures and related emergencies.

Therefore, the elaboration of a regulatory framework for taking into account the complex of impacts is an urgent problem in the oil and gas sector.

Domestic and foreign scientists were engaged in the development of methods of reliability, operability and evaluation of the residual strength of trunk pipelines: I.N. Birillo, L.I. Bykov, K. M. Gumerov, Fu Bin, P. Hopkins, G.E. Korobkov, I. Milne, A. T. Stewart, J.F. Kiefner, F. J Klever, T.A. Netto, D. R. Stephens and others and various organizations. The results of the work done have made a significant contribution to the development of methods for assessment of operability of trunk pipelines with flaws. The main defect parameters and different failure criteria influencing strength of pipeline were investigated, different failure concepts and their accuracy determined.

1.2 Scope and Objectives

The scope is to improve standard technical documentation (STD) for the assessment of the residual strength of trunk pipelines with flaws subjected to combined loading.

Research objectives:

1. Analyze the available Russian, European and American standards, codes and methods for the assessment of trunk pipelines with corrosion flaws.

2. Create a numerical model for the assessment of the operability of pipelines with corrosion flaws using Ansys software package.

3. Using Ansys software package prove the importance of bending stresses, which always present in almost any pipeline.

4. Improve the methodology for assessing trunk pipelines with interacting flaws subjected to both internal pressure and longitudinal stresses, considering flaws location relative to the impact zone of compressive/tensile longitudinal stresses.

1.3 Achievements

During the work, the following activities were carried out and the following results were obtained:

1. Initially, several models of the pipe segment with flaws were modeled in COMPASS 3D. Then the numerical model was created in Ansys software package and was verified by burst tests performed by [46]. In addition, numerical model was verified by analytical calculations and compared with the methodology [5,8], the improvement of which is considered in this paper.

2. The characteristic of elastic bending of trunk pipelines was considered on the example of in-line diagnostics, which indicates that the values of the bending radius do not correspond to the limit values of the bending radius during the actual construction of trunk pipelines.

3. The importance of considering the bending stresses has been proved using Ansys software package.

4. Based on numerical model and Table Curve 3D program a correction factor was obtained to consider the interaction of two nearby corrosion defects.

5. The improved methodology (which is missing in [5,8]) for the assessment operability of trunk pipelines with interacting corrosion flaws subjected to combined loading has been developed.

6. The ability of accounting flaws location relative to the impact zone of compressive/tensile longitudinal stresses has been added to the methodology.

1.4 Technical Issues

In the course of the master's work, a theoretical description of the methodology was carried out, as well as a verification of the numerical model to confirm that the methodology is technically feasible. For more accurate results full-scale burst tests should be carried out in real conditions. Then it will be possible to judge the exact distortions in the readings between the methodology and the real data.

1.5 Overview of Dissertation

The following points were considered in the dissertation:

1. Literature review on this topic has shown that the topic is relevant and currently the assessment of interacting corrosion flaws subjected to combined loading is not considered in codes and standards.

2. The characteristic of elastic bending of trunk pipelines was considered on the example of in-line diagnostics, which indicates that the values of the bending radius do not correspond to the limit values of the bending radius during the actual construction of trunk pipelines.

3. Comparison analysis was performed in order to understand the most accurate existing methods.

4. During development of finite element model selection of the boundary conditions, step incrementation, mesh optimization and model verification were described.

5. Influence of the main flaw parameters on the failure pressure and failure criterion of the steel were investigated.

Chapter 2

State of the Art

2.1 Characteristics of the normative calculation for the strength and stability of trunk pipelines

In domestic practice [1], the assessment of pipelines for strength and stability begins with the determination of the standard tensile (compression) resistances of the metal of pipes $R_1^н$ and $R_2^н$, which are assumed to be equal to the corresponding values of the tensile strength and yield strength of the pipe material.

In turn, the calculated values of the tensile (compression) resistance R_1 and R_2 are calculated from the expressions:

$$R_1 = \frac{R_1^н m}{k_1 k_н}, \tag{2.1}$$

$$R_2 = \frac{R_2^н m}{k_2 k_н}, \tag{2.2}$$

where m – coefficient of pipeline operating conditions;

k_1 and k_2 – reliability coefficients for the material;

$k_н$ – reliability coefficient for pipeline responsibility.

Further, the strength and stability of underground pipelines (without defects) are evaluated to prevent unacceptable plastic deformations according to the conditions:

$$\left|\sigma_a^н\right| \le \psi_1 \frac{m}{0,9k_н} R_2^н, \tag{2.3}$$

$$\sigma_h^н \le \frac{m}{0,9k_н} R_2^н, \tag{2.4}$$

where ψ_1 – coefficient that takes into account the biaxial stress state of the pipe material;

$\sigma_h^н$ – circumferential (hoop) stresses from internal pressure, MPa, defined by equation:

$$\sigma_h^н = \frac{pD_i}{2\delta}, \tag{2.5}$$

where p – internal pressure, MPa;

D_i – inner pipe diameter, mm;

δ – wall thickness of the pipe, mm.

In turn the total longitudinal (axial) stresses in the pipeline $\sigma_a^н$ are determined from all (taking into account their combination) loads. Therefore, the calculation should take into account: internal pressure, temperature difference and elastic bending and it is defined by equation:

$$\sigma_a^н = \mu\sigma_h^н - \alpha E \Delta t \pm \frac{ED}{2\rho}, \tag{2.6}$$

where μ – Poisson's ratio,

α – coefficient of linear expansion of the steel, $\frac{1}{°C}$,

E – Young's modulus, MPa,

Δt – calculated temperature difference, °C,

D – outer pipe diameter, mm,

ρ – elastic bending radius of the pipeline, mm.

2.2 Characteristics of bending radii of trunk pipelines

The huge system of oil and gas pipelines of the Russian Federation is aging, which is confirmed by data [2], according to which in 2017 more than half of the trunk pipelines have been in operation for 40 years or more. The operation of trunk pipelines (TP) according to technical condition is the main concept of maintaining reliability and stability. In turn, the technical condition is assessed using the information obtained during the TP diagnostics, in which the most widely used is in-line inspection (ILI).

The main advantage of this method is a direct way to detect defects and bending radii. The technology makes it possible to accurately measure the actual bending radii of the examined section of the trunk pipeline (TP), which makes it possible to reliably determine bending stresses. In turn, calculations related to the assessment of the stress-strain state, residual strength and operability of the TP (both with and without defects) are carried out in accordance with the standart technical documentation (STD) intended for the design, construction and operation of the TP.

The graph of the measured radii of elastic bending of a straight section of the main gas pipeline is represented on the Figure 2.1. The data is obtained from VTD "NPC In-Tube Diagnostics" for the main pipeline with a diameter of 1420 mm and selection of radii of 1000D and less [3].

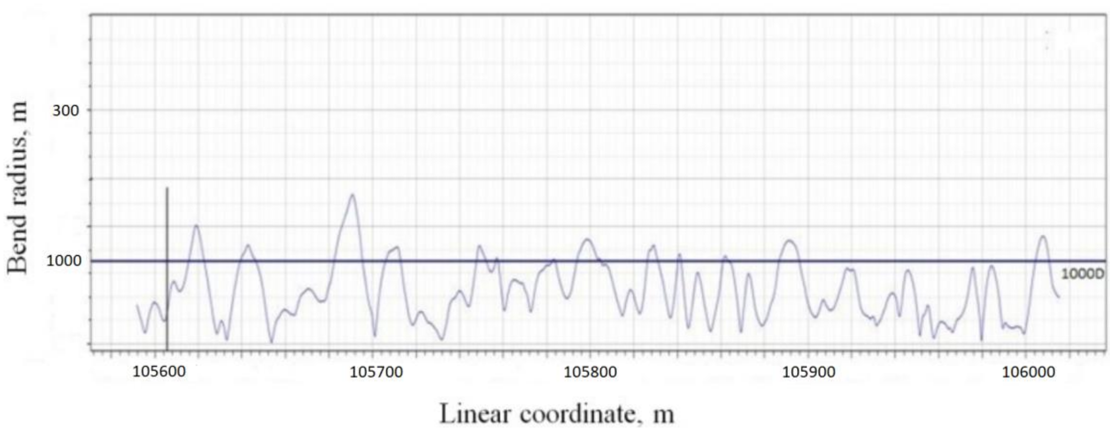

Figure 2.1 Elastic bending radii of the section of the trunk gas pipeline

One can note, that on the straight gas pipeline section with the length of 400 meters and exploitation period of 35 years there are 13 sections with the elastic radius of bending 1000D and less. It is an actual non-compliance with the requirements [1] where it is said that radius of elastic bending must be at least 1000D. In addition, another interesting fact is that due to such bending radii, both compressive and tensile stresses may be present in different pipe sections. These kind of radii of elastic bending from Figure 1 can develop from a variety of cases: pipe sections are welded at an angle (but still the regulations are met); the bottom of the pipeline trench cannot be made straight; when pipe lay on some local solid bumps, the pipe bends under the weight (own and soil weight); fluctuations of the in-line diagnostics reading. Even during the construction of straight sections of pipes, taking into account all the requirements and norms of design, construction and operation, sections with a bending radius of less than 1000D may occur [4].

The situation is even worse and more complicated in cases of sections laid by elastic bending and cold bending. Figure 2.2 shows an example of in-line diagnostics of the pipeline crossing the ravine (cold bends are highlighted in blue), where only short sections satisfy the condition 1000D or more.

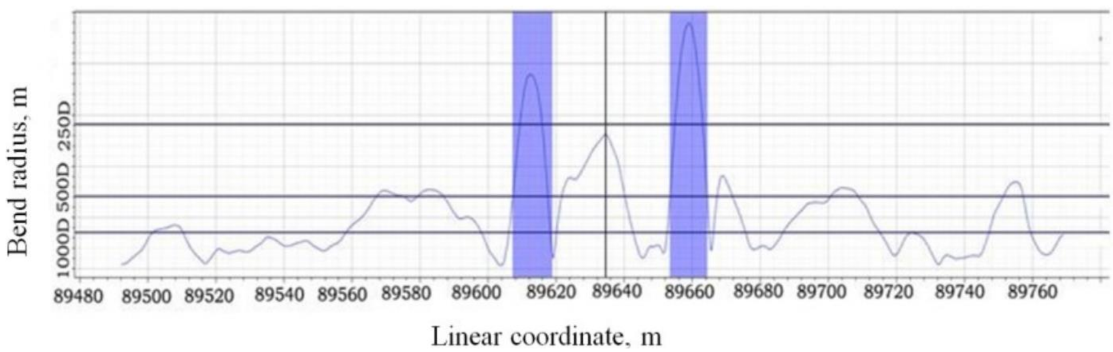

Figure 2.2 Elastic bending radii of the section of the trunk gas crossing

Considering all the above, we can conclude that almost all linear part of the main pipeline system in Russian Federation does not meet the requirement of the bending radius of 1000D [1]. However, there was no increase in the accident rate and means that it is desirable to provide bending radii within the standards (but it is not practical and even possible) and it is necessary to update the regulations and standards and justify the permissible value of the axial stresses [4].

2.3 Domestic standard technical documentation for assessing the operability of trunk pipelines with flaws

In domestic practice, as well as in the world, to determine the operability of trunk pipelines, they usually use the experience of scientific and practical work on existing pipelines, databases obtained during testing of pipe samples and subsequent experimentally confirmed criteria and methods.

The Russian Gazprom standard [5] describes the provisions and requirements for assessing the technical condition and conducting an examination of trunk gas pipelines with corrosion flaws. This standard is applicable for corroded sections of underground and surface gas pipelines of category I-IV, made of low-carbon and low-alloy steels, operated at wall temperatures from minus 40 °C to +60 °C with an excess gas pressure of up to 9.8 MPa. However, this standard is not applicable for evaluating sections of gas pipelines under cyclic load conditions, as well as for evaluating the growth rate of corrosion defects over time.

The disadvantages of this standard include the fact that within the framework of this methodology, most corrosion flaws are evaluated except for: cracks and crack-like defects; corrosion cracking, factory and assembly welding defects unacceptable according to the instructions [6]; mechanical damage (tears and scratches) and violations of the shape of the pipe section (ovality, dents and wrinkles); corrosion flaw combined with mechanical damage; corrosion defects located in places of stress concentration (pipes, block valves, tees, elbow bends, joints of parts and other elements of pipeline structures with abrupt changes in geometry); and defects with a depth of more than 80% of the nominal (without corrosion) thickness of the pipe wall.

The essence of the technique is to replace a real defect on the surface of the gas pipeline with an equivalent defect, which is characterized by the following geometric parameters, indicated in Figure 2.3:

- Length *l*, denoting the length of the projection of the defect on the longitudinal plane of the section of the pipe wall;

- Width *c* denoting the length of the projection of the defect on the circumferential plane of the section of the pipe wall;

- Depth *d*, varying in length and width of the defect;

- Area *A* denoting the area of projection of the defect on the longitudinal plane of the pipe wall section;

- Area A_c denoting the area of projection of the defect on the circumferential plane of the section of the pipe wall.

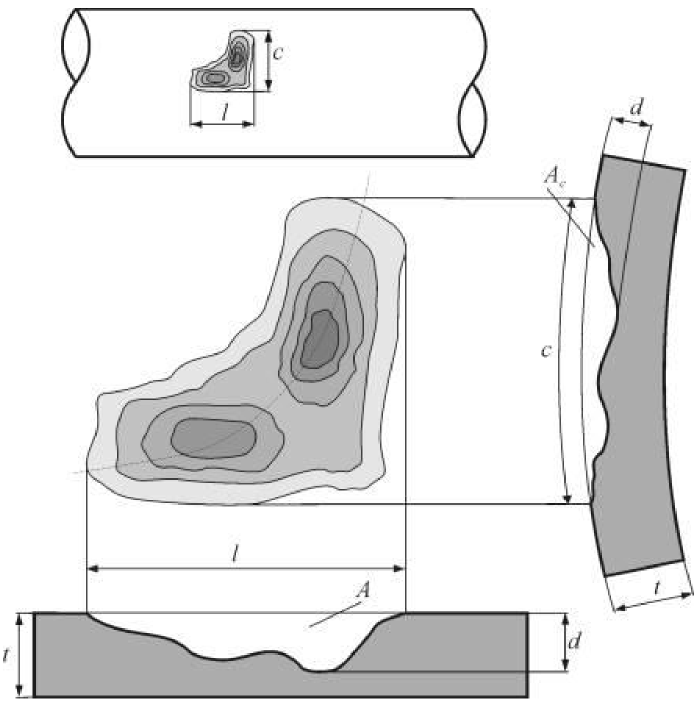

Figure 2.3 Schematics of the flaw

After that, the operability of pipeline sections with corrosion damage is evaluated according to the algorithm given in the form of a diagram in Appendix A and including the following basic actions:

- Determination of the initial data listed below for a section of a gas pipeline with corrosion damage:

a) Geometric characteristics of the pipe cross-section;

b) Pipeline section categories;

c) Physical and mechanical characteristics of the pipe material;

d) Loading conditions of the gas pipeline section;

e) Geometric characteristics and location of corrosion defects, as well as distances between adjacent defects.

- Calculation of the safety margin *K*, considering the rules and regulations [1], and recommendations [7]:

$$K = \frac{0,9\gamma n_p k_1 k_н}{m}, \tag{2.7}$$

where 0,9 – correcting factor;

γ – coefficient that takes into account the operating (normative) pressure p on the estimated section of the pipeline, calculated by the formula:

$$\gamma \approx 1 - \frac{n_p p}{R_1}, \tag{2.8}$$

where R_1 – calculated tensile (compression) resistance calculated in accordance with the norms and rules [1] by the formula:

$$R_1 = \frac{m}{k_1 k_н} \sigma_u, \tag{2.9}$$

where σ_u – the minimum value of the tensile strength accepted by state standards for pipe specifications;

coefficients n_p, m, k_1, and $k_н$ – defined according to standard [1].

- Evaluation of single flaws for all sections of the pipeline with the above-mentioned damages, considering (if necessary) stresses from internal pressure in combination with longitudinal tensile or compressive stresses from axial and bending loads;

- Assessment of corroded areas without taking into account longitudinal stresses from axial and bending loads, taking into account stresses only from internal pressure for cases:

a) Individual single flaws or flaws treated in a group of defects as single;
b) Group interacting defects when establishing their interaction;
c) Defects of a complex profile in the presence of a measured defect profile along its entire length.

- Assessment of the permissible pressure of the p_p, for all the above variants of the defect analysis according to the formula:

$$p_p = \frac{p_f}{K}, \tag{2.10}$$

where p_f is the failure pressure calculated in each variant of the flaw analysis.

- Comparison of the obtained permissible pressure with the working pressure and deciding on further operation or repair of a section of pipelines with corrosion damage.

In this method, the criterion for the operability of a section of a gas pipeline with corrosion damage is the excess of the above-described design permissible pressure over the operating pressure at which the section is operated:

$$p_p \geq p. \tag{2.11}$$

Accordingly, according to Appendix A, in this methodology, at the first stage, a preliminary assessment is used for any defects to calculate the permissible pressure, considering any corrosion damage as a single defect and taking into account either only stresses from internal pressure, or stresses from internal pressure, axial and bending loads. If it is possible to measure the distances between adjacent single defects, their interaction is taken into account and the permissible pressure for a group of interacting single defects is calculated. This document was created on the basis of the foreign standard of DNV [8], however, the advantage of the Russian analogue is the ability to account for both compressive and tensile longitudinal stresses, while the DNV standard only accounts for compressive stresses. Another disadvantage is that in this standard there is an assessment of a group of interacting defects only from internal pressure, but without taking into account axial and bending loads.

The Russian document [9] provides recommendations for assessing the operability of gas pipelines with corrosion spots and erosive wear of the inner surface of the pipe, depending on the mechanical and geometric parameters of the pipeline, internal pressure and site category.

This technique is recommended for defects from 155 mm and up to half the diameter of the pipe. In the presence of defects with a length along the axis of more than half of the pipe diameter, it is recommended to determine the permissible internal pressure based on the strength and stability check according to [1].

The essence of the technique is to determine the permissible thinning of the pipeline wall *[c]* and the permissible operating pressure *[p]*:

$$[c] = \delta - \frac{pD_i}{2([\sigma_h] + p)}, \tag{2.12}$$

where δ – nominal thickness of the pipe wall, mm;

c – actual thinning of the pipe wall, mm;

p – operating pressure on the gas pipeline section, MPa;

D_i - inner diameter of the pipe, mm;

$[\sigma_h]$ - permissible hoop stresses, MPa.

The permissible hoop stresses are determined by the formula:

$$[\sigma_h] = \frac{m}{0,9k_н} R_2^н, \tag{2.13}$$

where m – service factor of the gas pipeline;

$k_н$ – safety factor for the gas pipeline;

$R_2^н$ – yield strength of the metal, MPa.

$$[p] = \frac{2[\sigma_h](\delta - c)}{D - 2(\delta - c)}, \tag{2.14}$$

where c – actual wall thinning.

The obvious disadvantages of the technique include the lack of consideration of the parameters of the length and width of the defect, as well as the inability to take into account axial and bending loads.

In the regulatory document of the Russian company Transneft [10], two-parameter criteria of ultimate strength and ultimate plasticity are used.

The criterion of ultimate strength is determined by the formula:

$$\Theta_u\left(\varepsilon_i,\varepsilon_0,\varphi_\varepsilon\right)=\frac{\varepsilon_i\cdot\cos\varphi_\varepsilon}{\varepsilon_{iu}}+\frac{\varepsilon_0}{\varepsilon_{0u}}. \tag{2.15}$$

The ultimate strength is reached at a value of Θ_u equal to one, in the inequality:

$$\Theta_u\left(\varepsilon_i,\varepsilon_0,\varphi_\varepsilon\right)\le 1. \tag{2.16}$$

The criterion of ultimate plasticity is determined by the formula:

$$\Theta_\varepsilon\left(\varepsilon_i,\varepsilon_0\right)=\frac{\varepsilon_i}{\varepsilon_{i\varepsilon}}+\frac{\varepsilon_0}{\varepsilon_{0\varepsilon}}. \tag{2.17}$$

The ultimate plasticity is achieved at a value of Θ_ε equal to one, in the inequality:

$$\Theta_\varepsilon\left(\varepsilon_i,\varepsilon_0\right)\le 1. \tag{2.18}$$

The coefficients ε_{iu}, ε_{0u}, ε_{ic}, ε_{0c} are the mechanical characteristics of the metal. The values of the parameters ε_i, ε_0, φ_ε are determined by the components of nominal, local deformations, which are related by the equations of the corresponding design scheme for a pipe with dimensions D, δ (or a nipple with dimensions D_n, δ_n), pressure p and defect parameters: length L, width W, depth H and the area of the longitudinal section of the defect A (the latter for the defect metal losses).

To calculate the strength of a pipe with a defect, it is necessary to compile and numerically solve nonlinear equations corresponding to the design scheme of the defect. The solution of systems of equations is achieved when at least one of the criteria is equal to one (Θ_u = 1 condition for achieving ultimate strength, Θ_c = 1 condition for achieving ultimate plasticity).

The result of solving systems of equations is:

a) failure pressure p_f (MPa), pipes with a defect at the specified defect parameters L, W, H, A (the last parameter for the metal loss defect):

$$p_\mathrm{f}=p_\mathrm{f}(L, W, H, A). \tag{2.19}$$

b) depth H_f (mm) of the defect leading to the failure of the pipe at the specified dimensions L, W and pressure $k_{\text{тр}}\cdot(p_{\text{проект}}+\Delta_p)$ (for the defect of metal loss is not determined):

$$H_f = H_f(p, L, W). \tag{2.20}$$

2.4 International standard technical documentation for assessing the operability of trunk pipelines with flaws

The Russian standard of Gazprom [5] uses some of the experimentally confirmed criteria and methods developed for practical application in the recommendations of DNV [8]. This standard was verified by conducting 400 finite element modeling processes (FEM). The modeling processes themselves were confirmed by 12 full-scale burst tests of pipes with defects conducted by DNV and 70 tests performed by BG Technology. DNV standard includes 2 methods. The first approach is to calculate the pressure resistance (allowable pressure) and safety factors taking into account the properties of the material, wall thickness and changes in internal pressure. The second approach is based on the principle of permissible stresses. The failure pressure of the corrosion flaw is calculated, and this value is multiplied by the usage factor.

The residual strength assessment in this standard is based on a database that contains more than 80 full-scale burst tests of pipes with corrosion flaws (single, interacting and complex defects) and extensive analysis by the finite element method of pipes containing defects. Part of the tensile tests was performed taking into account the influence of the imposed axial and bending loads.

Since this foreign standard is the main one for the domestic analogue [5], this recommended practice covers the following combinations of loads and defects:

1) Internal pressure load:

- Single defects;

- Interacting defects;

- Defects of complex shape.

2) Internal pressure load combined with longitudinal compressive (tensile not considered) stresses:

- Single defects.

Again, the disadvantages include the fact that the behavior of corrosion defects under combined internal pressure and bending loads and/or longitudinal loads is beyond the scope of this document. In addition, this methodology considers the behavior of single corrosion defects subject to internal pressure and axial loads during compression, but the analysis of axial loads during tension is beyond the scope of this document.

The following items are also not taken into account in the framework of the standard:

1) Materials other than carbon tube steel;

2) Linear pipe grades exceeding X80;

3) Cyclic load;

4) Sharp defects (e.g. cracks);

5) Combined corrosion and cracking;

6) Combined corrosion and mechanical damage;

7) Defects of metal loss associated with mechanical damage;

8) Defects in the manufacture of welds;

9) Defects whose depth exceeds 85 % of the original wall thickness.

The foreign standards [11,12] describe many requirements for the assessment of stresses, reactions and displacements of the pipeline that occur due to changes in pressure and temperature. These standards describe the requirements for the design, installation, inspection and testing of pipeline structures transporting gas. These documents are applicable in the range from -29 to 230 ºC. It also contains a methodology for determining the residual strength of corroding pipelines operated at hoop stresses amounting to 40 % or more of the minimum yield strength of the material. According to the method, if the depth of the defect d is less than 10 % of the wall thickness t (Figure 2.4), then a reduction in the permissible pressure is not required. If there is a defect depth d, which is more than 80 % of the pipe wall thickness t, this section with the defect should be cut out of the pipeline.

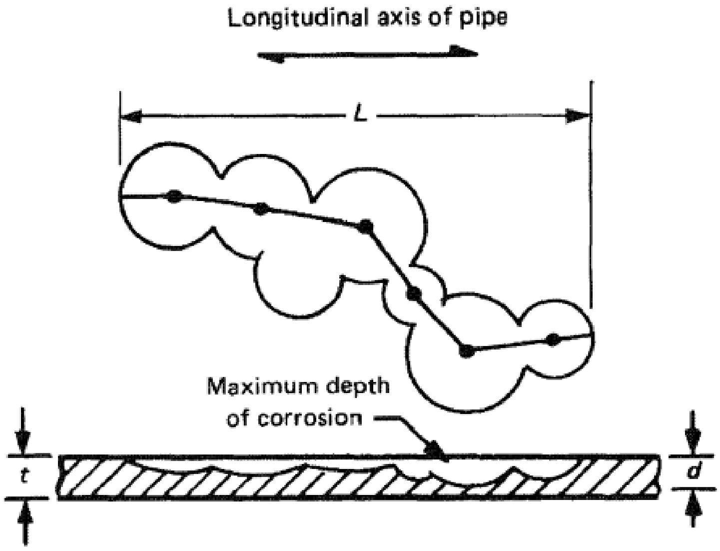

Figure 2.4 Key defect parameters

First, the dimensionless coefficient A is calculated:

$$A = \frac{0{,}893 \cdot L}{\sqrt{D \cdot t}}, \tag{2.21}$$

where L – effective length of corrosion in the axial direction of the pipe, in;

D – outer diameter of the pipeline, in;

t – pipe wall thickness, in.

For values A less than or equal to 4, a new allowable pressure is calculated:

$$P^{'} = 1{,}1P\left(\frac{1 - \frac{2d}{3t}}{1 - \frac{2}{3}\frac{d}{t\sqrt{A^2 + 1}}}\right), \tag{2.22}$$

where P – largest value between the design pressure (excluding reliability coefficients) and the established maximum allowable pressure, psig;

P' – permissible safe pressure for the corroding zone, psig.

For values A greater than 4:

$$P^{'} = 1{,}1P\left(1 - \frac{d}{t}\right). \tag{2.23}$$

Again, one can note that in this technique, when calculating the residual strength, the complex state of the pipeline, including bending loads, is not taken into account in any way.

One of the most well-known and widely used criteria for assessing the operability of pipelines with corrosion flaws is the American standard ASME B31 G [13], which provides a methodology for assessing the operability of pipelines based on a deterministic approach. The deterministic approach is the process of assessing the condition of pipelines by obtaining data from controls, while the probabilistic approach predicts the future probability of failures.

Unlike numerical methods that have a number of difficulties, which will be discussed later in the paper, semi-empirical methods are based on the analysis of the database and are obtained through full-scale experiments of pipe destruction. These data, in turn, are generalized into analytical expressions with a small amount of input data (geometric and mechanical properties of the pipe, loading conditions). ASME B31G [13] contains industrial recommendations for safe operating pressure based on the parameters of the pipeline dimensions obtained in inspections. An excellent example of the above-mentioned approach is DNV-RP-F101 [8], which recommends a probabilistic approach for evaluating pipelines with corrosion, exposed to internal pressure and internal pressure in combination with longitudinal compressive stresses.

The ASME methodology consists of a series of tables that show the relationship between the depth of the defect and the thickness of the pipe wall for different pipe sizes. These tables were obtained during a variety of experiments and modeling processes. Also in the course of the methodology, new safety and reliability factors are calculated. The simplicity of the methodology makes it possible to use it to assess the operability of pipelines with both corrosive and crack-like defects. The essence of the technique is based on the semi-empirical dependence of the Maxey fracture mechanics, the Folias value for a defect in the longitudinal direction of the wall of a cylindrical surface with internal pressure, the Dugdale model for the plastic zone and the experimentally (empirically) obtained values of the fracture stresses of damaged pipes, which eventually results in a relationship between the depth of the defect and the thickness of the pipeline wall (taking into account the outer diameter of the pipe).

The basic equation of the failure stress σ_f is calculated depending on length of the metal loss L, depth of the metal loss d, pipe wall thickness t, according to the expression:

- when the value $z = \frac{L^2}{dt} \leq 20$:

$$\sigma_f = \sigma_P \left[\frac{1 - \frac{2d}{3t}}{1 - \frac{2d}{3t}(M)^{-1}} \right], \tag{2.24}$$

where σ_F – flow stress, MPa, calculated by the formula:

$$\sigma_F = 1{,}1 \cdot \sigma_Y, \tag{2.25}$$

where σ_Y – specified yield strength, MPa.

- when the value $z = \frac{L^2}{dt} > 20$:

$$\sigma_f = \sigma_F \left[1 - \frac{d}{t} \right]. \tag{2.26}$$

The Folias factor M defined by the expression:

$$M = (1 + 0{,}8z)^{0{,}5}. \tag{2.27}$$

Later, at the time of the development of the engineering tool RSTRENG [14, 15, 16], which became the basis for the upgraded version of the B31G, the values of the Folias factor were clarified:

$$M = \begin{cases} (1 + 0{,}6275z - 0{,}003375z^2)^{0{,}5} \;\; for \;\; z \le 50 \\ 0{,}032z + 3{,}3 \;\; for \;\; z > 50 \end{cases}. \tag{2.28}$$

The standard of the American Society of Mechanical Engineers [17], applicable to almost all types of liquid hydrocarbons, uses the following methodology to determine the residual strength of the pipeline:

The dimensionless coefficient B is calculated according to Figure 2.5 or the formula:

$$B = \sqrt{\left(\frac{c/t}{1{,}1c/t - 0{,}15} \right)^2 - 1}, \tag{2.29}$$

where c – maximum flaw of the corroding zone, mm;

t – nominal wall thickness of the pipeline, mm.

After that, the maximum length of the defect in the axial direction is determined:

$$L = 1{,}12B\sqrt{Dt_n}, \tag{2.30}$$

where D – nominal outer diameter, mm.

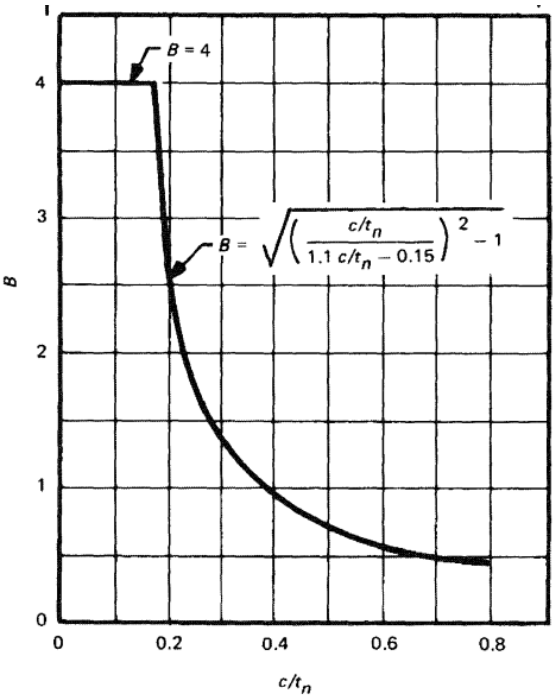

Figure 2.5 Parameters used in the analysis of the strength of corroded areas

At the end, a new reduced internal pressure P_d in the section is determined:

$$P_d = 1{,}1P_i\left[\frac{1-0{,}67\left(\frac{c}{t}\right)}{1-\frac{0{,}67c}{t\sqrt{G^2+1}}}\right], \tag{2.31}$$

where P_i – initial design pressure, psi

G – value that should not exceed 4.0 and calculated by the formula:

$$G = 0{,}893L/\sqrt{Dt}. \tag{2.32}$$

In case G is greater than 4:

$$P_d = 1{,}1P_i\left(1-c/t\right), \tag{2.33}$$

It is worth emphasizing that the method used in the above-described standard is almost identical to the method [11,12]. The main disadvantage is the lack of consideration of axial and bending stresses.

In turn, many standards of various countries for assessing the strength and reliability of pipelines when assessing the operability of pipe wires with defects refer to the documents considered in this paper. For example, the standard technical document of the Canadian Standards Association [18] for determining the residual strength of pipeline systems is based on documents [8,11,12,13,17].

The methodology for assessing the operability of pipelines with defects, presented in the British standard of BSI Standards Limited [19], is also based on the standards [8,13]. This technique is the most complete among all the above and takes into account the simultaneous impact of axial (including bending) and hoop stresses.

The above-mentioned technique is applicable to the assessment of various types of defect: cracks, dents, corrosion, erosion, stress corrosion, etc. Within the framework of the current work, the subject of interest are surface defects, which are considered in this standard in a separate section as Locally thinned areas (LTAs). As part of the LTA calculation, the following types of defects can be considered:

1) Internal corrosion;

2) External corrosion;

3) Corrosion in or near welds;

4) Interacting corrosion defects;

5) Defects loss of metal (burring or grinding).

The disadvantages of this technique include the inability to account for the following cases:

1) Materials with minimum yield strength exceeding 555 MPa or values of the yield strength to tensile strength ratio exceeding 0.93;

2) Cyclic loads;

3) Sharp defects (e.g. cracks);

4) Interaction of cracks and LTA;

5) LTA interaction and mechanical damage;

6) Defects of metal loss associated with mechanical damage (for example, recess);

7) Factory defects in the weld;

8) LTA defects with a depth of more than 80% of the pipeline wall;

9) LTA defects in stress concentration areas such as nozzles, tees, etc;

10) Areas subject to brittle fracture.

The technique follows the next algorithm shown in Figure 2.6:

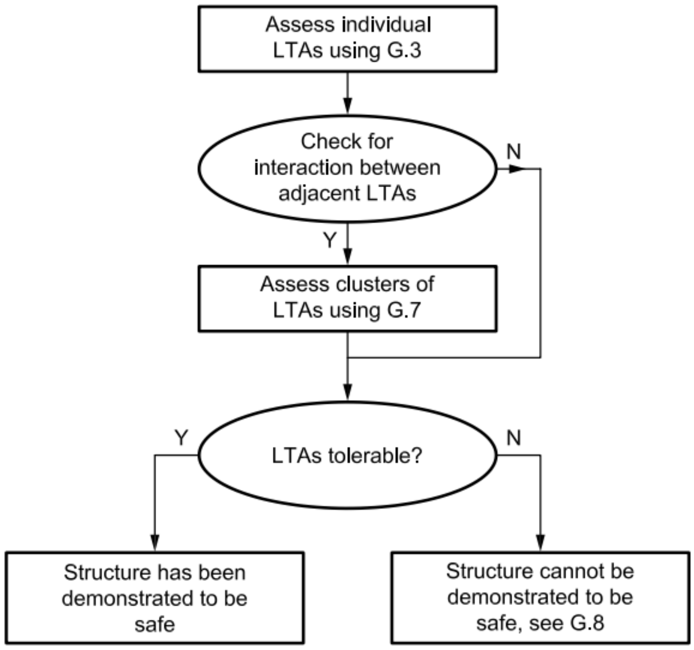

Figure 2.6 Block diagram of the evaluation procedure

The hoop stresses σ_{ref2} are determined by the formula:

$$\sigma_{ref2} = \left[\frac{1 - \left(\frac{a}{B} \right) \frac{1}{M}}{1 - \left(\frac{a}{B} \right)} \right] \sigma_2 , \qquad (2.34)$$

where $M = \sqrt{1 + 0,62 \left(\frac{c_1^2}{rB} \right)}$;

a – depth of the defect, mm;

B – nominal wall thickness of the pipeline, mm;

c_1 is half the length of the defect in the axial direction, mm (see Figure 2.7);

σ_2 – hoop stresses, MPa;

r – the outer radius of the pipeline, mm (see Figure 2.8).

R – bending radii, mm.

In turn , axial stresses are determined by the following formula:

$$\sigma_{ref1} = \left\{ \sigma_1 + 0,9 \left[\frac{r^2}{BR} \right]^{\frac{2}{3}} \left[\frac{4r}{\pi\left(r^4 - (r-B)^4\right)} \right] M_i \right\} \left[\frac{\pi\left(1 - \frac{a}{B}\right) + 2\frac{a}{B}\sin\left(\frac{c_2}{r}\right)}{\left(1 - \frac{a}{B}\right)\left(\pi - \left(\frac{c_2}{r}\right)\left(\frac{a}{B}\right)\right)} \right] \sigma_1 , \qquad (2.35)$$

where c_2 – half the length of the defect in the circumferential direction, mm;

σ_1 – axial stresses, MPa;

M_i – bending moment, MPa.

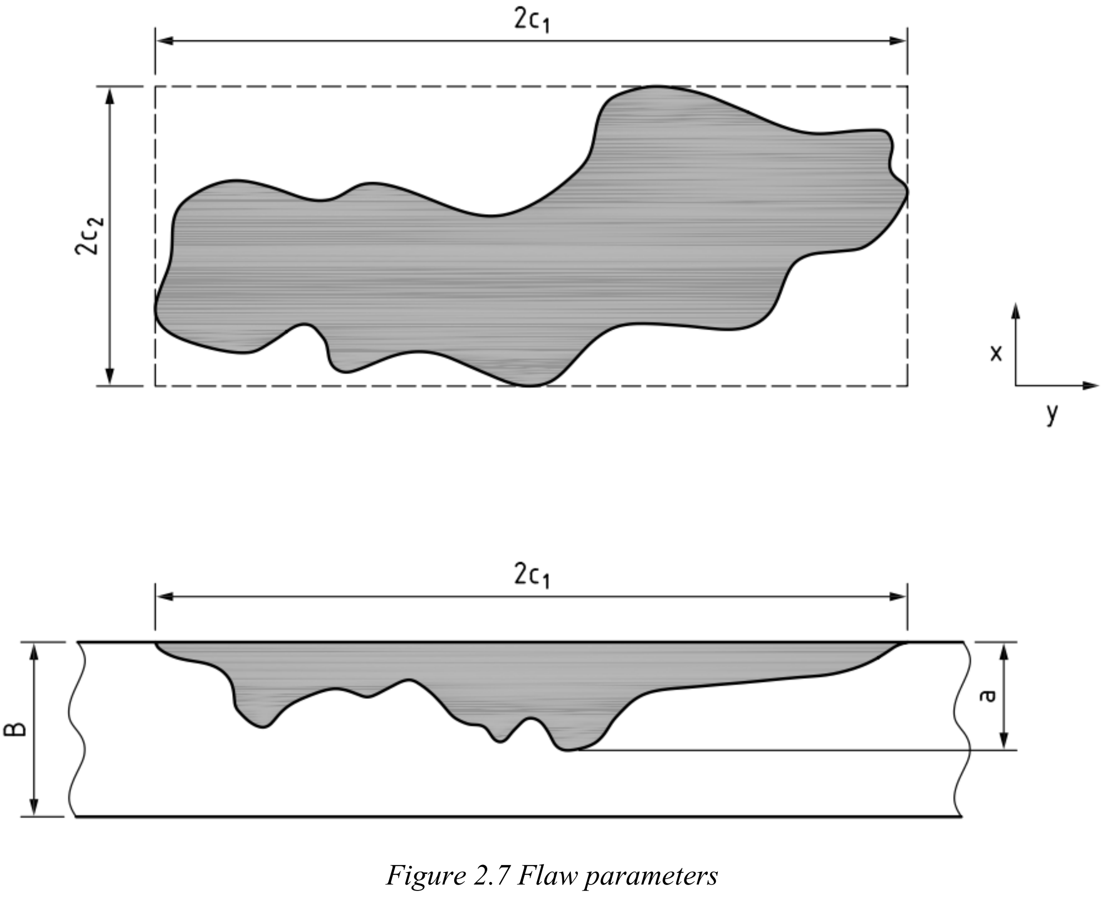

Figure 2.7 Flaw parameters

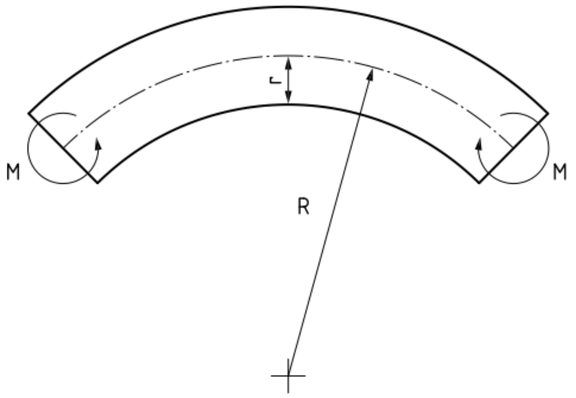

Figure 2.8 Bending parameters

The total stresses are taken as follows:

$$\begin{aligned} \sigma_{ref} &= \max(\sigma_{ref1}, \sigma_{ref2}) \;\; for \;\; \sigma_1 \geq 0 \;\; and \;\; \sigma_2 \geq 0 \\ \sigma_{ref} &= \sigma_{ref2} - \sigma_{ref1} \;\; for \;\; \sigma_1 < 0 \;\; and \;\; \sigma_2 > 0 \end{aligned} \quad . \tag{2.36}$$

The assessment according to this method is based on the plastic destruction of the pipeline. If the estimated evaluation point is below the limit value $L_r < L_{r,max}$, then the defect is acceptable, otherwise the defect is unacceptable:

$$L_{r,\max} = \frac{\sigma_Y + \sigma_U}{2\sigma_Y} , \tag{2.37}$$

where $L_{r,max}$ – maximum permissible load, MPa;

σ_Y – yield strength, MPa;

σ_U – ultimate tensile strength, MPa.

The current load L_r in the flaw is calculated using the following formula:

$$L_{r,\max} = \frac{\sigma_Y + \sigma_U}{2\sigma_Y} , \tag{2.38}$$

where f_c – safety factor;

In the standard of the American Petroleum Institute API 579-1/ASME FFS-1 2016 Fitness-For-Service [20], it is possible to evaluate many different defects, which also include the defects considered in this work, these are defects representing metal loss due to corrosion/erosion, mechanical damage, grinding/abrasion. Metal loss defects include:

a) Locally thinned area (LTA);

b) Groove-Like Flaw:

1) Groove – local elongated thin spot caused by directional erosion or corrosion; the length of the metal loss is much greater than the width.

2) Gouge – elongated local mechanical removal and/or relocation of material from the surface of a component, causing a reduction in wall thickness at the defect; the length of a gouge is much greater than the width and the material may have been cold worked in the formation of the flaw. Gouges are typically caused by mechanical damage, for example, denting and gouging of a section of pipe by mechanical equipment during the excavation of a pipeline.

The essence of the method is to find the reduced allowable working pressure of the pipeline section. The method consists of two levels of evaluation. The first level of assessment is supposed to be applied only in the case of loading the object under study only by internal pressure and when it is in non-cyclic operation (less than 150 cycles for the entire period of operation). The second level of assessment is applied in cases when the object under study is loaded with internal pressure and additional loads.

One of the alternative criteria for assessing residual strength in medium and high-strength pipelines with corrosion defects is the PCORRC criterion [21], developed by the Line Pipe Research Supervisory Committee of PRC International, which evaluates blunt defects (because of failure due to plastic collapse) through the maximum depth and maximum length of the defect. According to their analysis, the failure of pipes with such defects depends more on the ultimate tensile strength than on the yield strength or flow stress. In the obtained methodology, it is considered that the values of the length and depth of the defect are the most critical parameters in the evaluation, at a time when the width of the defect and the parameter of deformation hardening of the material are not such critical parameters.

According to PCORRC, the formula for calculating failure pressure is as follows:

$$p_b = \sigma_u \frac{2t}{D}\left((1-\frac{d}{t}(1-\exp(-0{,}157\frac{L}{\sqrt{R(t-d)}})) \right), \tag{2.39}$$

where R – radius of pipe, mm,

σ_U – ultimate tensile strength, MPa.

In addition, in this work, a parametric analysis of the effect of the geometric characteristics of the defect on the failure pressure was performed.

On the Figure 2.9 it is shown that the failure pressure decreases with increasing flaw size until a lower plateau is reached. For flaws of the uniform depth, the plateau pressure is proportional to the depth of the flaw. Unsurprisingly, this confirms that the flaw depth is the main variable of the flaw geometry controlling the fracture pressure of defects of the uniform depth. It should be noted, that in all parametric studies from this paper, the flaw was loaded only by internal pressure.

Figure 2.10 represents failure pressure for a series of rectangular uniform half of the wall thickness flaws. Each line shows a flaw with a different width. Again, it can be stated that according to this plot that length is the primary parameter that can change failure pressure almost 2 times. Another interesting thing is that the width of the flaw plays a secondary role. Changing the width of the flaw from 12,7 mm to 304,8 mm change failure pressure less than 5 percent.

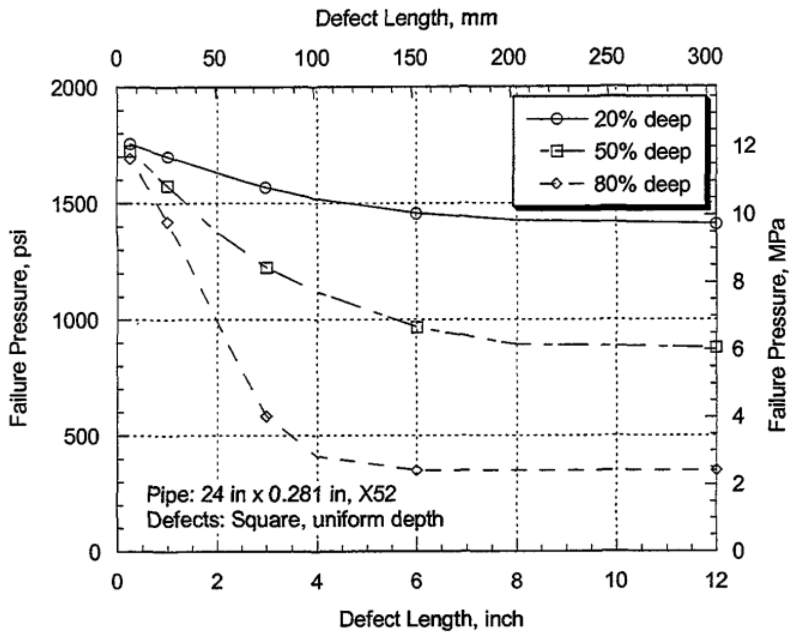

Figure 2.9 Comparison of the impact of flaw depth on failure pressure

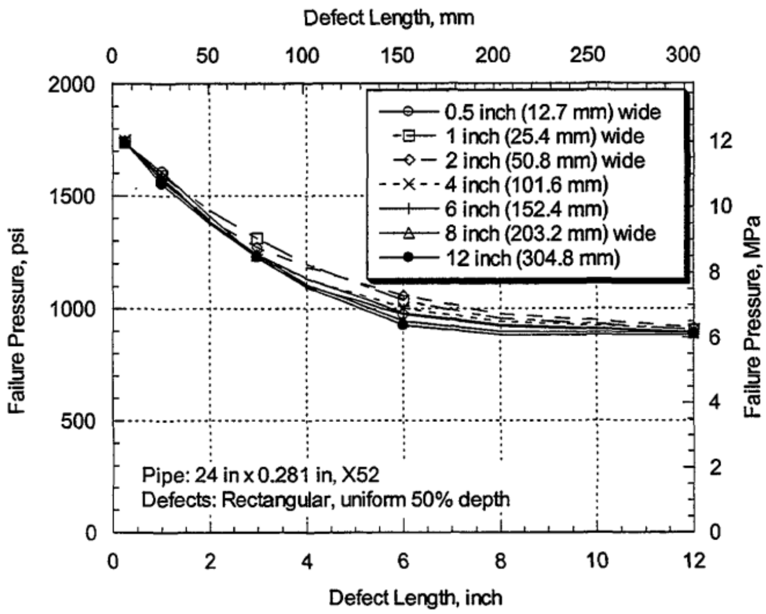

Figure 2.10 Comparison of the impact of flaw length and width on the flaw failure pressure

Defect shape has been also analyzed in this paper and is presented in Figure 2.11. Comparison of the results of the failure pressure for square uniform depth flaws and elliptical (in both axial and hoop direction) depth flaws. The results show that the parameter of defect shape is more important for deep flaws. As one can note, the difference in failure pressure between elliptical and square uniform depth flaws can reach almost 18 percent. That fact suggest that generally defect shape is more important than flaw width, especially for deeper flaws.

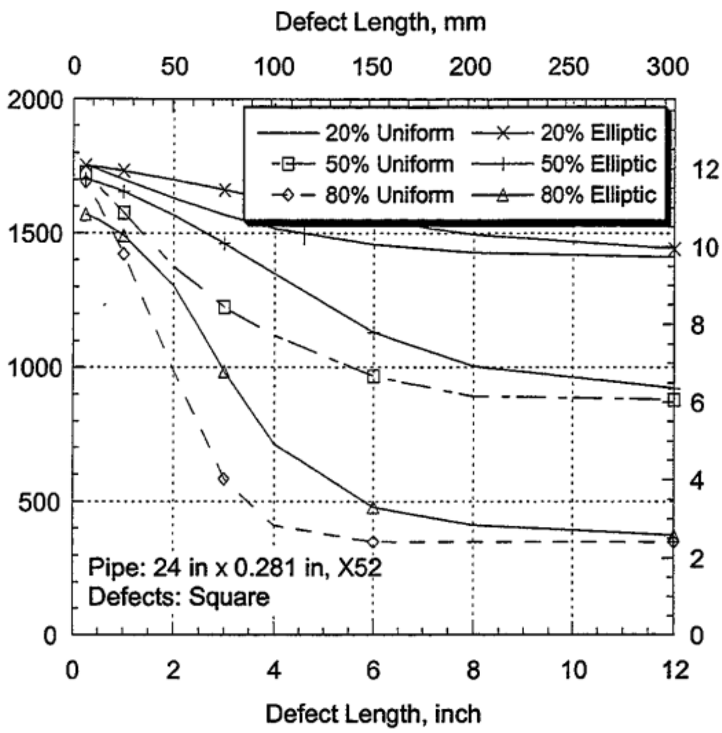

Figure 2.11 Evaluation of the influence of uniform depth and elliptic shape on flaw failure pressure

2.5 Theoretical methods for assessing the operability of trunk pipelines with flaws

In [22] the author has developed a method of hydraulic testing of gas pipes, which allows determining the real coefficients of the safety margin of defective pipes and a method for assessing the influence of geometric parameters of the defect on the stress concentration coefficient. However, the disadvantages of this work can again be attributed to the lack of consideration of bending, since the stress-strain state and the maximum stress concentration coefficient were determined by changes in the pressure and geometric parameters of the defect.

In [23] results were obtained indicating that at a large depth of the defect (about 50% of the value of the pipe wall), the effect of the width of the defect on the value of the stress concentration coefficient is weaker. Comparisons of the data obtained in the work with the rejection standards make it clear that there is a significant reinsurance in the size of permissible defects, which is most likely due to the lack of consideration of longitudinal and bending loads.

In [24], the dependences of the amount of accumulated plastic deformation for a gas pipeline in the defect zone on the action of annular stresses caused by internal pressure were obtained. However, these dependencies do not take into account the elastic bending of the pipeline and can be used only in rare cases, for example, on sections of crossings.

The authors of [25] propose their mathematical model for assessing the durability of pipes with corrosion loss of metal and crack-like corrosion-mechanical defects. The resulting model allows to build diagrams to assess the strength and durability of pipes with defects. However, during the calculation, the authors note that the magnitude of temperature stresses and bending radius are set to zero in the calculations, which is a clear disadvantage of this model.

Abroad, Kieffer, J. F. [26] was one of the first to conduct studies of the residual strength of the pipeline with corrosion. The authors conducted a number of subsequent studies in which the failure pressure of the pipeline sections with corrosion was evaluated using real experiments and numerical analysis methods. These studies have become the basis for the normative documents used in the oil and gas industry to assess the working capacity of pipelines with corrosion defects. One of the first such documents was the standard of the American Community of Mechanical Engineers [13].

In [27], a mathematical model has been developed that determines the criterion for failure of pipelines with a defect that can easily be applied in operational setting. The authors proved that for any pipe, a rupture occurs when the hoop stresses reach a value of 2/3 of the tangential stiffness. In addition, the model shows that an important parameter characterizing the defect is the ratio of elastic modulus and plasticity in the thin and thick section of the defect. It was also found that the circumferential magnitude of the defect affects the failure pressure, but it has a greater effect on the magnitude of the circumferential deformation during fracture. It is worth noting that in this paper only the hoop stresses of the pipe are considered.

In [28], the results of 79 rupture tests and many simulation cycles were used to obtain the failure pressure of high-strength pipelines with external corrosion-type defects. Based on the results of the study, it can be concluded that the failure pressure of the pipeline: a) decreases with an increase in the ratio of the depth of the defect to the value of the pipeline wall *d/t* for all defects; moreover, when *d/t* is a small value, the fracture pressure decreases more slowly; b) decreases with decreasing $\frac{L}{\sqrt{d \cdot t}}$ for all defects, where L is the length of the corrosion defect; moreover, when $\frac{L}{\sqrt{d \cdot t}}$ is a small value, the failure pressure decreases more slowly.

In [29], a comparative analysis of the reliability of pipelines with and without corrosion defects in cases of low- and high-strength pipelines (>X80) was carried out. The study revealed that in cases of high-strength pipelines, due to less deformation hardening (due to high pressures), the susceptibility to internal corrosion increases. In addition, the authors claim that an increase in the ratio of yield strength to tensile strength may lead to greater risk (under high pressure loading). This is due to the fact that the plasticity of high-strength steels is locally limited, since these types of steel cannot exhibit the same plasticity in areas remote from the defect. Based on this, it can be concluded that the failure pressure and reliability are decreasing.

The authors of [30] obtained a solution for a cylindrical shell of pipeline (with a diameter of D and a wall thickness t) made of a rigid plastic material with axisymmetric thinning of the pipeline wall in the presence of a defect. The shape of the damage is presented in the form of a rectangular groove with a maximum depth of h_{max}, the length of the defect L and the residual thickness of the pipe wall in the defect zone t_d. In this paper, the main parameter of strength reduction is the coefficient φ_T determined by the formula:

$$\varphi_T = 0,5\left(t_d - \frac{n}{\lambda^2}\right) + \sqrt{0,25\left(t_d - \frac{n}{\lambda^2}\right)^2 + \frac{n}{\lambda_2}}, \tag{2.40}$$

where R_c – average value of the radius of the circle describing the profile of the corrosion defect and determined $R_c = \left(D_н - t\right)/2$;

t_d is calculated accordingly:

$$t_d = t_d / t \quad (2.41)$$

λ_2 is calculated by the formula:

$$\lambda_2 = 1{,}82 \frac{(L/2)}{\sqrt{R_{cp} t}} \quad (2.42)$$

In [31,32], a cross-section with the maximum depth of the corrosion defect was selected for calculation. The solution took into account all components of the complex-stressed state of the pipeline (hoop stresses σ_h, radial stresses σ_r, and the sum of longitudinal stresses from internal pressure, elastic bending and temperature drop σ_a).The above stress parameters were described using the following functional dependencies:

$$\sigma_{h\ def} = f\left(t_{def}; D / D_i; \ldots\right) \quad (2.43)$$

$$\sigma_{a\ def} = f\left(t_{def}; D; \rho; \Delta t; \ldots\right) \quad (2.44)$$

$$\sigma_r = -P \quad (2.45)$$

In [33], studies are conducted to assess the reliability of corroding pipelines in a complex state. In the course of the research, a complex three-dimensional finite element model is being developed that can simulate a pipeline with corrosion defects in a complexly stressed state. The model was verified using 13 experiments of pipeline destruction with artificially inflicted corrosion defects.

However, the results obtained turned out to be overestimated due to the simplification of the defect schematization and taking into account the complex loading of the pipe body with strict pipe fixing conditions. So, for example, according to this method, the pressure of destruction of a pipeline section turned out to be 1.7 times higher than at the actual pressure of destruction of this section.

As expected, with an increase in the depth of corrosion, the failure pressure decreased, as well as maximum moment capacity decreased with all other parameters constant. In turn, an increase in the longitudinal magnitude of the defect reduced the value of the failure pressure. However, interesting results of the influence of the defect width were also obtained: for a defect with a length of 18 inches and a width of 18 inches, the failure pressure was less than for a defect with a length of 30 inches and a width of 6 inches (with the same depth of defects). It was found out that the circumferential location of the defect relative to the tension side or the compression side of the pipe has a strong influence on the moment of destruction of the pipe, but at the same time this effect is practically irrelevant in the case of considering a longitudinally located defect.

When studying the influence of axial stresses from thermal expansion, it was found that these stresses have a slight effect on the failure pressure, but significantly reduce the maximum moment capacity of corroded section. The last effect described above is again more pronounced in the case of a circumferentially located defect.

2.6 Comparative analysis of existing methods

The authors of [34] have developed their own methodology for assessing the risk of defects, confirmed by 41 rupture tests and 58 stages of finite element modeling. In addition, their results were compared with the experiments of [35], which showed that the difference in results is less than 5%.

The basic equation of failure pressure is calculated by the expression:

$$p_f = \frac{t}{R}\frac{\sigma_{cr}}{f_\theta}, \tag{2.46}$$

where σ_{cr} – failure stress, MPa;

f_θ – stress concentration coefficient.

In addition, a comparative analysis was carried out with the main standards and codes for assessing the residual life of pipelines with corrosion: ASME B31G [13], modified ASME and DNV RP-F101 [8]. According to the analysis obtained, it can be concluded that with respect to the developed analytical solution, DNV RP-F101 shows identical results in the case of a defect depth of more than half of the pipeline wall. In turn, ASME B31G and modified ASME provides unsafe results in the case of deep defects. It is also worth noting that ASME B31G and DNV RP-F101 underestimate the residual strength of shallow defects. In general, the paper noted good convergence of the results of DNV RP-F101. ASME B31G showed results that were excessively conservative. In turn, the use of modified ASME is not recommended at all for the assessment of deep defects, especially in the case of high-strength steels. In addition, neither DNV nor ASME provide reliable results for defect geometries representing circumferential grooves.

The authors of the work [36] analyzed the most popular methods from the industry for assessing the residual life of pipelines with defects: ASME B31G [13], DNV RP-F101 [8] and PCORRC [21]. In the same way as in many of the above works, experimental tests were carried out for the rupture of a pipe with a defect in the number of 67 experiments, which became the basis for a finite element (FE) model. Subsequently, based on experiments and the FE model, a new failure pressure formula was obtained, which includes such defect parameters as: length, width and depth. Comparison of the results according to this formula with known standards from the industry showed satisfactory accuracy.

According to this work, a new formula for calculating the failure pressure was obtained:

$$p_f = \frac{2\sigma_U t}{D}\left[1-\frac{d}{t}\left(1-\left(0,1075\left(1-\left(\frac{w}{\pi D}\right)^2\right)^6+0,8925\exp\left(\frac{-0,4103L}{\sqrt{Dt}}\right)\right)\left(1-\frac{d}{t}\right)^{0,2504}\right)\right], \qquad (2.47)$$

where σ_U – tensile limit of pipe material, MPa;

L – defects length, mm;

d – defect depth, mm;

w – defects width, mm;

D – diameter of pipe, mm;

t – thickness of pipe, mm.

In this paper the authors also made comparison review of the most popular methods and it is concluded that the failure pressure predicted by methods described in DNV-RP-F101 [8] is un-conservative mainly for low-grade steels according comparison results while PCCORC shows 12 un-conservative results. However, ASME B31G [13] is more conservative than the other methods. Hence it can be concluded that ASME B31G is more suitable for low grade steel pipes and overall is the most conservative method. DNV-RP-F101 and PCCORC are good to evaluate high grade steel pipes and it is noted that PCCORC better than DNV-RP-F101.

Paper [37] studies influence of external corrosion on the residual strength of the pipeline as a subject only to internal pressure. The paper presents small-scale experiments and numerical modeling methods. Corrosion defects were considered as LTA and for conducting experiments, defects on pipes were applied artificially. Figure 2.12 shows the defect applied to the pipe, obtained using the spark erosion process.

Figure 2.12 Test specimen with an induced defect

During the modeling process, two defect models were created. One was a rectangular shape – simplified defect shape model (SDS), which is almost always used in regulatory procedures, such as, for example, ASME B31G. The second defect model completely repeated the shape of experimental defects – and the exact defect shape model (EDS). As expected, the simulation results showed that the fracture pressure for the numerical model with SDS for all cases turned out to be less than for EDS. This happens simply due to the fact that less material remains in the defect zone.

Next, the SDS model was used for parametric analysis and the results presented in Figures 2.13 and 2.14 were obtained.

Again, as in the previous study, one can note, that the depth of the flaw is the most significant parameter that has impact on the failure pressure.

If the ratio of the flaw depth to the wall thickness *d/t* is less than 2, a slight decrease in the failure pressure can be observed (no more than 5%). However, with an increase in the depth of the defect, this effect becomes more significant, achieving a decrease in the failure pressure by 75% for the ratio *d/t* = 0.8. For the length parameter, the opposite effect can be observed. The effect of length on the failure pressure decreases as the length of the defect increases. As a result, a plateau is reached between the values of the ratio of the flaw length to the pipe diameter *l/D* in 1.5D and 2D, which means that the effect of the defect length is practically insignificant for the ratio *l/D* greater than 1.5.

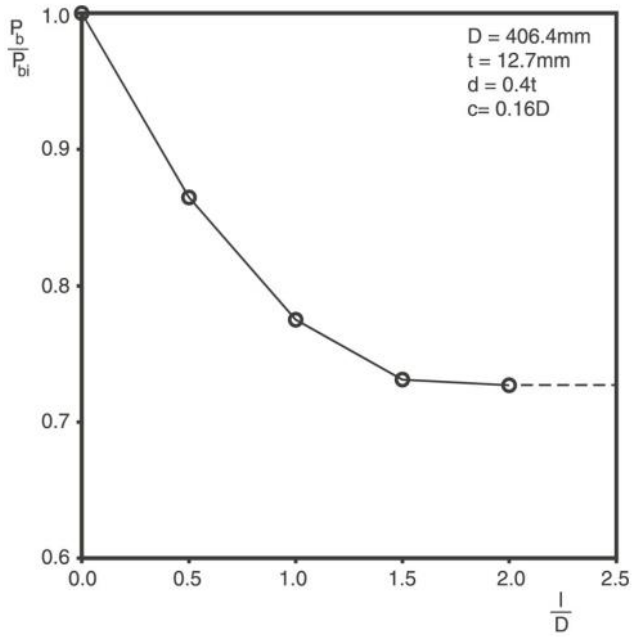

Figure 2.13 Failure pressure versus length of the flaw

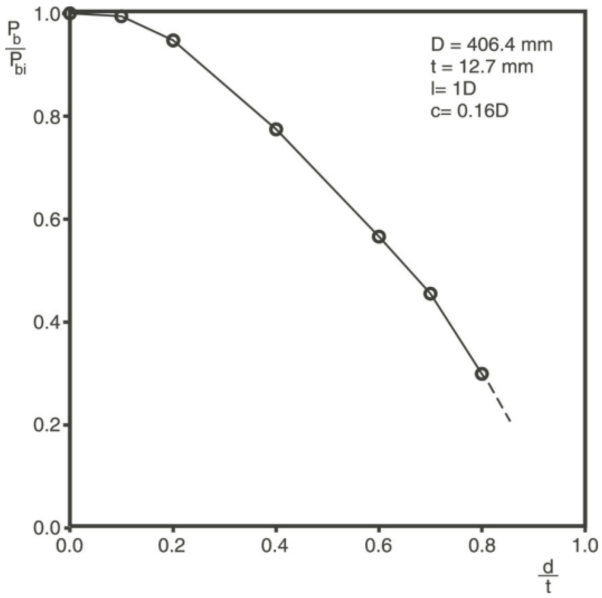

Figure 2.14 Failure pressure versus depth of the flaw

Based on the results obtained, it can also be concluded that an increase in the width of the flaw practically does not affect the value of failure pressure in the range of values of the ratio of the width of the defect to the outer diameter of the pipe $0.0785 \leq c/D \leq 0.1571$. Taking into account such data, it can be concluded that the value of failure pressure changes practically or not much when the following values of the defect parameters are exceeded: $c/D = 0.0785$ and $l/D = 1.5$.

After calibration based on the experimental results, a numerical model was used to determine the failure pressure depending on the material and geometric parameters of various pipes and flaws. The result of this process produced the following equation of the failure pressure Pb for c / D ≥ 0.0785, 0.1 ≤ d / t ≤ 0.8, and l / D ≤ 1.5:

$$\frac{P_b}{P_{bi}} = 1 - 0,9435\left(\frac{d}{t}\right)^{1,6}\left(\frac{l}{D}\right)^{0,4}, \tag{2.48}$$

where P_{bi} – failure pressure of intact pipe, MPa.

The last, but not least, authors made comparison evaluation of the linear fit of the empirical function and predictions obtained by the equations from DNV [8] and ASME B31G [13]. Analyzing results from the Figure 2.15 it can be observed that B31G code provide quite over-conservative results. Even more conservative results were obtained using the DNV method.

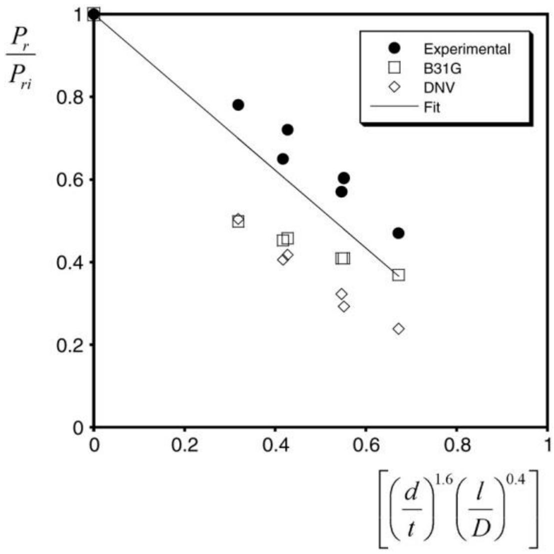

Figure 2.15 Evaluation between experiments, the obtained new method and methods from B31G and DNV codes

Aim of the work [38] is to investigate key parameters of corrosion flaw that has influence on residual strength of the pipeline under combined internal pressure and bending moment. Analysis techniques based on numerical analysis and empirical formula was used and comparisons were made in terms of ultimate burst pressure prediction.

From the Figure 2.16 and Figure 2.17 it can be seen that increase of the thickness significantly influence failure pressure and bending moment capacity. However, in this study the effect of axial load is not investigated.

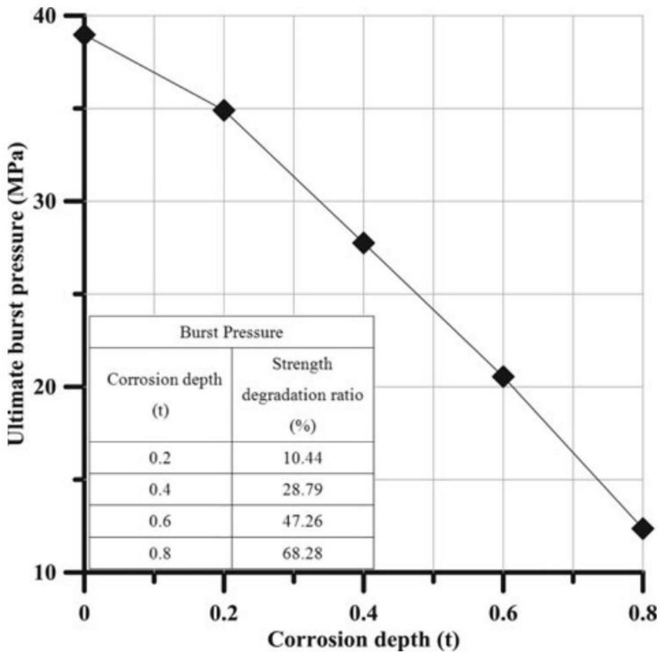

Figure 2.16 Impact of the flaw depth on maximum failure pressure

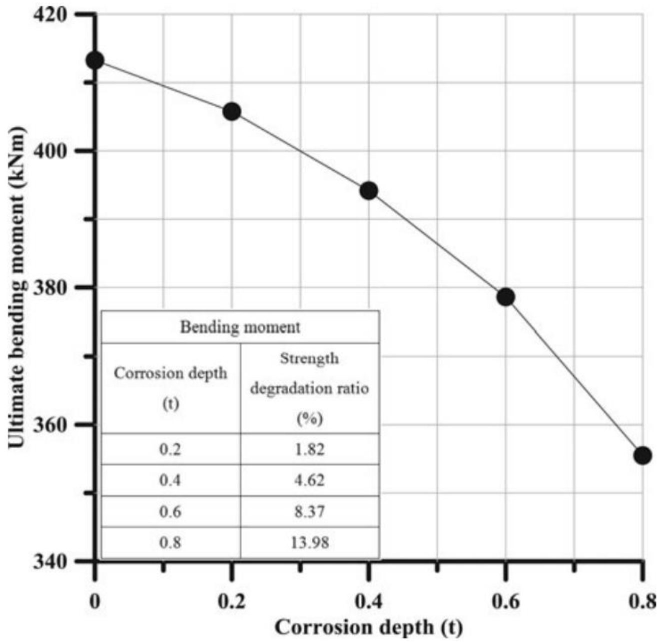

Figure 2.17 Impact of the flaw depth on maximum bending moment

In the study [39] numerical model for predicting the failure of pipelines with corrosion flaws is constructed using the nonlinear finite element method, in which technical aspects are recommended, including an element mesh, a model of materials, a nonlinear solution and a failure criterion. Using this model, full-size experiments on the failure of pipes made of various materials, sizes and flaws were analyzed and computed. Based on the calculation results of the model, a new formula for calculation failure pressure is proposed. It should be noted, that in comparison with some methods, this solution contain consideration of the width of the flaw, but no assessment of the axial loads. The failure pressure p_b formula was selected based on the test results of 39 pipes:

$$p_b = \frac{2\sigma_b t}{D}\cdot\left\{1-\frac{d}{t}+\frac{d}{t}\cdot\left[\left(0,0973\cdot\left(1-\left(\frac{w}{\pi D}\right)^2\right)^6+0,903\cdot\exp\left(\frac{-0,474L}{\sqrt{Rt}}\right)\right)\cdot\left(1-\left(\frac{d}{t}\right)^{4,491}\right)\right]\right\} \quad (2.49)$$

Where σ_b – tensile limit of pipe material, MPa;

L – flaw length, mm;

d – flaw depth, mm;

w – flaw width, mm;

D – diameter of pipe, mm;

t –wall thickness of pipe, mm.

In addition, the results of 39 pipes were computed by using DNV-RP-F101 [8], ASME B31G [13], PCORRC [21] methods. Authors performed error analysis of the above methods (results are shown in Table 2.1) and it can be concluded that proposed fitting formula is better that above national codes. Another thing that one can note, that among all codes DNV and PCORRC are the most accurate methods.

Table 2.1 Calculation errors by the four methods

Calculate method	DNV	PCORRC	B31G	Proposed fit formula
Max error	-23,54	-22,89	33,52	-13,57
Min error	0,26	0,53	-0,66	0,4
Square sum of residual error	102,09	80,21	303,9	19,69

Finally, all four methods were used to evaluate the impact of the parameters of the flaw (length, width and depth) on the failure pressure of the corroded pipe, which is shown in Figure 2.18 – 2.20.

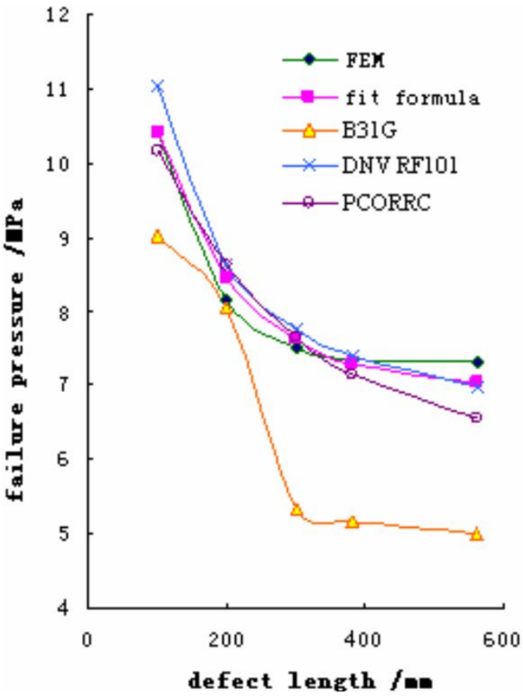

Figure 2.18 Assesment of the influence of the defect length on the failure pressure

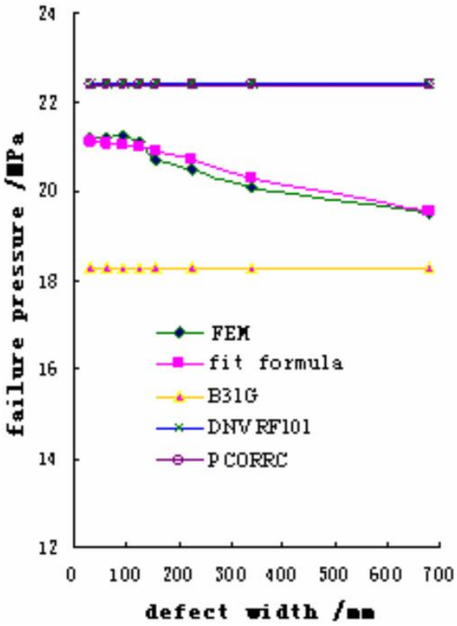

Figure 2.19 Assessment of the influence of the defect width on the failure pressure

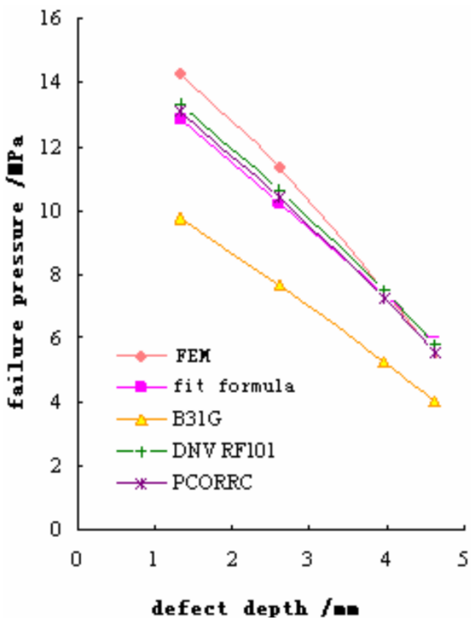

Figure 2.20 Assessment of the influence of the defect width on the failure pressure

Firstly, fit formula predicts the closest to FEM (Finite Element Method) result. Also in all three parametric studies ASME B31G code provides over-conservative results as well as in the previous studies. Last, but not least, DNV and PCORRC show quite satisfied results when considering the flaws length and depth. However, these two methods underestimate the effect of the width on the failure pressure.

Therefore, considering all above sources it might be concluded that DNV-RP F 101 provides the most comprehensive approach with quite precise results among all standards and codes, but it still has unaccounted cases that can be improved.

2.7 Conclusions. Scope and objectives of the study

As a result of the analysis of methods for assessing the operability of pipelines with defects in a complex state, conclusions are made:

1. DNV-RP F 101 [8] provides the most comprehensive approach with quite precise results among all standards and codes, but it still has unaccounted cases that can be improved. Particularly, the case of interacting flaws subjected to both internal pressure and longitudinal stresses is not considered in the document. Moreover, this document is the basis for Gazprom's Russian standard. However, in comparison with DNV-RP F101, Russian analogue accounts for both tensile and compressive stresses when analyzing single corrosion flaw while DNV can estimate only compressive stresses.

2. Neither document or code (DNV and STO Gazprom) take into account defects location relative to the impact zone of compressive/tensile longitudinal stresses.

Research objectives:

1. Analyze the available standards, codes and methods for the assessment of trunk pipelines with corrosion flaws.

2. Create numerical model for the assessment of the operability of pipeline with corrosion flaws on the basis of the Ansys software package.

3. Based on Ansys software package prove the importance of taking into account bending stresses.

4. Improve the methodology for assessing interacting flaws subjected to both internal pressure and longitudinal stresses, taking into account flaws location relative to the impact zone of compressive/tensile longitudinal stresses.

Chapter 3

Improvement of the methodology for determining the operability of a pipeline with interacting defects subjected to combined loading

3.1 Calculation of the operability of a pipeline section with defects according to the method DNV and Gazprom

First, the guidelines for assessing the defect and the load only from internal pressure are considered.

To perform the analytical calculation, a pipeline with the following parameters was selected (Table 3.1) and a defect (Table 3.2).

Table 3.1 Pipe parameters

Name of the parameter	Value
Material	X70
Strength class	K60
Wall thickness, mm	19,5
Diameter, mm	1420
Model length, mm	4000
Tensile strength, MPa	570
Yield strength, MPa	460

Table 3.2 Defect parameters

Name of the parameter	Value
Defect length, mm	117
Defect width, mm	50
Defect depth, mm	8

First of all , the safety margin *K* is calculated according to the formula 2.7:

$$K = \frac{0,9 \cdot 0,982 \cdot 1,1 \cdot 1,34 \cdot 1,21}{0,99} = 1,59.$$

where γ and R_1 is calculated according to the formula 2.8 and 2.9:

$$\gamma \approx 1 - \frac{1,1 \cdot 5,5}{348} = 0,982,$$

$$R_1 = \frac{0,99}{1,34 \cdot 1,21} \cdot 570 = 348 \ MPa.$$

Following the positions [5,8] sequentially calculate:

- Coefficient *Q* according to the formula:

$$Q = \sqrt{1 + 0,31\left(\frac{l}{\sqrt{Dt}}\right)^2}, \tag{3.1}$$

where l – length of the flaw in the axial direction, mm;

D – outside diameter of the pipe, mm;

t – pipe wall thickness, mm.

$$Q = \sqrt{1 + 0,31\left(\frac{117}{\sqrt{1420 \cdot 19,5}}\right)^2} = 1,07.$$

- Failure pressure according to the formula:

$$p_f = \frac{2t\sigma_U\left(1-\frac{d}{t}\right)}{(D-t)\left(1-\frac{d}{tQ}\right)}, \tag{2.51}$$

where σ_U – tensile strength of the pipe material, MPa;

d – maximum value of the defect depth, mm.

$$p_f = \frac{2\cdot 19{,}5\cdot 570\cdot\left(1-\frac{8}{19{,}5}\right)}{(1420-19{,}5)\left(1-\frac{8}{19{,}5\cdot 1{,}07}\right)} = 15{,}1\ MPa.$$

- Permissible pressure according to the formula 2.10:

$$p_p = \frac{15{,}1}{1{,}59} = 9{,}49\ MPa.$$

Further, methodological guidelines for assessing the operability of a pipeline with a defect are considered when taking into account loads from internal pressure and axial loads.

The values of the failure pressure only from the internal pressure, the reserve coefficient K and Q are taken from the previous calculation.

Following the provisions, sequentially calculated:

- Ratio of the length of the defect in the circumferential direction to the nominal outer perimeter of the pipe:

$$\theta = \frac{c}{\pi D}, \tag{3.2}$$

where c – defect width, mm.

$$\theta = \frac{50}{\pi\cdot 1420} = 0{,}011.$$

- Condition for various options for calculating the G_2 coefficient is checked:

$$\theta < \frac{1}{\left(2-\frac{d}{t}\right)}, \tag{3.3}$$

$$\theta < \frac{1}{\left(2-\frac{8}{19,5}\right)} = 0,629.$$

- Coefficient G, which corresponds to the minimum of the two values G_1 and G_2:

$$G = \min(G_1; G_2).$$

- Coefficients G_1 and G_2 are calculated taking into account the value of θ according to the equations:

$$G_1 = 0,5\left(1-\frac{d}{t}\theta\right)\left(1+\frac{\sigma_Y}{\sigma_U}\right), \tag{3.4}$$

$$G_2 = \frac{2}{\pi}\left(1+\frac{\sigma_Y}{\sigma_U}\right)\left\{\cos\left[\frac{d}{2t}\theta\pi\right]-\frac{d}{2t}\sin(\theta\pi)\right\}, \tag{3.5}$$

where σ_Y – yield strength of the pipe material, MPa.

$$G_1 = 0,5\left(1-\frac{8}{19,5}\cdot 0,011\right)\left(1+\frac{460}{570}\right) = 0,899,$$

$$G_2 = \frac{2}{\pi}\left(1+\frac{460}{570}\right)\left\{\cos\left[\frac{8}{2\cdot 19,5}\cdot 0,011\cdot\pi\right]-\frac{8}{2\cdot 19,5}\sin(0,011\cdot\pi)\right\} = 1,14,$$

$$G = \min(0,899; 1,14) = 0,899.$$

- Checking the need to take into account the tensile stresses according to the condition:

$$\sigma_a > \sigma_2, \tag{3.6}$$

where σ_2 is defined according to the formula:

$$\sigma_2 = \sigma_U\left[G - 0,5\frac{\left(1-\frac{d}{t}\right)}{\left(1-\frac{d}{tQ}\right)}\right], \tag{3.7}$$

$$\sigma_2 = 570\cdot\left[0,899 - 0,5\frac{\left(1-\frac{8}{19,5}\right)}{\left(1-\frac{8}{19,5\cdot 1,07}\right)}\right] = 240,65\ MPa.$$

σ_a is calculated according to the formula 2.6:

$$\sigma_a = 0,3 \cdot 247,87 - 1,2 \cdot 10^{-5} \cdot 2,05 \cdot 10^5 \cdot (-60) + \frac{2,05 \cdot 10^5 \cdot 1420}{2 \cdot 500 \cdot 1420} = 426,96 \; MPa.$$

$$426,96 \; MPa > 240,65 \; MPa.$$

- Checking the condition for different variants of calculating the H_2 coefficient:

$$\theta \ge \frac{1}{\pi} \arcsin \left[\frac{\sigma_b}{\sigma_Y + \sigma_U} \frac{\pi}{2 - \frac{d}{t}} \right], \tag{3.8}$$

where σ_b – axial stresses from elastic bending, MPa,

$$\theta \ge \frac{1}{\pi} \arcsin \left[\frac{205}{460 + 570} \frac{\pi}{2 - \frac{8}{19,5}} \right] = 0,128.$$

- Coefficient H_2, taking into account the influence of tensile longitudinal loads in the pipe wall:

$$H_2 = \left(1 + \frac{\sigma_Y}{\sigma_U}\right) \left\{ \left(1 - \frac{d}{t}\theta\right) - \frac{2\sigma_{\Delta t,p}}{\sigma_Y + \sigma_U} - \frac{2}{\pi}\left(1 - \frac{d}{t}\right) \arcsin \left[\frac{\left[\frac{\sigma_b}{\sigma_Y + \sigma_U}\right] \frac{\pi}{2} + \frac{d}{2t} \sin(\theta\pi)}{\left(1 - \frac{d}{t}\right)} \right] \right\}, \tag{3.9}$$

where $\sigma_{\Delta t,p}$ – longitudinal stresses from temperature difference and internal pressure, MPa.

$$H_2 = \left(1 + \frac{460}{570}\right) \left(\left(1 - \frac{8}{19,5} \cdot 0,011\right) - \frac{2 \cdot 221,96}{460 + 570} - \frac{2}{\pi}\left(1 - \frac{8}{19,5}\right) \cdot \right.$$
$$\left. \cdot \arcsin \left[\frac{\left[\frac{205}{460 + 570}\right] \frac{\pi}{2} + \frac{8}{2 \cdot 19,5} \cdot \sin(0,011 \cdot \pi)}{\left(1 - \frac{8}{19,5}\right)} \right] \right) = 0,445.$$

- Failure pressure according to the equation:

$$p_f = \frac{2t\sigma_U\left(1-\frac{d}{t}\right)}{(D-t)\left(1-\frac{d}{tQ}\right)}H_2, \tag{3.10}$$

$$p_f = \frac{2\cdot 19{,}5\cdot 570\cdot\left(1-\frac{8}{19{,}5}\right)}{(1420-19{,}5)\left(1-\frac{8}{19{,}5\cdot 1{,}07}\right)}\cdot 0{,}445 = 6{,}56\ \ MPa.$$

As a result of the calculation, minimum value of failure pressure from both cases is taken as the failure pressure and the permissible pressure for the analysed section of the pipeline is determined by the formula 2.10:

$$p_p = \frac{6{,}74}{1{,}59} = 4{,}23\ \ MPa.$$

Considering two calculations for a single defect with and without taking into account axial stresses caused by internal pressure, temperature drop and bending loads, it is worth noting that the permissible pressure has decreased by more than 2 times. Based on this, we can draw an unambiguous conclusion about the need to take into account axial and bending loads. However, in this technique, for interacting defects, there is only a calculation taking into account the internal pressure. This aspect is discussed in the next section.

3.2 Analysis of the operability of a pipeline section with defects using numerical methods of finite element modeling

3.2.1 Methodology concept

In the process of the finite element analyses (FEA) several simplifications were allowed. First, as it can be seen from Figure ## the model of the corrosion flaw is assumed to be rectangular shape. This configuration of shape might seem a bit conservative (just because of more material is present in the flaw), but it is a common practice for the failure assessment of the corrosion flaws. In fact, a comprehensive DNV corrosion assessment method [8] has uses the idealization of the rectangular shape of the defect to predict the failure pressure. In addition, using rectangular shape allows failure pressure prediction for lower bound (safer margin) and it has been proven to give reasonable accuracy [40-44].

Second, in order to reduce the computer processing time as much as possible with an adequate accuracy result, the advantage of using a symmetry model was considered. Symmetry model reduce the size of node, element, and mesh, which leads to reduced computing time. This similar methodology of using a smaller (quarter) model has been used by Belachew et al. [45].

ANSYS 2020 R2 Workbench a widely-employed FE software for studying integrity of pipes was utilized, which henceforth will be referred to as ANSYS. With the usage of quarter pipe models, appropriate boundary conditions were applied at the required areas of the model, so that the quarter model would be treated as a whole-pipe model during the FEA. The degrees of freedom (DOF) of the pipe was constrained to prevent unwanted rigid body motion. The quarter model was constrained in the x, y and z directions on a node. The load applications were controlled via timestep in the static non-linear analysis. The axial stress is incrementally applied during the first time step. Beyond that, the axial stress is maintained, while internal pressure is incrementally applied. Timestep divisions were set to 500 to allow for accurate FEA, as well as to prevent convergence failure due to the elements undergoing high deformation.

3.2.2 Methodology verification

3.2.2.1 Validation against full-scale burst (internal pressure only)

Table 3.3 Corrosion flaw parameters for full-scale burst tests

Sample	Flaw configuration	depth, mm	length, mm	width, mm	S_L, mm	S_C, mm
1	1 flaw	5,39	39,6	32	-	-
2	2 axial flaws	5,32	39,6	32	20,5	-
3	2 circumferential flaws	5,62	39,6	32	-	9,9

Results from full-scale burst test conducted by Benjamin et al. [46] were used to validate the FEM for cases where internal pressure only were concerned. Before moving onto the methodology validation, a brief description of the full-scale burst test will be outlined. The full-scale burst tests were conducted on API 5L X80 steel pipes. The pipe specimens had a nominal outer diameter of 458.8 mm, nominal wall thickness of 8.1 mm, and length of 1.7 m. Table 3.3 lists the dimensions, orientation and type of interactions of the artificially machined corrosion defects, where S_L is the longitudinal distance between interacting defects and S_C is the circumferential distance between interacting defects.

These corrosion defects were modelled, and the loads were applied as per previously described methodology for internal pressure application only. Table 3.4 shows the tabulation of failure pressures from the full-scale burst test, FEA failure pressure predictions, and the percentage difference between both failure pressure values. The comparison shows a good correlation between both methods, with the deviation around 2,5% and 4% for those three cases.

Table 3.4 Failure pressures of the pipe segments with corrosion flaws

Sample	Burst test failure pressure, MPa	FEA failure pressure, MPa	Absolute percentage difference, %
1	22,68	22,1	2,5
2	20,31	19,7	3
3	21,14	22	4

3.2.2.2 Validation against thin-walled pressure vessel theory

The bending moment characteristics of corroded pipelines can be divided into two categories depending on the type of load applied. If the corroded area is subjected to a tensile load, this will lead to the opening of the bending and, accordingly, vice versa. The type of bending for corroded pipelines is shown in Figure 3.1.

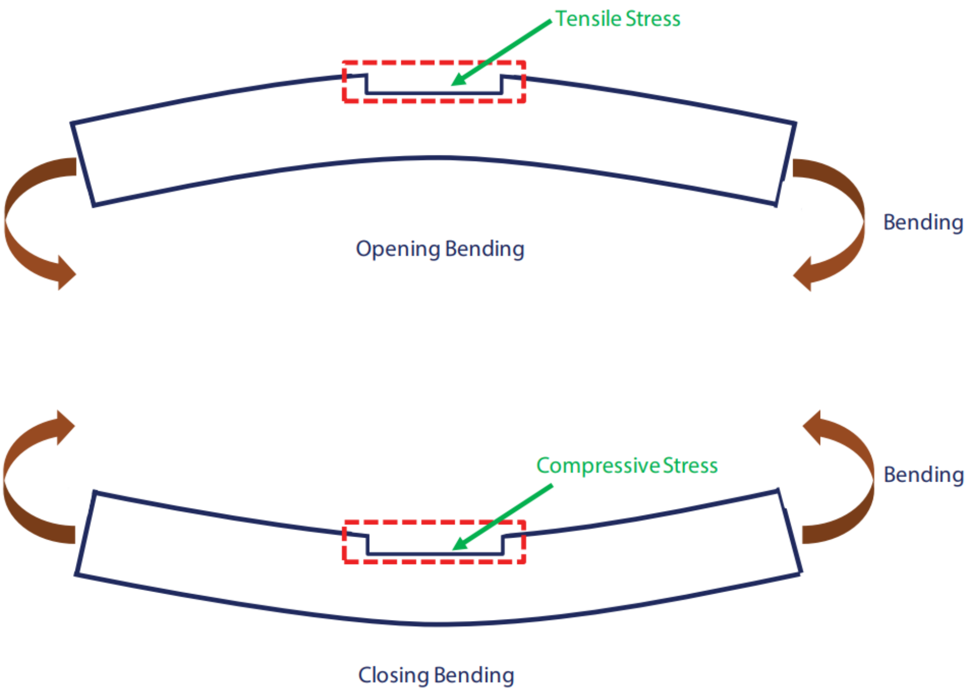

Figure 3.1 Types of bending moment on corroded pipelines

Initially, it was decided to check the boundary conditions of the finite element model for the convergence of the results with analytical data. To do this, all boundary conditions were set and the model of the pipeline section without a defect was bent with a bending radius of ρ = 500D.

The bending radius was set using the Remote Displacement function via the angle of rotation and calculated according to the equation:

$$\alpha = \frac{L \times 180}{\pi \times r} . \tag{3.11}$$

$$\alpha = \frac{2000 \times 180}{\pi \times 500 \times 1420} = 0{,}161^{o} .$$

During the testing of the model, stresses of about 198,15 MPa were obtained (Figure 3.2), which correlate with the obtained analytical data:

$$\sigma_b = \frac{ED}{2\rho} . \tag{3.12}$$

$$\sigma_b = \frac{2{,}05 \cdot 10^5 \cdot 1{,}420}{2 \cdot (500 \cdot 1{,}420)} = 205 \;\; MPa .$$

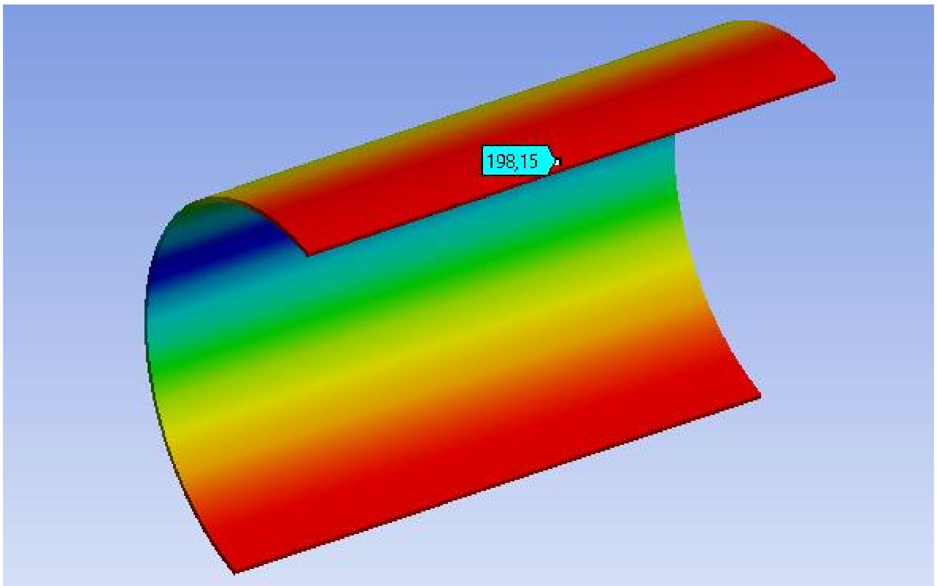

Figure 3.2 The model of the pipeline with a bend

In order to further validate the FEM, the outputs of the FEM were compared to results obtained using the thin-walled pressure vessel theory. The FEM outputs of interest are the hoop stress and longitudinal stress. Without the influence of the corrosion defect, the FEM should output results that are close to that of theoretical calculations. Input data was used from the Table 1. The theoretical values of the hoop stress and longitudinal stress can be determined according to [1] using equations:

$$\sigma_h = \frac{pD_i}{2\delta} . \tag{3.13}$$

$$\sigma_a = \mu\sigma_h - \alpha E\Delta t \pm \frac{ED}{2\rho} . \tag{3.14}$$

Table 3.5 Comparison of the theoretical and FEM results

Theoretical calculation		FEA analysis results		Absolute percentage difference	
Hoop stress, MPa	Axial stress, MPa	Hoop stress, MPa	Axial stress, MPa	Hoop stress, %	Axial stress, %
247,8	426,9	251	415	1,29	2,78

Referring to the percentage difference values column of Table 3.5, the percentage difference of values obtained from theoretical calculations and FEM are approximately 1,29% and 2,78% for hoop and axial stress respectively. This amount of difference is relatively small, thus proving that the outputs of the FEM are in line with the current theoretical knowledge.

3.2.2.3 Failure criterion concept

Three kinds of burst failure criteria have been put forward to judge the failure of pipe containing a defect in the numerical simulation.

1) Criterion of elastic limit [47]. Only the elastic stress state of the pipe calculated. The criterion recognizes that pipe is safe if the Mises stress in the residual wall thickness at a defect in pipe does not exceed the yield strength of pipe. This criterion completely limit the stress in the residual wall thickness in the elastic range. It is simple and convenient.

2) Criterion based on the plastic limit state [48] In this criterion, the failure of pipe can be defined by the circumferential stress value in the residual wall thickness at a defect. If the maximum circumferential stress value in the residual wall thickness at a defect reaches the tensile strength of materials, the pipe fails due to plastic deformation, otherwise, it is safe.

3) Criterion based on the plastic failure [49] If the minimum equivalent stress value in the residual wall thickness at a defect reaches the tensile strength of materials, i.e. the final post-yield stress of materials, a pipe collapses. This criterion appears to be reasonable because of the inherent ductility of oil and gas pipeline steels. For pit defects of smaller size, it is recommended that using 0.9 of tensile limit is more appropriate [50].

3.2.3 Proof of the need to take into account bending stresses

To implement the proof of the need to take into account bending stresses, it is necessary to analyze the dependence of the internal working pressure in the pipe on equivalent stresses in the pipe body for two cases: a corroded pipeline section with bending and a corroded pipeline section without bending.

In the Ansys program, the following pipeline model (Table 1) and defect (Table 2), presented in Figure 3.3, were selected to determine the above dependence.

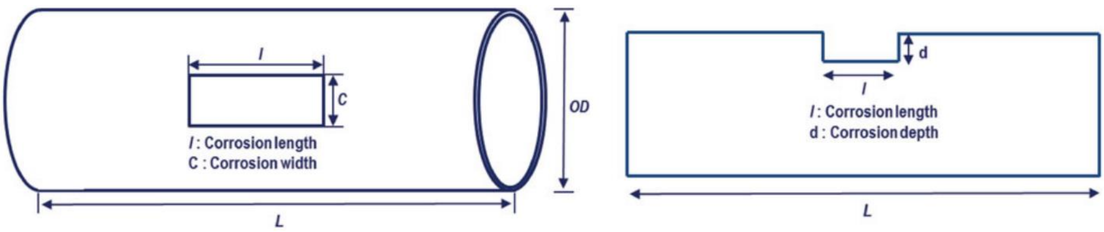

Figure 3.3 Schematic view of corroded pipe target structure

The calculation principle is to consider the range of values for the internal pressure depending on the equivalent stresses. First of all, the case of loading an intact pipeline section with a bend and without was considered (Figure 3.4).

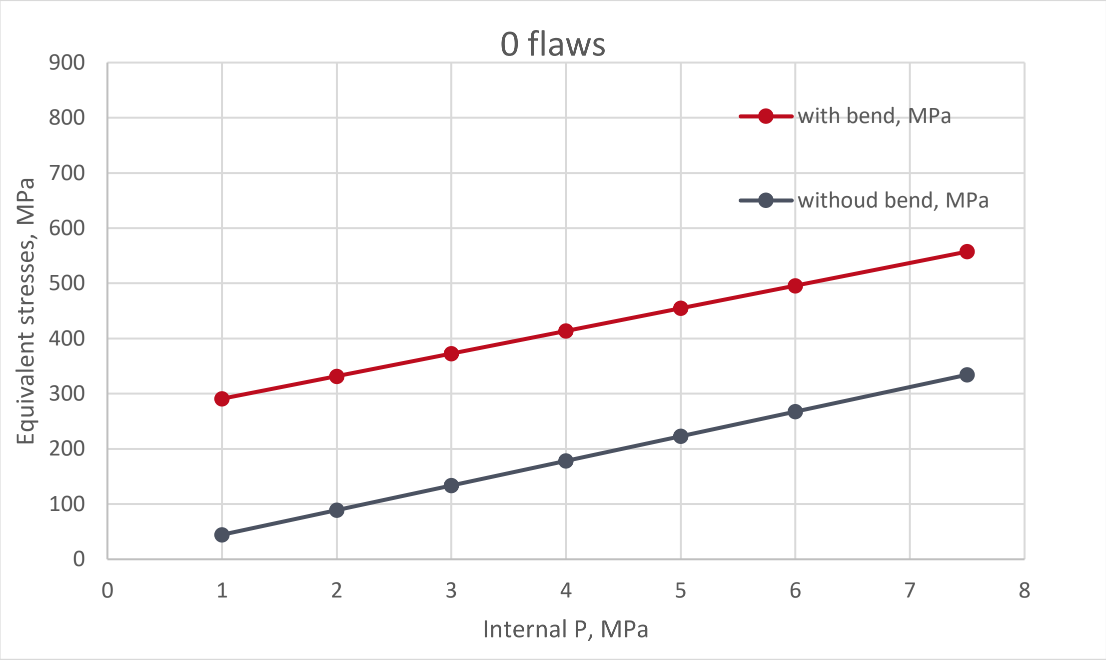

Figure 3.4 Loading of the intact pipeline section

One can notice a significant superiority of the values of the obtained stresses for the model with bending over the model without bending, which will later play an important role in determining the permissible operating pressure. The more interesting is how the stresses in the area of defects will behave.

Next, the case of loading a pipeline section with 1 defect was considered, taking into account and without taking into account bending loads (Figure 3.6).

In this case, we are once again convinced of the influence of bending stresses, which make a significant contribution to the total equivalent stresses. It is worth noting that the magnitude of bending stresses prevails at the beginning, after which the stresses increase mainly due to hoop stresses from pressure. By correlating the value of the yield strength σ_Y, the permissible operating pressure for this section is determined in the form of P_p = 4.3 MPa, which in turn correlates well with the value obtained according to the method [5,8] from the previous chapter.

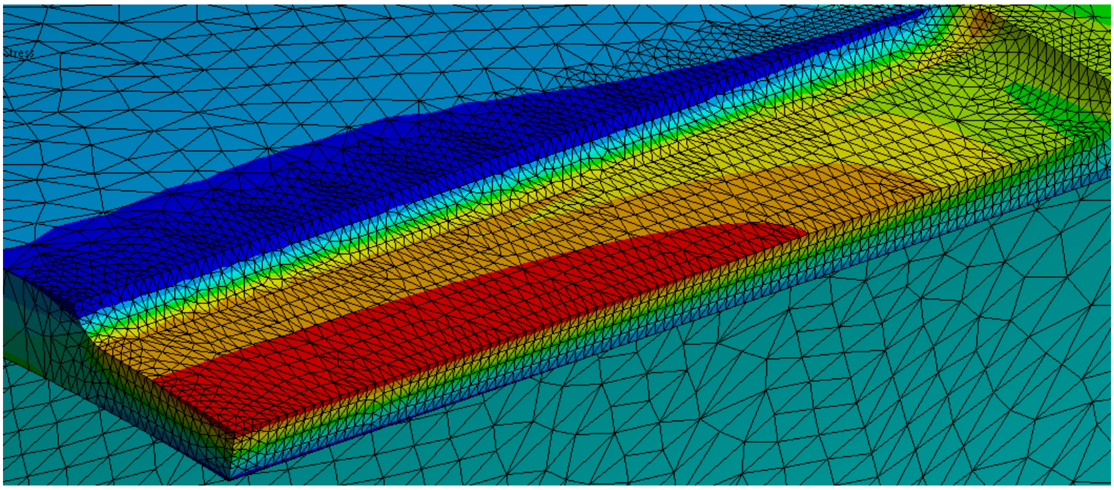

Figure 3.5 Von Mises stresses for the case with one defect

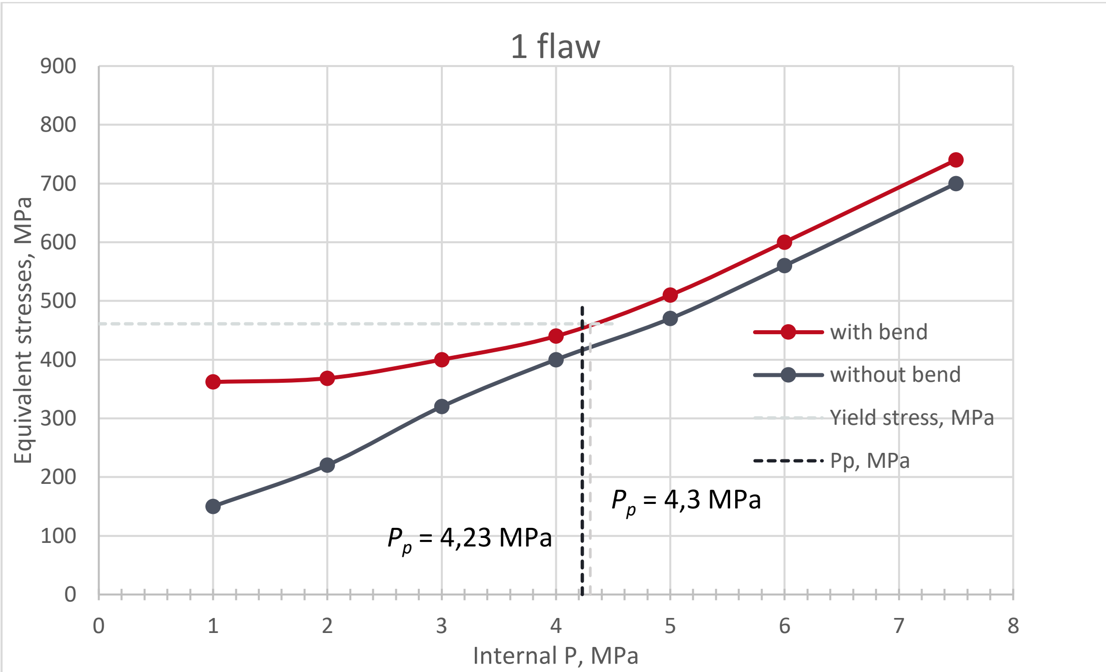

Figure 3.6 Loading of the pipeline section with 1 defect

Last of all, the case of interacting defects in the pipeline wall was considered in two cases: 1) taking into account bending loads; 2) without taking into account bending loads (Figure 3.7). The case was considered when two identical defects lie at an axial distance equal to 20 mm from each other, which corresponds to the condition of interaction of defects according to [5,8].

In this considered case, an even greater difference is noticeable between the values of the stresses obtained for the model with and without bending, which shows even greater importance of the need to take into account bending loads. When the yield strength σ_Y is correlated with the stress distribution curves, it can be expected that the permissible operating pressure will be approximately 2.6 MPa and 3.8 MPa for the model with and without bending, respectively.

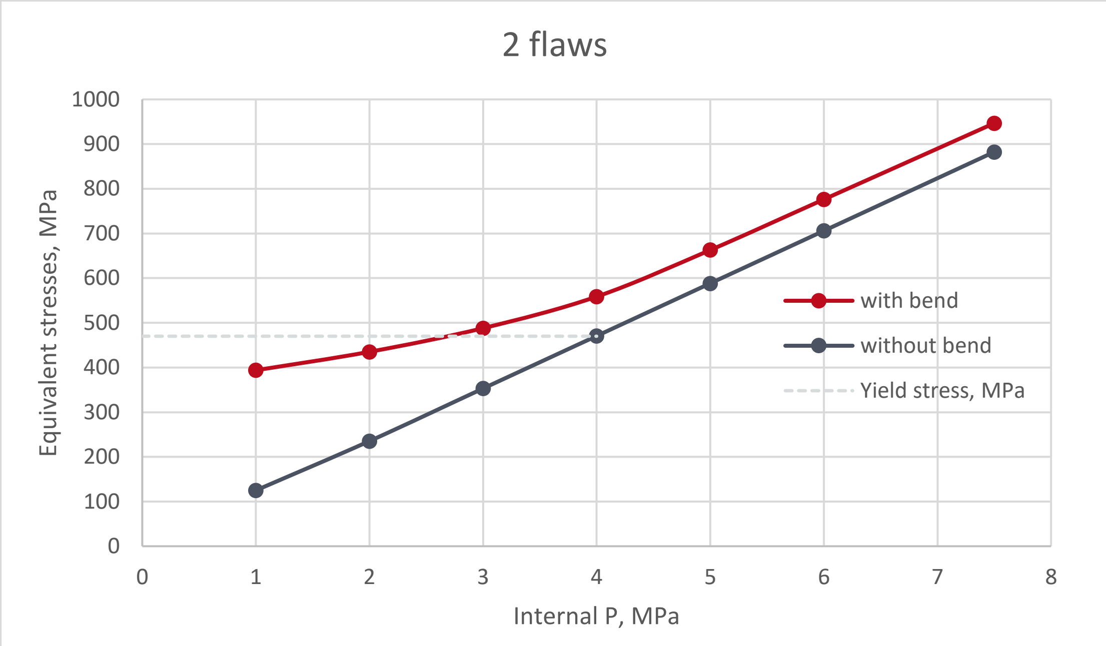

Figure 3.7 Loading of the pipeline section with 2 defects

Drawing a conclusion from all the cases considered, we can draw an unambiguous conclusion: bending stresses are significant, which means they must be taken into account in the methodology of standard technical documents. Interacting defects with bending stresses are especially dangerous – the very case that is not considered in the documents [5,8]. The revision of this part of the documentation by taking into account axial and bending loads is discussed in the next section.

3.3 Analysis of the interaction of defects in a pipeline

In this part, the case of interacting defects is considered, taking into account stresses from internal pressure, axial and bending effects. The previous pipeline model was chosen as the model (Table 3.1), with new defect parameters (Table 3.6). The analysis was performed using numerical methods of Ansys, and plastic properties of the material were also added to the model.

Table 3.6 Parameters of the defects

Name of the parameter	Value
Defect length, mm	500
Defect width, mm	100
Defect depth, mm	8

At the beginning, it was decided to analyze the effect of the distance between defects: various axial distances between defects in the zone of their interaction were considered according to the conditions described below and one value outside the interaction condition, and the results of equivalent stresses were obtained depending on a number of internal pressure values (Table 3.7).

According to [5,8], defects are considered interacting if both conditions are not met:

$$s_i > 2\sqrt{Dt}, \tag{3.15}$$

$$s_c > \pi\sqrt{Dt}, \tag{3.16}$$

where s_i – distance between two adjacent defects in the axial direction, mm,

s_c – distance between two adjacent defects in the circumferential direction, mm.

$$s_i > 2\sqrt{1420 \cdot 19{,}5} = 332{,}8 \ \ mm,$$

$$s_c > \pi\sqrt{1420 \cdot 19{,}5} = 522{,}7 \ \ mm,$$

Table 3.7 Equivalent stresses depending on internal pressure at different axial distances between defects

P, MPa	σ, MPa					
	s_i = 20 mm	s_i = 40 mm	s_i = 80 mm	s_i = 160 mm	s_i = 330 mm	s_i = 500 mm
1	401,21	397,67	394,15	388,44	385,87	381,21
2	429,4	423,09	419,26	414,36	408,73	404,4
3	443,59	438,51	431,37	426,29	421,8	411,59
4	463,86	451,98	449,51	444,23	442,46	438,86
5	484,59	478,9	471,39	467,02	465,12	462,59
6	519,25	511,54	506,15	503,17	495,39	489,25
7	555,89	547,98	539,77	530,85	521,69	515,89
8	585,18	578,04	568,23	562,31	556,02	549,18

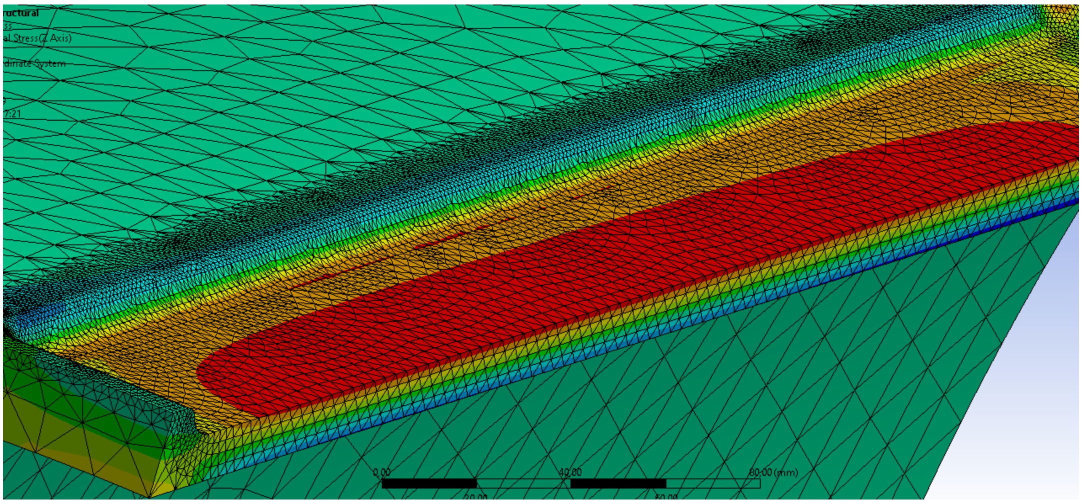

Figure 3.8 Von Mises stresses for the case with two defects

According to the results obtained, it is possible to verify the significance of the influence of the distance between the defects. Therefore, it is necessary to obtain an analytical solution for taking into account the distance between defects, which can be implemented in the existing methodology for a single defect subjected to internal pressure and axial loads [5,8].

From the Figure 3.9 one can note the function σ_{eq} *(P, s_i)*. Therefore, the computing program Table Curve 3D was employed in order to obtain the equation that accurately describe the above mentioned dependence.

The following equation was obtained, the graph of which can be observed in Figure 3.10:

$$\left(\sigma_{eq}\right)^{-1} = 0{,}002496 - 0{,}0001112 \cdot P + 3{,}4966 \cdot \ln\left(s_i\right), \tag{3.17}$$

where P – internal pressure in the pipeline, MPa,

s_i – axial distance between interacting defects, mm.

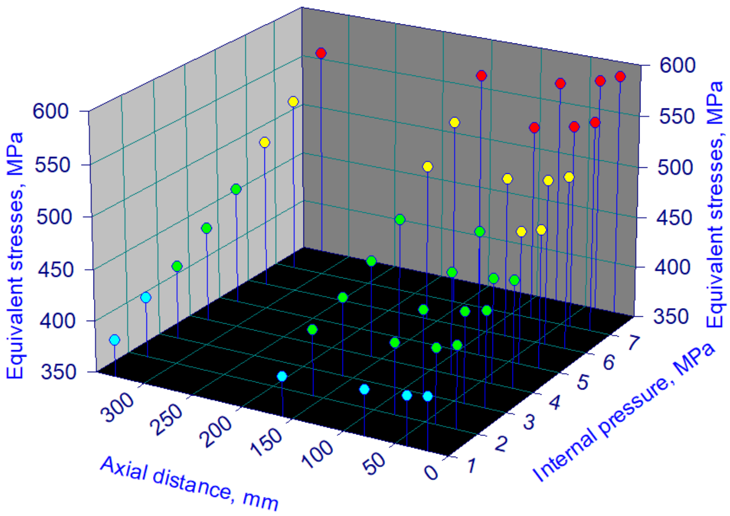

Figure 3.9 The graphical representation of the distribution of dependence σ_{eq} (P, s_i)

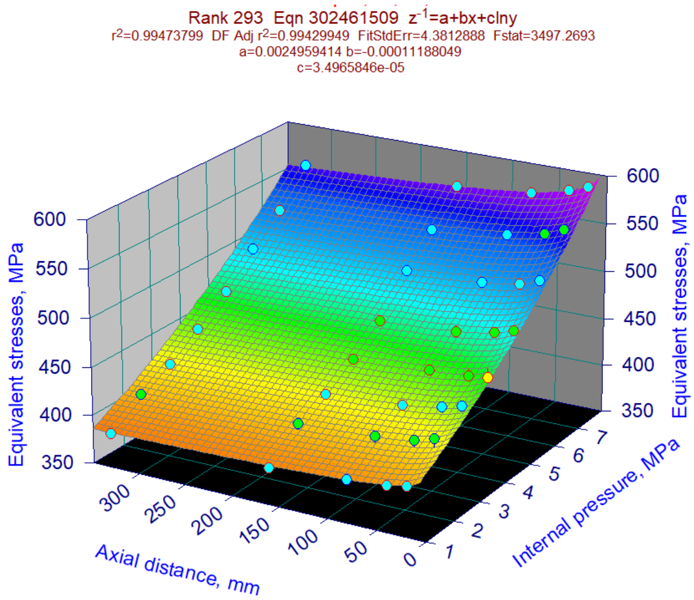

Figure 3.10 The functios σ_{eq} (P, s_i) computed in Table Curve 3D

As one can note, obtained with the help of the computer program equation for the function σ_{eq} *(P, s_i)* has a sufficiently high coefficient of determination, $r^2 = 0.995$.

Further, it is necessary to take one additional point (Figure 3.11) outside the interaction of defects according to the condition [5,8], which is taken as the point of absence of interaction between defects and based on which the value of the influence of the axial distance between defects on the residual strength of the pipeline will be obtained.

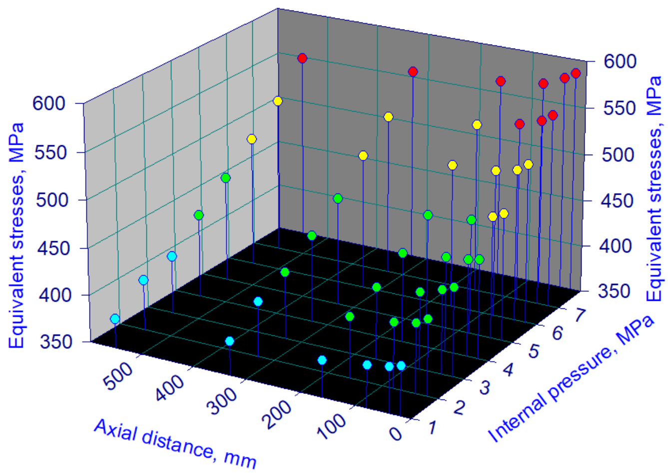

Figure 3.11 The graphical representation of the distribution of dependence σ_{eq} (P, s_i)

Outside of the interaction according to the condition (3.15), the stresses do not change. Accordingly, outside of the interaction, the correction factor is assumed to be equal to one. For the zone of interaction of defects, it is necessary to obtain a correction factor ξ_i that is calculated by dividing particular stresses (in particular axial distance between defects) by stresses outside interaction and that will be greater than one. Therefore, among the eight cases of the considered pressure, the average value of the stress ratio at the selected axial distance to the stress outside the interaction is calculated and a dependence is built depending on the distance between the defects (Figure 3.12).

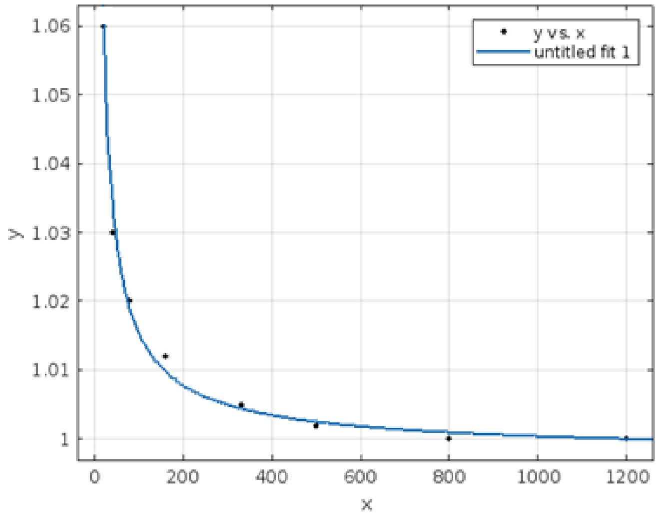

Figure 3.12 The graphical representation of the dependence ξ_i (s_i)

Further, with the help of mathematical instrument Matlab and particularly function Curve Fitter the following equation for this dependence was obtained:

$$\xi_i = 0,6284 \cdot s_i^{-0,7769} + 0,9976, \tag{3.18}$$

where s_i – axial distance between defects, mm

It should be noted that obtained with the help of the computer program equation for the function ξ_i (s_i) has a sufficiently high coefficient of determination, r^2 = 0.993. In addition, it should be said that larger and nearby defects are limited by a coefficient of 1.06. In case of general (continuous) corrosion, with a length along the axis of more than half of the pipe diameter, the permissible operating pressure must be assigned based on the residual thickness of the pipe wall, checking strength and stability in accordance with SP 36.13330.2012 “Main pipelines”.

3.4 Consideration of the defects location relative to the impact zone of longitudinal stresses

It should be said that there may be cases when defects will lie in the zone of compressive stresses from bending, thereby bending stresses will compensate for tensile stresses from pressure and axial loads in the defect zone.

In that cases methodology from [5,8] may overestimate axial stresses, therefore consideration of tensile/compressive axial loads in the defect zone should be added.

To determine the hourly location of the defect, half of the arc is sufficient (this will be either a compression or stretching zone). The maximum value of bending stresses is 12 hours, this point is taken as the starting point of the angle between the defect and the maximum value of bending stresses (Figure 3.13). Any distance of the defect(s) from 12 hours will reduce the amount of bending stresses in the defect and will be taken into account by the expression:

$$\sigma_b = \frac{ED}{2\rho} \cdot \cos\alpha, \qquad (3.19)$$

where α – angle between flaw axis and pipeline axis.

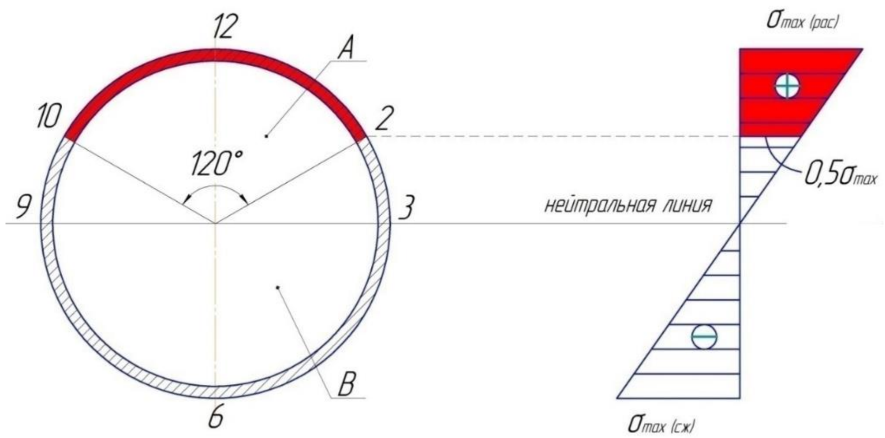

Figure 3.13 Scheme of the distribution of bending stresses along the pipeline section (in hourly coordinates)

3.5 Modified algorithm

The development of a methodology for assessing the operability of pipelines with interacting defects while taking into account loads from internal pressure and axial stresses is based on algorithms used for single defects.

The following is a sequential algorithm for evaluating the operability of a pipeline section with interacting defects when taking into account loads from internal pressure and axial loads:

- In the pipeline section the condition of interaction of defects by formulas (3.15-3.16) is checked;

- In the considered zone, taking into account the condition of interaction of defects, all defects are sequentially numbered i = 1, 2, 3, ..., N;

- Failure pressure p^i_f is calculated for each single defect without taking into account the interaction with other defects:

$$p_f^i = \frac{2t\sigma_U\left(1-\frac{d_i}{t}\right)}{\left(D-t\right)\left(1-\frac{d_i}{tQ_i}\right)},\ i=1,2,3,...,N, \tag{3.19}$$

where coefficient Q_i is calculated for each number of defect:

$$Q_i = \sqrt{1+0,31\left(\frac{l_i}{\sqrt{Dt}}\right)^2},\ i=1,2,3,...,N. \tag{3.20}$$

- Compilation of all possible sets of combining defects is carried out according to the following principles:

a) Pairs of defects are made sequentially (1,2), (2,3), (3,4) etc.;

b) Among all possible cases two the most dangerous cases must be considered: pair of defects where presented defect with the maximum value of the depth and pair of defects with smallest value of axial distance between them.

d) Compilation of sets continues until all possible interaction pairs from the total number of N defects are taken into account.

- Ratio of the width of the defect in the circumferential direction to the nominal outer perimeter of the pipe, calculated by the formula:

$$\theta = \frac{c}{\pi D}. \tag{3.23}$$

- Hoop stresses from internal pressure are determined:

$$\sigma_h = \frac{pD_i}{2\delta}. \tag{3.24}$$

- Axial stresses from internal pressure and temperature difference are calculated:

$$\sigma_{\Delta t,p} = \xi_i\left(\mu\sigma_h - \alpha E\Delta t\right). \tag{3.25}$$

- Then axial stresses from elastic bending are determined:

$$\sigma_b = \xi_i\left(\pm\frac{ED}{2\rho}\cos\alpha\right). \tag{3.26}$$

- Total axial stresses from internal pressure, temperature difference and elastic bending:

$$\sigma_a = \xi_i\left(\mu\sigma_h \pm \alpha E\Delta t \pm \frac{ED}{2\rho}\cos\alpha\right). \tag{3.27}$$

- Coefficient H_1 is determined taking into account the impact of longitudinal loads:

a) In the case of tensile total longitudinal stresses for a range of values of θ expressed by the inequality:

$$\theta \geq \frac{1}{\pi}\arcsin\left[\frac{\sigma_b}{(\sigma_Y+\sigma_U)}\frac{\pi}{\left(2-\frac{d}{t}\right)}\right], \tag{3.28}$$

$$H_1 = \left(1+\frac{\sigma_Y}{\sigma_U}\right)\left\{\left(1-\frac{d}{t}\theta\right)-\frac{2\sigma_{\Delta t,p}}{\sigma_Y+\sigma_U}-\frac{2}{\pi}\arcsin\left[\frac{\sigma_b}{\sigma_Y+\sigma_U}\frac{\pi}{2}+\frac{d}{2t}\sin(\theta\pi)\right]\right\}. \tag{3.29}$$

b) In the case of tensile total longitudinal stresses for a range of values of θ expressed by the inequality:

$$\theta < \frac{1}{\pi}\arcsin\left[\frac{\sigma_b}{(\sigma_Y+\sigma_U)}\frac{\pi}{\left(2-\frac{d}{t}\right)}\right], \tag{3.30}$$

$$H_1 = \left(1+\frac{\sigma_Y}{\sigma_U}\right)\left\{\left(1-\frac{d}{t}\theta\right)-\frac{2\sigma_{\Delta t,p}}{\sigma_Y+\sigma_U}-\frac{2}{\pi}\left(1-\frac{d}{t}\right)\arcsin\left[\frac{\left[\frac{\sigma_b}{\sigma_Y+\sigma_U}\right]\frac{\pi}{2}+\frac{d}{2t}\sin(\theta\pi)}{\left(1-\frac{d}{t}\right)}\right]\right\}. \tag{3.31}$$

c) In the case of compressive total longitudinal stresses:

$$H_1 = \frac{1+\frac{\sigma_a}{\sigma_Y}\frac{1}{A_r}}{1-\frac{1}{2A_r}\frac{\left(1-\frac{d}{t}\right)}{\left(1-\frac{d}{tQ}\right)}}, \tag{3.32}$$

where A_r – circumferential area reduction factor, calculated by the expression:

$$A_r = 1-\frac{d}{t}\theta. \tag{3.33}$$

- Failure pressure is determined for each defect:

$$p_f = \frac{2t\sigma_U\left(1-\frac{d}{t}\right)}{(D-t)\left(1-\frac{d}{tQ}\right)}H_1. \tag{3.34}$$

- Failure pressure is determined by determining the minimum value of the failure pressure from all defects:

$$p_f = \min(p_f^1, p_f^2, ..., p_f^N). \tag{3.35}$$

-At the end, the permissible pressure on the pipeline section is determined:

$$p_p = \frac{p_f}{K}. \tag{3.36}$$

According to this method, a verification calculation was carried out. Data were taken from Tables 3.1 and 3.6. The case of tensile axial stresses was considered. The results are presented in Table 3.8 and absolute percentage difference accounted for 1,6%.

Table 3.8 Comparison of the improved method and FEM results

Theoretical calculation	FEA analysis results	Absolute percentage difference
Permissible pressure, MPa	Permissible pressure, MPa	
3,64	3,7	1,62%

3.6 Conclusions. Chapter results.

1. Numerical model was created in Ansys software package and was verified by burst tests performed by [46]. In addition, numerical model was verified by analytical calculations and compared with the methodology [5,8], the improvement of which is considered in this paper.

2. Based on Ansys software package the importance of the need to take into account bending stresses has been proved.

3. Based on numerical model and Table Curve 3D program a correction factor was obtained to consider for the interaction of two nearby corrosion defects.

4. The improved methodology (which is missing in [5,8]) for the assessment operability of trunk pipelines with interacting corrosion flaws subjected to combined loading has been developed.

5. The ability of accounting flaws location relative to the impact zone of compressive/tensile longitudinal stresses has been added to the methodology.

Chapter 4

Conclusion

4.1 Summary

A review of the literature on this topic showed that the topic is relevant and that methodology presented by DNV [8] is the most comprehensive one. In addition, Russian standard of Gazprom [5] is fully based on the DNV standard. However, both documents do not consider taking into account axial and bending loads when evaluating pipeline sections with interacting defects.

As a result of the work, comprehensive analysis of technical documentation and theoretical methods was performed. Considering different flaw options numerical model for assesment of residual strength of trunk pipeline in combined stress state was created and was verified by burst tests performed by [46]. In addition numerical model was verified by analytical calculations and compared with the methodology [5,8], the improvement of which is considered in this paper.

Improved methodology for assessing the operability of trunk pipelines with interacting flaws taking into account longitudinal stresses was obtained. In addition, the ability to take into account the actual location of the defect relative to the applied loads was added to the methodology.

4.2 Evaluation

The work has been done qualitatively and corresponds to the set goals. For more accurate results full-scale burst tests should be carried out in real conditions, which was not possible because the funds and access to industrial conditions and technologies was not available for this work.

4.3 Future Work

It is necessary to conduct a full-scale burst test and then conduct a series of additional studies of finite element modeling, which will describe the nature of the impact of interacting defects better. It is also worth considering the influence of various properties of steel: strain hardening, high and low grade steels, true stress-strain curve, etc. In addition, it would be great to take into account the influence of cyclic loading.

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Abbreviations

STD	Standard Technical Documentation
FEM	Finite Element Modeling
TP	Trunk Pipeline
ASME	American Society of Mechanical Engineers
API	American Petroleum Institute
LTA	Locally Thinned Area
ILI	In-Line Inspection
DNV	Det Norske Veritas