**International Journal of Emerging Engineering Research and Technology**
**Volume 6, Issue 8, 2018, PP 31-42**
**ISSN 2349-4395 (Print) & ISSN 2349-4409 (Online)**

# Calculation of Static and Dynamic Stress-Deformed State of Pipeline in Deformable with the Environment

**Zafar Ihterovich Boltayev[1], Akhmedov Maqsud Sharipovich[1], Ro'ziyev To'lqin Razzoqovich[2], Hamroyeva Zilola Qahramonovna[1]**

[1]*Bukhara Technological-Institute of Engineering, Bukhara, 15 K. Murtazoyev Street. Republic of Uzbekistan.*

[2]*Bukhara Medical Institute names Abu Ali Ibn Sina. Republic of Uzbekistan.*

***Corresponding Author:** Akhmedov Maqsud Sharipovich, Bukhara Technological- Institute of Engineering, Bukhara, 15 K. Murtazoyev Street. Republic of Uzbekistan.*

## ABSTRACT

*The values and nature of the distribution of stresses in the defective section of the pipeline are proposed to be determined by calculating the stressively deformed state of the pipeline model with a defect in the finite element method in the ANSYS Mechanical Workbench software package.*

**Keywords:** *Pipeline, stress, Defect, Finite element method, Material.*

## INTRODUCTION

One of the most important structures of the national economy is pipeline transport. There are no industrial enterprises where pipelines of various purposes are not used, so its reliability should be given special attention. To ensure the structural reliability of the pipeline, a strength calculation is used, which reflects the operation of the pipeline. The principles of the calculation of pipelines are the choice of the design scheme for various loads and impacts, their accounting, and also the designation of limiting states, i.e. allowable level of the stress-strain state, taking into account the accepted hypotheses and assumptions in the calculation [1,2]. Static limit states - by carrying capacity (strength and stability of structures, fatigue of the material), having reached this state, the design is no longer able to resist external influences. The possibility of achieving a particular design limit state depends on many factors, of which the most important are external loads and other impacts, the properties of materials and defects, and the like. [3-5]. The calculation of the pipeline is important for predicting the operability and durability of the structure in real conditions. In many respects this is important not only from the technical side, but also from the economic side. Calculation can reduce the costs of the manufacturer. The purpose of this work is to study the stress-strain state of the pipe and support structure, as well as the deformed state of the pipeline with a defect. The expanded concept of numerical simulation of the main pipeline networks was developed and scientifically justified [5,6]. Initially, this concept was considered as a set of interrelated and mutually agreed provisions formulated in the form of key rules and detailed recommendations on the application of high-precision numerical modeling to solve production problems of pipeline transport at all stages of its life cycle. The most promising form of practical use of such modeling is the technology of high-precision computer simulation of life cycles of main pipeline networks [7, 8]. By high accuracy here is meant the maximum attainable approximation of the model to reality. Here it is necessary to emphasize that the analysis of the system's behavior during its life cycle and the development of recommendations for achieving the goals set for the system when it is created or formed during its functioning is a fundamental task of system analysis [9-12]. This formation took place at the intersection of mathematical physics, applied mathematics, management theory, mathematical programming and technical sciences on pipeline transport. Until the beginning of the XXI century, computer modeling of pipeline systems had very limited application. When constructing a continuous function model, the following steps are performed:

- A finite number of points are fixed in the region under consideration. These points are called nodal points, or simply nodes.

- The value of a continuous value at each node point is considered a variable, which must be determined.
- The domain of definition of a continuous quantity is divided into a finite number of sub domains, called elements. These elements have common nodal points and in aggregate approximate the shape of the region.
- A continuous value is approximated on each element by a polynomial, the coefficients of which are determined with the help of the values of this quantity at the node points. For each element, a polynomial is defined, but the polynomials are chosen in such a way that the continuity of the value along the boundaries of the element.

## Finite Element Method

The finite element method is one of the most widely used methods for solving problems in mathematical physics. This is due to the great versatility of the method, which combines the best qualities of variation and difference methods. Its undoubted merits include the possibility of using a variety of grids, the comparative simplicity and uniformity of methods for constructing schemes of high orders of accuracy in areas of complex shape. The constructions of modern pipeline systems are often operated in conditions of increased levels of external mechanical influences: vibrations, impacts, linear accelerations and acoustic pressure. In order to determine the mechanical modes of structural and electronic components, it is necessary to calculate the mechanical stresses and overloads of structural elements at the design stage of pipeline systems. The finite element method is widely used for structural calculations. To automate the compilation of a finite-element mathematical model, it is necessary to use finite element grid generators, which automatically divide a given area of a design into finite elements. The complexity of the shapes and dimensions of the structures make it difficult to carry out the full-scale experiment. Thanks to the development of the computer, it became possible to model complex physical phenomena. Among all numerical methods, the finite element method was most widely used. This method is the most effective and universal. FEM is by far the generally accepted method of structural analysis in a number of areas of science and technology. To describe the strained and deformed state of a deformed body, dissected into finite elements, the stiffness matrix for which is known, it is necessary to combine all the elements into a single system that approximates the calculated one, i.e. to satisfy the conditions of static and kinematic compatibility for the structure as a whole. It should be noted that the design is represented by a set of elements interacting at a finite number of node points, and therefore these conditions must be established for these points of the system. Most often these questions are solved on the basis of the energy principles of the mechanics of deformable media, mainly based on the fact that the energy of the system is equal to the sum of the energy, each of which relates to the corresponding finite element. Let the external load at the nodes of the system be represented by the column vector

$$\{\overline{P}\} = \left\{\{\overline{P}\}^{(1)}\{\overline{P}\}^{(2)}\ldots\{\overline{P}\}^{(k)}\ldots\{\overline{P}\}^{(m)}\right\}$$

Under the influence of this load, the nodes of the system receive displacements

$$\{\overline{q}\} = \left\{\{\overline{q}\}^{(1)}\{\overline{q}\}^{(2)}\ldots\{\overline{q}\}^{(k)}\ldots\{\overline{q}\}^{(m)}\right\}$$

Based on the principle of possible displacements for a system in an equilibrium position,

$$\{\delta q\}^T\{P\} = \iiint_V \{\delta\varepsilon\}^T\{\sigma\}dV \tag{1}$$

Here the integral is over the whole volume of the body.

Replacing the energy represented by the integral in (1) by the sum of the integrals taken over all m finite elements, we obtain

$$\{\delta\overline{q}\}^T\{\overline{P}\} = \sum_{i=1}^{m}\iiint_{V_i}\{\delta\varepsilon\}_i^T\{\sigma\}_i dV \tag{2}$$

We use the dependence

$$\{\delta q\}_i^T\{R\}_i = \iiint_{V_i}\{\delta\varepsilon\}^T\{\sigma\}dV \tag{3}$$

And rewrite equation (2)

$$\{\delta\overline{q}\}^T\{\overline{P}\} = \sum_{i=1}^{m}\{\delta q\}_i^T\{R\}_i \tag{4}$$

By grouping on the right-hand side of expression (4) those terms whose variations have identical directions of displacements at the same sites, we have

$$\sum_{i=1}^{m}\{\delta q\}_i^T\{R\}_i = \{\delta\overline{q}\}^T\{\overline{R}\} \tag{5}$$

Where $\{\overline{R}\} = \left\{\{\overline{R}\}^{(1)}\{\overline{R}\}^{(2)}\ldots\{\overline{R}\}^{(k)}\ldots\{\overline{R}\}^{(m)}\right\}$ – a vector of complete internal nodal forces for the entire structure, caused by the movements of the nodes of its discrete model. Moreover,

$$\{\overline{R}\}^{(k)} = \sum_{i\in k}\{R\}_i^{(k)} = \left\{\sum_{i\in k}\{R\}_{1i}^{(k)} \sum_{i\in k}\{R\}_{2i}^{(k)} \dots \sum_{i\in k}\{R\}_{ri}^{(k)}\right\} \tag{6}$$

Vector of the resultant internal nodal effort along the i-th all elements converging at the k-th node. Vector-column $\{\overline{R}\}$ represents the forces acting on the part of the nodes on the finite elements. Obviously, the nodes themselves are acted upon by the elements themselves $\{\overline{R}\}$, those. Reactions caused by movements of the nodes of the system and are the result of the action of internal forces, reduced to nodal exposure. Substituting expression (5) to equation (4), we obtain

$$\{\delta\overline{q}\}^T \{\{\overline{P}\} - \{\overline{R}\}\} = 0 \tag{7}$$

In order to go from (7) to the equilibrium equations, it is necessary either to assume that in $\{\delta\overline{q}\}$ There are no zero terms associated with moving the system as a rigid whole, or enter into consideration $E_1$ with the number of elements diagonally equal to the order $\{\overline{q}\}$. In this case, each component of the vector $\{\overline{q}\}$ a diagonal matrix term is mapped $E_1$. Where the component $\{\overline{q}\}$ is known from the kinematic conditions of the problem, the diagonal element of the matrix $E_1$ we take zero, and set all other diagonal elements equal to unity. Then, if we take into account that the possible displacements at the nodes where the kinematic conditions are given are zero, we can write

$$\{\delta\overline{q}\} = E_1\{\delta\overline{q}\} \tag{8}$$

Where $\{\delta\overline{q}\}$ - vector of possible displacements in the nodes of the system for all displacement components. Substituting expression (8) toDependence (7), we obtain

$$[E_1\{\delta\overline{q}\}]^T \{\{\overline{P}\} - \{\overline{R}\}\} = 0$$

As $E_1^T = E_1$ то $\{\delta\overline{q}\}^T = \{\delta\overline{q}\}^T \cdot E_1$. Hence, taking into account that $\{\delta\overline{q}\}$, we get

$$E_1\{\{\overline{P}\} - \{\overline{R}\}\} = 0 \tag{9}$$

Equation (9) is the matrix form of the equilibrium conditions for all forces applied to the nodes of the system.

Between the column vector of the total nodal reactive forces for the entire body $\{\overline{R}\}$ and moving nodes $\{\overline{q}\}$ there is a link

$$\{\overline{R}\} = [\overline{K}]\{\overline{q}\} \tag{10}$$

Where $\{\overline{K}\}$ – matrix of rigidity of the whole system.

Matrix $\{\overline{K}\}$ can be obtained with the help of known stiffness matrices for individual elements, if, for example, the vector-column is represented in the expanded matrix form $\{\overline{R}\}$

$$\{\overline{R}\} = \begin{Bmatrix} \{\overline{R}\}^{(1)} \\ \{\overline{R}\}^{(2)} \\ \vdots \\ \{\overline{R}\}^{(j)} \\ \vdots \\ \{\overline{R}\}^{(m)} \end{Bmatrix} = \begin{Bmatrix} \sum_{i\in 1}\left([K]_{i1}^{(1)}\{\overline{q}\}^{(1)} + \dots + [K]_{i1}^{(k)}\{\overline{q}\}^{(k)} + \dots + [K]_{i1}^{(m)}\{\overline{q}\}^{(m)}\right) \\ \sum_{i\in 2}\left([K]_{i2}^{(1)}\{\overline{q}\}^{(1)} + \dots + [K]_{i2}^{(k)}\{\overline{q}\}^{(k)} + \dots + [K]_{i2}^{(m)}\{\overline{q}\}^{(m)}\right) \\ \vdots \\ \sum_{i\in j}\left([K]_{ij}^{(1)}\{\overline{q}\}^{(1)} + \dots + [K]_{ij}^{(k)}\{\overline{q}\}^{(k)} + \dots + [K]_{ij}^{(m)}\{\overline{q}\}^{(m)}\right) \\ \vdots \\ \sum_{i\in m}\left([K]_{im}^{(1)}\{\overline{q}\}^{(1)} + \dots + [K]_{im}^{(k)}\{\overline{q}\}^{(k)} + \dots + [K]_{im}^{(m)}\{\overline{q}\}^{(m)}\right) \end{Bmatrix} =$$

$$= \begin{Bmatrix} \sum_{i\in 1}[K]_{i1}^{(1)} \dots \sum_{i\in 1}[K]_{i1}^{(k)} \dots \sum_{i\in 1}[K]_{i1}^{(m)} \\ \sum_{i\in 2}[K]_{i2}^{(1)} \dots \sum_{i\in 2}[K]_{i2}^{(k)} \dots \sum_{i\in 2}[K]_{i2}^{(m)} \\ \vdots \\ \sum_{i\in j}[K]_{ij}^{(1)} \dots \sum_{i\in j}[K]_{ij}^{(k)} \dots \sum_{i\in j}[K]_{ij}^{(m)} \\ \vdots \\ \sum_{i\in m}[K]_{im}^{(1)} \dots \sum_{i\in m}[K]_{im}^{(k)} \dots \sum_{i\in m}[K]_{im}^{(m)} \end{Bmatrix} \begin{Bmatrix} \{\overline{q}\}^{(1)} \\ \{\overline{q}\}^{(2)} \\ \vdots \\ \{\overline{q}\}^{(j)} \\ \vdots \\ \{\overline{q}\}^{(m)} \end{Bmatrix}. \tag{11}$$

Here the index $i \in j$ means summation over all i-th elements converging at the node j. Sub matrix $[K]_{ij}^{k}$ is a block

$$[K]_{ij}^{(k)} = \iiint_{V_i} \left([B]^{(j)}\right)^T [D][B]^{(k)} dV = \left(\iiint_{V_i} \left([B]^{(k)}\right)^T [D][B]^{(j)}\right)^T dV$$

Matrix stiffness for the i-th element, which determines the reactions at the j-th node from unit displacements at the k-th node. Moreover, if the i-th element does not contain either the node j or the node k, then the sub matrix $[K]_{ij}^{k}$ should be set equal to zero. Thus, the rigidity matrix for the entire system will have the form

$$[\bar{K}] = \begin{bmatrix} [\bar{K}]_1^{(1)} \cdots [\bar{K}]_1^{(k)} \cdots [\bar{K}]_1^{(m)} \\ [\bar{K}]_2^{(1)} \cdots [\bar{K}]_2^{(k)} \cdots [\bar{K}]_2^{(m)} \\ \vdots \\ [\bar{K}]_j^{(1)} \cdots [\bar{K}]_j^{(k)} \cdots [\bar{K}]_j^{(m)} \\ \vdots \\ [\bar{K}]_m^{(1)} \cdots [\bar{K}]_m^{(k)} \cdots [\bar{K}]_m^{(m)} \end{bmatrix} \quad (12)$$

Where $[\bar{K}]_j^{(k)} = \sum_{i \in j} [K]_{ij}^{(k)}$

Substituting now (10) into equation (9), we obtain the solving matrix EQ equation in the form of the displacement method [12-16].

$$E_1\{[K]\{\bar{q}\} - \{P\}\} = 0 \quad (13)$$

Note that since the matrix $E_1$, containing zero rows is special; it turns the matrix of a system of linear non-homogeneous algebraic equations into a special one. In view of this, expression (13) should be considered only as a system of equations for the unknown displacement components without zero rows. However, if the given kinematic conditions of the system are represented in the form

$$[E - E_1]\{\bar{q}\} = E_2\{\bar{q}\} \quad (14)$$

Where $E = E_1 + E_2 -$ the identity matrix, then equation (13) can be written as follows

$$E_1[\bar{K}][E_1 + E_2]\{\bar{q}\} - E_1\{\bar{P}\} = 0$$

From this we obtain the expression

$$[E_2 + E_1][\bar{K}][E_1]\{\bar{q}\} = E_1\{\bar{P}\} - E_1[\bar{K}]E_2\{\bar{q}\} + E_2\{\bar{q}\}$$

in which there are no null lines. Moreover, the right-hand side of equation (15) includes only known components of the displacements of the system $E_2\{\bar{q}\}$, in the left - all the sought-for components of the vector $\{\bar{q}\}$. Therefore, equation (15) should be considered as a system of equations for the sought components of a column vector of nodal displacements.

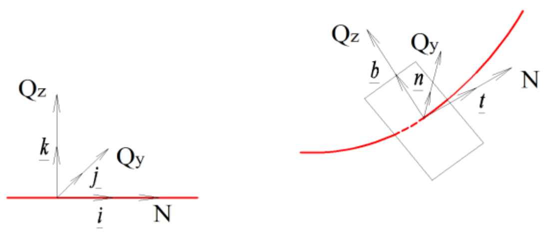
However, to perform the calculation, a special geometrically nonlinear rod element of the pressure pipeline is required.

The main reasons that make it necessary to take into account the geometric nonlinearity of the pipeline are: longitudinal-transverse bending, the development of landslide processes, the development of uneven sediments of the structure.

It should be noted that at present there are no scientific and technical works in which the smallness of the angles of rotation or deformation is justified. The assignment of small quantities, according to the literature review, is carried out in a directive way.

In order to take into account the variety of conditions for the construction and operation of underground pipelines in the work presented, the final element of the pipeline is constructed without taking into account the restrictions on the magnitude of movements, turns and deformations.

In view of the considerable length of the pipelines (tens, hundreds of kilometers), the rod element of the pressure pipeline should be classified as a thin rod and constructed in accordance with the Bernoulli-Euler hypothesis. This fact avoids the effect of degeneracy, jamming of shear deformations [17-19].




a) б)

**Fig. 1.** *Transition to the rotated vectors: a) components of the "rotated" vector N in the basis i, j, k b) components of the vector N in the basis b ,n ,t*

In the present work, the rod material is linearly elastic.

Let's note the distinctive features of the constructed final element:

- The absence of restrictions on the magnitude of displacements, rotations and deformations;
- Write the functional with the use of "rotated" vectors. Rotated vectors are vectors that differ from the original ones in that they are expressed in another basis;
- The construction of dependencies for rods in accordance with the Bernoulli-Euler hypothesis;
- The additional term in the expressions for the components of the rotation vector:

$$\varepsilon_{кв} = \begin{pmatrix} e_{кв} \\ 0 \\ 0 \end{pmatrix} \qquad \begin{cases} e = u' + \dfrac{w'^2}{2} + \dfrac{v'^2}{2} \\ \varphi_2 = \dfrac{\varphi_1 v'}{2} - w' + u'w' \\ \varphi_3 = \dfrac{\varphi_1 w'}{2} + v' - u'v' \end{cases} \qquad (15)$$

Where $u,v,w$- displacement of the rod in a spatial Cartesian coordinate system, $u',v',w'$ – displacement derivatives, $\varphi_1,\varphi_2,\varphi_3$ – components of the rotation vector. A feature of the component expressions $\varphi_2$and $\varphi_3$ (15) the rotation vector is the presence of terms $u'w'$ and $-u'v'$, respectively. These members are absent in the work of A.V. Perelmuter, V.I. Slinger, in which the construction of the rod is carried out according to the theory of the second order, starting from the equations of the theory of elasticity. It should be noted that $u',v',w'$ in the nonlinear case are not the angles of rotation with respect to any axis. All three quantities are equivalent. If we remove the quadratic terms from expressions (15), we obtain relations known from the linear theory of rods. The pipeline is considered in the program as a spatially repeatedly statically indeterminate system. The program has a standard structure of the finite element program.

## Formulation of the Problem

With temperature oscillations along the tube axis, deformations occur, which in turn create considerable stresses. In the conditions of the northern climate, the pipes can be exposed to low temperatures. Therefore, we will consider how the temperature decrease will affect the stress-strain state of the pipe.

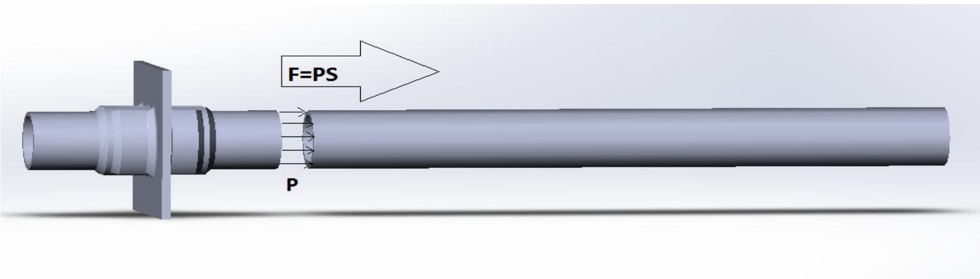



**Fig.2.** *Influence on the construction of the straight part of the pipeline.*

When analyzing the stress-strain state, we assume that the structure is rigidly fixed to the edges of the plate, and one of the ends is displaced along the axis of the tube (Fig. 2). It was assumed that this displacement occurs as a result of the action of a straight section of the pipeline when it is elongated or compressed. To calculate the magnitude of the linear expansion **ΔL** usually use the formula: $\Delta L = \alpha \cdot \Delta T \cdot L$, where $\alpha$ -coefficient of linear expansion, $1/^0 C$, L- linear size pipe, $\Delta T$ -change of temperature. When the temperature changes, the reaction force of the supports N acts on the fixed ends of the pipe, which prevents its elongation. Compressor stresses arise in the pipe wall, the magnitude of which is determined by equation: $\sigma = \alpha \cdot \Delta T \cdot E$ where $E$ – modulus of elasticity of the material. Another feature of the polyethylene pipeline is its propensity to relax: for a fixed deformation, the internal stresses are halved within an hour.

## Building a Model

In this problem, we consider a part of the pipeline design. The pipe is in a fixed support. All components are connected by welding and consist of the same material; material - PE 100. This design will be recreated in the software package Solid Works. Solid Works - software complex of the computer-aided design system for automation of works of an industrial enterprise at the stages of design and technological preparation of production. Provides the development of products of any complexity and purpose.

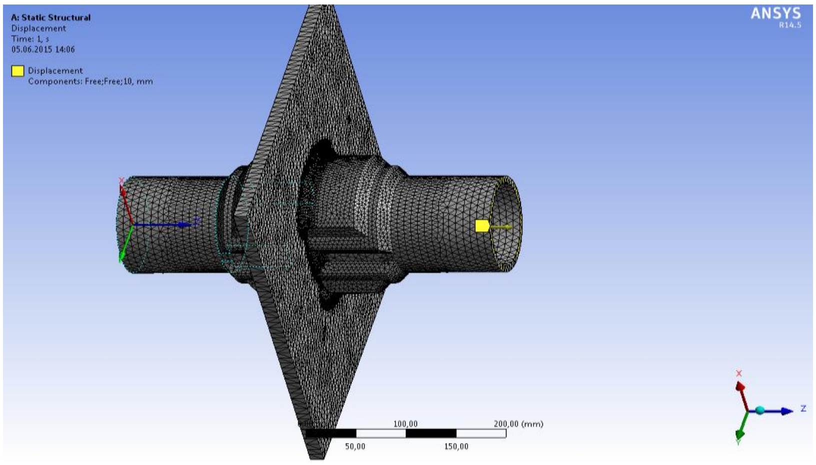



**Fig.3** *Finite element model. Projection 1*

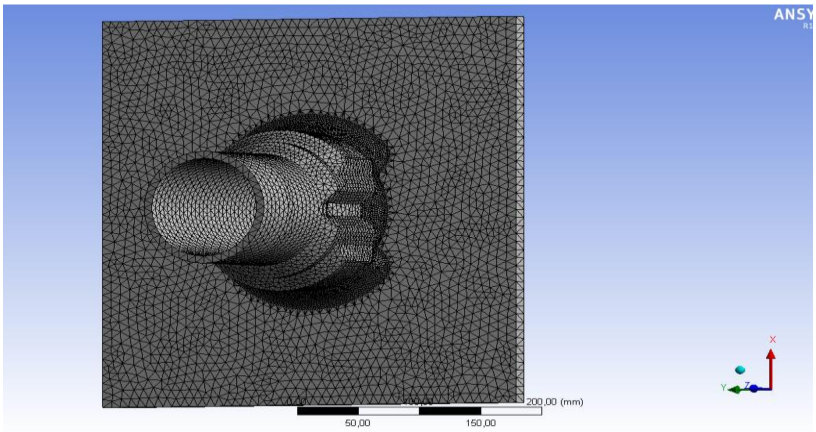



**Fig.4** *Finite element fashion. Projection 2.*

The Solid Works program uses a three-dimensional approach to design. When designing a part from the initial sketch to the final model, a three-dimensional object is created. From this three-dimensional object, you can create 2D drawings or match different components to create 3D assemblies. You can also create 2D drawings of 3D assemblies.

## CHOICE OF A GRID FOR SPLITTING THE PIPELINE STRUCTURE

After creating the structure, you need to select the appropriate partition to get the most accurate calculation results. This pipeline was divided into a triangular grid. A triangular decomposition is the most universal partition among all finite-element partitions. It allows you to superimpose a grid on models that have complex geometry (Fig. 3). The model is constructed, with a preliminary thickening of the grid in areas where high accuracy is required for calculating the stress-strain state. In the model there are 104710 elements of the 180174 node (Fig. 4).

## CALCULATION OF THE MODEL IN THE ANSYS PROGRAM

When calculating in the ANSYS program, polyethylene properties are set. These properties strongly depend on temperature, but we are interested in the ultimate strength. It has been experimentally established that the yield strength decreases with increasing temperature [7]. As a limit state of the design, the Huber-Misses criterion is used in the calculations, which determines the transition from the stage of elastic deformation of the material to the stage of plastic deformation

$$\sigma_{II} = \sigma_T,\ \sigma_{II} = \left(\frac{1}{\sqrt{2}}\right)\sqrt{(\sigma_1-\sigma_2)^2+(\sigma_2-\sigma_3)^2+(\sigma_3-\sigma_1)^2} \qquad (16)$$

Where $\sigma_{И}$ – stress intensity; $\sigma_T$ – yield strength of the material. Criterion (16) was used in studies to assess the state of the polymer pipe. Its execution for any point in the design indicates the exhaustion of the bearing capacity and the exit from the permissible operation mode. The figures show the results of calculating the stress intensity in the structure at a temperature $20^{\circ} C$, under the action of tensile force at the end of the pipe.

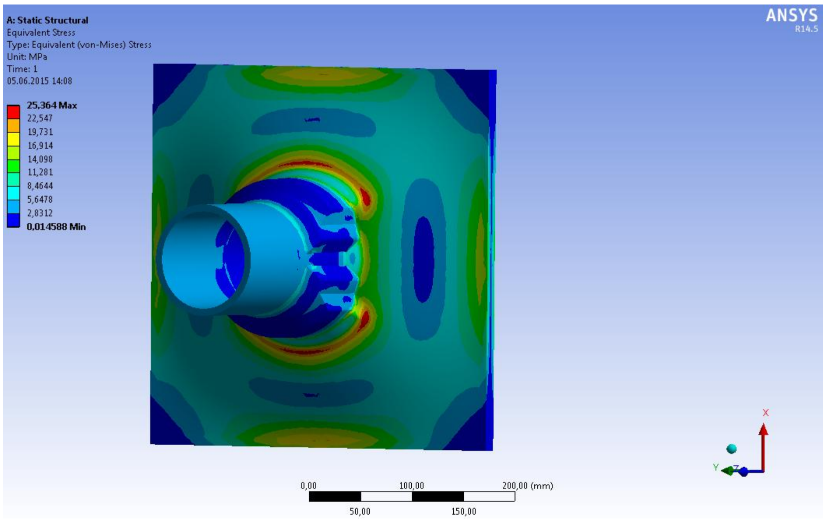



**Fig.5.** *Design stress intensity. Projection 1*

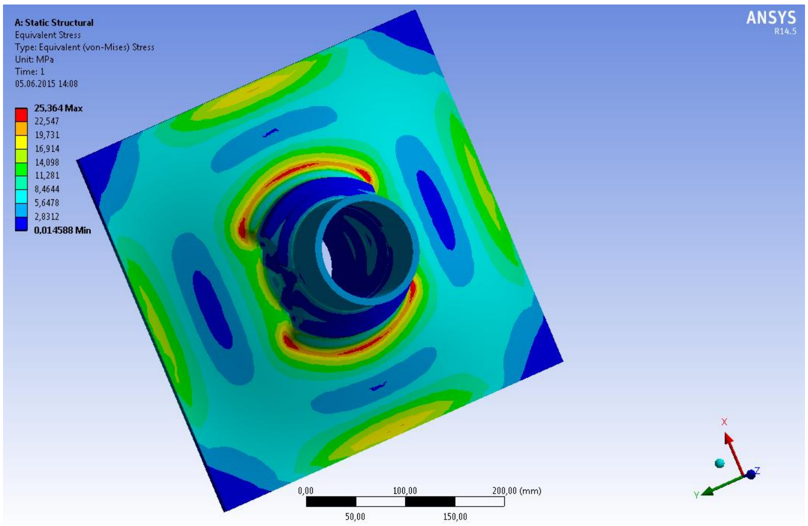



**Fig.6.** *Design stress intensity. Projection 2*

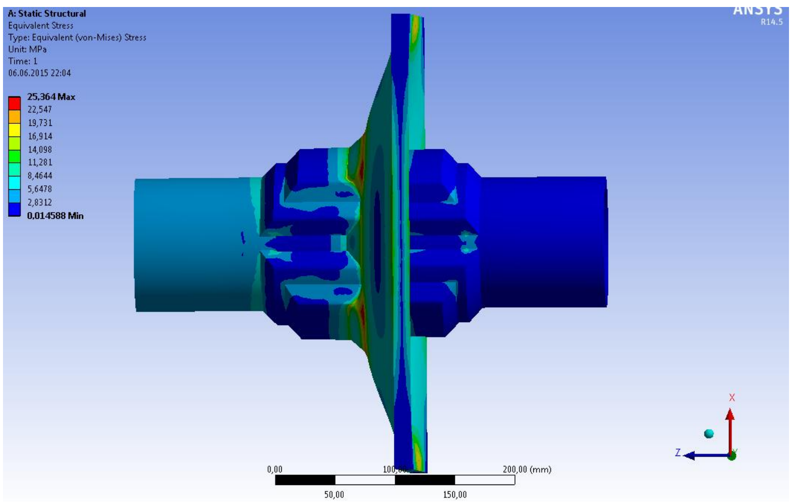



**Fig.7.** *Design stress intensity. Projection 3*

After carrying out the calculations, it can be seen that the most strained areas according to the Huber-Misses criterion are the places of welded seams and structural defects. At the end of the pipe, where the movement is applied, stresses equal to 4 MPa arise, and the seams work at the ultimate strength $\approx 25$ MPa.

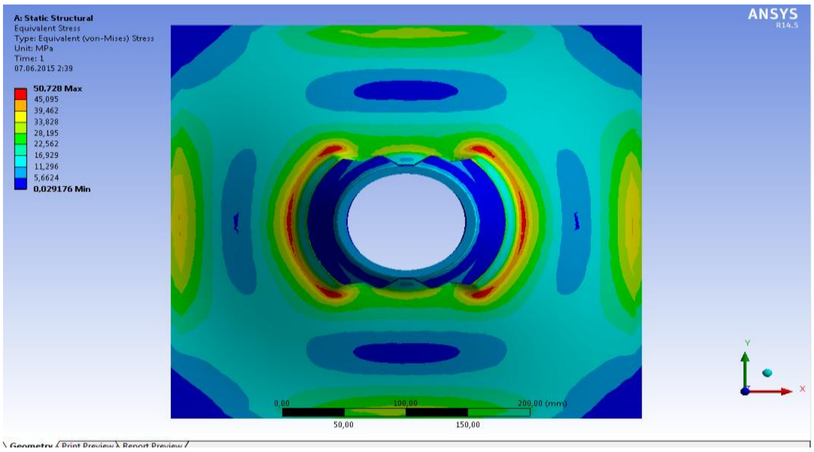



**Fig.8.** *Intensity of stresses. Projection 1*

From the results obtained, one can find the length of the straight part of the pipeline at which the structure will work. The pipe extension is set to 10 mm. According to the formula $L = \frac{\Delta L}{\alpha \cdot \Delta T}$, we find $L \approx 2.77 м$, где $\alpha = 2.4 \cdot 10^{-4}, \Delta T = 15^0 C$

We now show the picture of the stress-strain state of the model, by specifying the temperature change in 60℃ lowering the temperature from 5℃ before -55°C) .The calculation was obtained taking into account the peculiarities of the material properties change with decreasing temperature: the yield strength of the material increases, becoming approximately equal to 50 MPa, and the modulus of elasticity also behaves - it increased more than 3 times at the temperature -55℃ , than when the temperature was equal 20℃ .Limit stresses in the construction are achieved when moving the end of the pipe by 6 mm. Thus, we get, from the formula given above, that the length of a straight section will be no more than 2 meters.

## THE DEFORMED STATE OF A CURVED PIPELINE WITH A DEFECT

When carrying out flaw detection of process pipelines at oil transportation facilities, there is often a problem of assessing their technical condition and the possibility of further operation. According to preliminary estimates, there is no combination of external operational loads capable of triggering the ultimate state in the pipeline [1,2]. The purpose of this calculation is to determine the possibility of further operation of a pipeline with a defect of this type by establishing the values of the internal forces acting in the defective area and comparing the obtained values of the maximum stresses with the design resistances of the pipeline material (Fig. 9).

**Fig. 9.** *Pipe joint with defect of "dent" type.*

The state in which internal changes in the metal lead to its destruction is called the ultimate stress state. Conclusion on the reliability of the structure should be made on the basis of a comparison of the maximum stresses that can occur at the most dangerous point, with the maximum permissible values for a given material. The ultimate stress state of a structure is the boundary beyond which its operation is unacceptable. The reliability of the work is the higher, the further from the limit state the level of actual stresses inside the material of the part. The use of the ANSYS software for calculating the stress-strain state of the pipeline wall allows obtaining output results in the form of three values of the principal stresses $\sigma_0$, which are the roots of the cubic equation determined by the stress vector components [3, 4]

$$\begin{pmatrix} \sigma_x-\sigma_0 & \frac{1}{2}\sigma_{xy} & \frac{1}{2}\sigma_{xz} \\ \frac{1}{2}\sigma_{xy} & \sigma_y-\sigma_0 & \frac{1}{2}\sigma_{yz} \\ \frac{1}{2}\sigma_{xz} & \frac{1}{2}\sigma_{yz} & \sigma_z-\sigma_0 \end{pmatrix}=0 \qquad (1')$$

The principal stresses are denoted by $\sigma_1$, $\sigma_2$, $\sigma_3$. The main stresses are ordered in such a way that $\sigma_1$ is the greatest positive stress, and $\sigma_3$– the greatest negative. Intensity of voltage $\sigma_I$ is the absolute value of the greatest of the three differences: $\sigma_1$ - $\sigma_2$, $\sigma_2$ - $\sigma_3$ or $\sigma_3$ - $\sigma_1$, i.e.

$$\sigma_1 = MAX\left(|\sigma_1-\sigma_2|,|\sigma_2-\sigma_3|,|\sigma_3-\sigma_1|\right) \qquad (2')$$

Misses stresses, or equivalent stresses $\sigma_e$ (output SEQV) are calculated by the formula:

$$\sigma_e = \left(S\left[(\sigma_1-\sigma_2)^2+(\sigma_2-\sigma_3)^2+(\sigma_3-\sigma_1)^2\right]\right)^{\frac{1}{2}} \qquad (3')$$

The calculation method is based on the procedure for permissible conditional elastic stresses. The stress-strain state of a defect of the "dent" type on the cylindrical shell of a pipe is determined by the spatial work of the design model under the action of combinations of operational loads. Since the purpose of the work is to examine the stresses that arise in the region of the pipeline defect, the design model can be limited to the pipeline section containing the defect. The working load on the pipeline is the internal overpressure. Also, when calculating, you need the own weight of the branch pipe, wind and snow ice loads. Thus, in order to ensure the static definability of the model, the boundary conditions take into account the hinge fixation of the ends of the section from linear displacements (the design scheme is Figure 2).

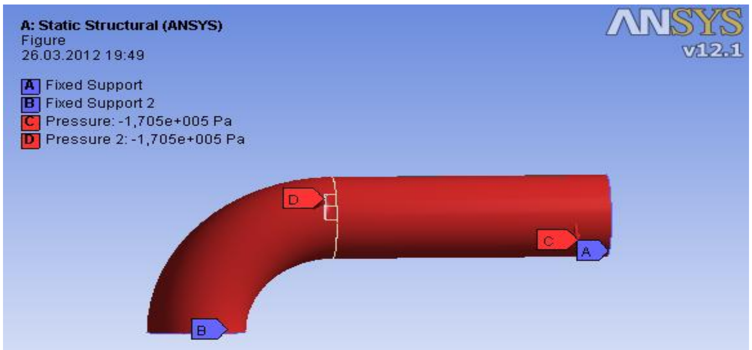



**Fig. 10.** *The design scheme of a receiving and dispensing branch pipe with a defect of the "dent" type*

The length of the shell should be chosen in such a way as to exclude the possibility of the influence of the endings on the defect area. Defect type "dent inward" is modeled in accordance with the data of measurements during the diagnosis. The loads distributed on the shell area are applied to the shell section. The weldedseamsdecided to neglect (Fig.10).

**Table 1.***Initial data for calculating the VAT of a pipeline with a dent*

| Parametername | Parametervalue |
|---|---|
| Product | Oil, water (duringhydrotesting) |
| Normativedocument for manufacturing | ASTM A694 F52 |
| Outsidediameter | 900 mm. |
| Wallthickness | 9,2 mm. – pipe / 11,4 mm. - withdrawal |
| Wallmaterials | API 5L Grx52 ASTM A694 F52 |
| Estimatedinternalpressure | 155kPa |
| Specificweight of steel | $\gamma_{cm}$ = 7850 kg/m$^3$ |
| Modulus of elasticity of steel | $E = 2\cdot10^5$MPa |
| Poisson'sratio for steels | $\nu = 0,3$ |
| Defect parameters (linear dimensions, depth) | S=110x275, h=7 mm. |

When analyzing membrane stresses in the ANSYS PC, finite elements from the ANSYS library of SHELL181 type are used to model the shell. To do this, the imported 3D model from the Autodesk AutoCAD software package is processed and optimized in the Design Modeler preprocessor in order to obtain surface elements. The finite element model consists of five shell objects, linked together by a bonded contact of the "bonded" type, i.e. It is ensured by the complete dependence of displacements in all nodes. The coordinate system is Cartesian. The imposition of a finite element mesh is assumed to be free, repeating the curvature of the surface. The grid is superimposed automatically. The minimum size of the grid face of 0.6 mm was obtained when thickening in the vicinity of the defect. In other constructive elements, the grid size is assumed equal to 10 mm. The nonlinear model is solved by iteration. Values of equivalent stresses are presented in the most loaded area of the section of the shell - the outer surface. The initial data for the calculation are given in Table. 1.

The calculation results are presented in the form of graphic images of the stress distribution fields in the pipeline wall in Figures 3-5.

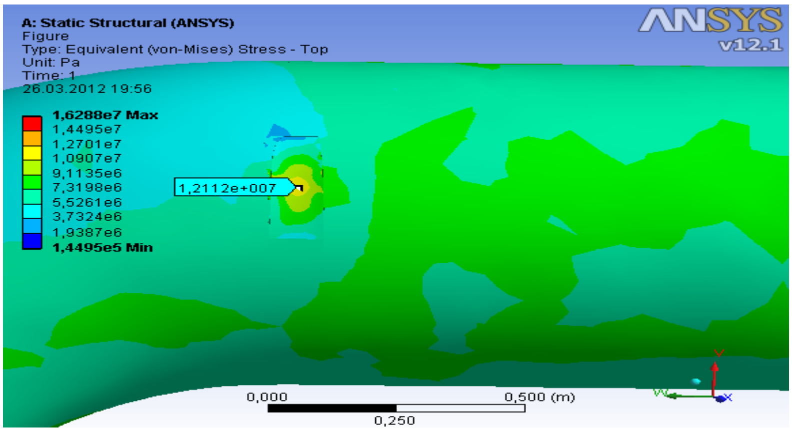



**Fig11.** *Distribution of equivalent stresses in the defective branch pipe*

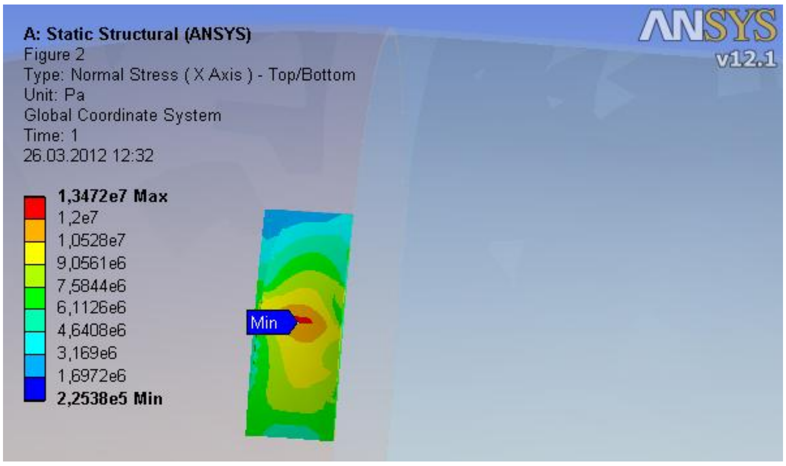



**Fig12.** *Distribution of equivalent stresses in the zone of defect of the branch pipe*

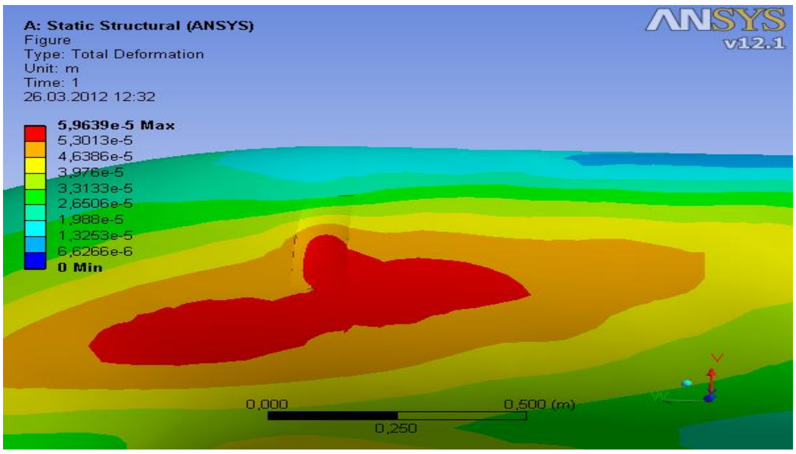



**Fig13.** *Deflections in the defective branch pipe*

The obtained values of the stresses are summarized in Table 2, where a comparative analysis of the ratio of the resulting stresses to the design resistance of the steel of the pipeline (189.58 MPa) is also given (Fig.11,12 and 13).

**Table2.** *Comparison of operating stresses with design resistance of pipeline steel*

| Name | Value | The relationship with the calculated resistance of steel |
|---|---|---|
| $\sigma_m$- The maximum value of membrane stresses in the defect area (normalstress) | 13,5 MPa | 7,12% |
| $\sigma_m$- the maximum value of membrane stresses outside the defect area | 7,54 MPa | 3,98% |
| $\sigma_e$- the maximum value of equivalent stresses along the background of the Mises in the region of the defect | 12,1 MPa | 6,38% |
| $\sigma_e$- the maximum value of the equivalent von Mises stresses outside the defect area | 9,64 MPa | 5,08% |

## CONCLUSION

This work is devoted to the solution of the problem of the suitability for the operation of a supporting structure of a pipeline made of polyethylene under the influence of temperature loads. The problem of deformation of a structure under the influence of temperature loads in the range from $5^0$Cbefore -$55^0$C.

Despite the fact that the pipeline is supposed to be used with the use of thermal insulation, which allows to reduce the negative impact of the environment, an emergency case of operation was considered in the work. In this case, the pipe will undergo a low temperature in a cold climate. The cooling of the pipe takes place in a short time, so the values of the elastic moduli were selected both for short-term loads, i.e. The property of relaxation does not have time to manifest itself to the proper degree. The following conclusions are drawn:

- A finite-element model of the nozzle PRP-900 with a metal wall defect of the "dent" type was developed in the ANSYS Workbench software package;
- The equivalent stresses arising in the defective pipeline from the operational loads are 3.98% of the calculated resistance of the steel. The dent in the pipeline causes a local stress disturbance. Thus, the growth of equivalent stresses in the region of the maximum depth of the dent amounted to 1.3% in comparison with the vicinity of the defect;
- The maximum values of membrane, tangential, and equivalent stresses do not exceed the design resistance of 189.58 MPa;
- The condition of static strength is satisfied, that is, 13.5 MPa <189.58 MPa;
- At a design pressure of 155 kPa within the pipeline with a wall thickness corresponding to the actual value, the strength of the pipeline is ensured and it can be said that there is no significant effect of the detected defect of the "dent" type.

## REFERENCES


[1] Agapov, V.P. Finite Element Method in Static, Dynamics and Stability of Constructions: Textbook. allowance / В.П. Agapov. - M.: Publishing House of the DIA, 2004. - 248 p.

[2] Alexandrov, A.V. Fundamentals of the theory of elasticity and plasticity: Textbook. for universities / A.V. Aleksandrov, V.D. Potapov. - M.:Высш. шк., 2007. - 400 p.

[3] Darkov, A.V. Building Mechanics: A Textbook for Universities / A.V. Darkov, N.N. Shaposhnikov. - St. Petersburg. : Lan, 2008. - 656 p.

[4] Ilyin, V.P. Numerical methods for solving the problems of building mechanics: Textbook. allowance / В.П. Ilyin, V.V. Karpov, A.M. Maslennikov. - M.: Publishing house of the ASV, 2005. - 426 p.

[5] Konstantinov, I.A. Building Mechanics: Textbook. for universities / IA. Konstantinov, V.V. Lalin, I.I. Lalina. - M.: Prospekt, KNORUS, 2010. - 432 p.

[6] Semenov, A.A. Design and computing complex SCAD in the educational process. Part I. Static calculation: Textbook. allowance / A.A. Semenov, A.I. Gabitov. - M.: Publishing House of DIA, 2005. - 152 p. 7. Construction fur

[7] LM Kachanov. Fundamentals of the theory of plasticity. - M: Science, 1969. 420 p.

[8] Mazalov VN, Nemirovsky Yu, V. Dynamics of thin-walled plastic constructions // Mechanics, - M: Mir, 1975, no. 5, - p. 155-247.

[9] Dikovich I.L. Dynamic bending of an elastoplastic beam-strip with axial fixation // Calculation of spatial structures - Issue 17. - Moscow: Stroyizdat, 1977. - P.79-86.

[10] Korenev BG, Rabinovich I.M. Handbook of the dynamics of buildings. MoscowStroyizdat 1972.p.512.

[11] Safarov I.I., Axmedov M.Sh. Free Oscillations of a Viscoelastic Toroidal Thin Shell with a Flowing Liquid. International Journal of Emerging Engineering Research and Technology. Volume 6, Issue 1, 2018, PP 1-14.

[12] Safarov I.I., Boltaev Z.I.Propagation of Natural Waves on Plates of a Variable Cross Section. Open Access Library Journal 2018, Volume 5, e4262, PP 1-29. https://doi.org/10.4236/oalib.1104262

[13] Safarov I.I., Boltaev Z.I. Methods for Assessing the Seismic Resistance of Subterranean Hydro Structures Under the Influence of Seismic Waves. American Journal of Physics and Applications. 2018; 6(2): PP 51-62. doi: 10.11648/j.ajpa.20180602.14. http://www.sciencepublishinggroup.com/j/ajpa

[14] Safarov I. I., Kuldashov N.U., Boltaev Z.I. Oscillations and Waves in a Layered Homogeneous Viscoelastic Medium. International Journal of Emerging Engineering Research and Technology . Volume 6, Issue 3, 2018, PP 27-32.

[15] Safarov I. I., Teshaev M.Kh, Ruziyev T.R. Methods for Assessing the Seismic Resistance of Subterranean Hydro Structures under the Influence of Seismic Waves. World Wide Journal of Multidisciplinary Research and Development (WWJMRD) 2018; 4(1): 128-139. Impact Factor MJIF: 4.25. www.wwjmrd.com

[16] Safarov I.I., Teshayev M.Kh.,Boltayev Z.I., Nuriddinov B.Z. Own Waves in a Cylindrical Shell in Contact with a Viscous Liquid. World Wide Journal of Multidisciplinary Research and Development ( WWJMRD) 2018; 4(1): 140-151. Impact Factor MJIF: 4.25. www.wwjmrd.com

[17] Safarov I.I., Teshaev M.Kh., Boltaev Z.I., Axmedov M.Sh. Of Own Vibrations of Cylindrical Bodies in the Deformable Medium-Specific Vibrations of Cylindrical Bodies in the Deformed Environment. World Wide Journal of Multidisciplinary Research and Development ( WWJMRD) 2018; 4(2): 42-53. Impact Factor MJIF: 4.25. www.wwjmrd.com

[18] Safarov I. I., Teshaev M. Kh., Boltayev Z. I. Harmonic Oscillations of Dissipative Viscoelastic Layered Bodies on a Deformable Half-Plane. World Wide Journal of Multidisciplinary Research and Development ( WWJMRD) 2018; 4(2): 54-58. Impact Factor MJIF: 4.25. www.wwjmrd.com

[19] Safarov I.I., Teshaev M.Kh., Boltaev Z.I . Own Vibrations of Bodies Interacting with Unlimited Deformable Environment. Open Access Library Journal. 2018, Volume 5, e4432 Access Library Journal, 5: e4432. PP 1-22.https://doi.org/10.4236/oalib.1104432

**Citation:** *Zafar Ihterovich Boltayev, Akhmedov Maqsud Sharipovich, Ro'ziyev To'lqin Razzoqovich, Hamroyeva Zilola Qahramonovna. (2018). "Calculation of Static and Dynamic Stress-Deformed State of Pipeline in Deformable with the Environment". International Journal of Emerging Engineering Research and Technology, 6(8), pp.31-42*