



UDC 539.4

## THE LOCAL STRESS STATE OF THE PIPELINE WITH AXIAL AND ANGULAR WELD MISALIGNMENT

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**Summary.** The paper is devoted to the evaluation of the stressed state of the shell containing two common defects in the weld shape – angular and axial misalignment at the same time. Expressions for stress resultants and stress concentration factors for axial and angular misalignment are proposed. The analytical approach is tested by means of numerical calculations for the internal pressure.

**Key words:** axial misalignment, angular misalignment, stress concentration coefficient, finite element method, cylindrical shell.

[https://doi.org/10.33108/visnyk\\_tntu2019.02.051](https://doi.org/10.33108/visnyk_tntu2019.02.051)

Received 03.05.2019

**Introduction.** Pipeline sections welding is the most important operation which greatly determines the quality of the carried out work. Pipeline sections welding is also the most essential technical operation affecting the structure quality and reliability. During welding it is necessary to provide pipes alignment achieved by qualitative welding edges alignment and assuring uniform gap between the joints for uniform root welding along the whole perimeter. In practice of constructing large-diameter pipelines it is difficult to comply with all welding specifications and avoid major defects in the weld shape.

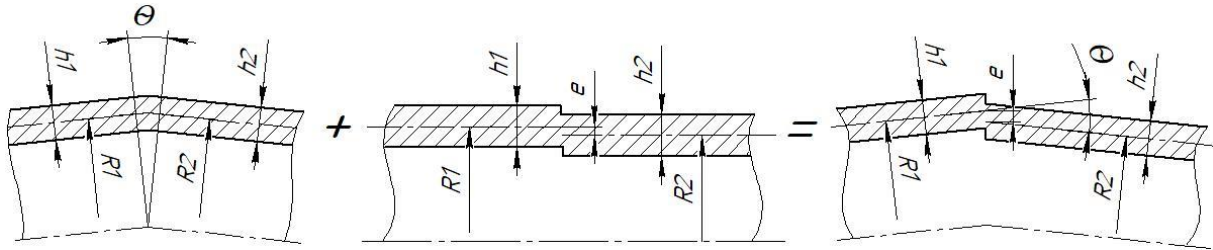
In previous publications we have analyzed stress state of pipes with shape defects in weld areas, such as: angular misalignment [1] and wall thickness variation (axial misalignment) [2]. According to [3] within tolerance weld joint of two pipe sections with angular misalignment  $\theta$  up to 3 degrees is accepted. The axial misalignment, which is the most common welding defect and is characterized by absolute joint displacement value  $e$  or dimensionless value  $\Delta = e/t$  ( $t$  is pipe wall thickness) in practice often exceed the accepted value. This is due to the fact that joint alignment process results, on the one hand, in the increase of welded elements assembly complexity and, on the other hand, in possible decrease of equipment service life caused by additional tension stresses [4]. Thickness variation of the connecting branches with wall thickness ratio greater than 1,5 is considered to be a defect [5]. Usually the smooth transition from one section to another with inclination angle which does not exceed 15 degrees should be provided in butt welds of pipes with different nominal thickness [6]. For above-ground pipelines the defect of axial misalignment type with the dimension up to 25% of wall thickness in most standards is acceptable for service.

The analytical method of force and strain parameters determination for stress state (SS) of pipes with circumferential weld imperfections in the form of axial misalignments was obtained in paper [2] on the basis of shell solutions. The accuracy of analytical approach was verified by numerical calculation. The analytical solution of pipeline stress state determination in the area of the weld axial misalignments under the action of internal pressure and bending

moment demonstrated by means of experimental and numerical data comparison was derived in paper [1].

This work is an extension of the previous works in this direction, considering combined form defect which contains angular and axial misalignment at the same time. It is reasonably common defect for welded pipelines.

Let us consider the stress strain state of the pipe joint by circumferential weld containing such defects as axial and angular misalignments (Fig. 1). The sections have the same wall thickness and different radii. Using the superposition principle the resultant stress state is the sum of SSS of two pipes: separately with angular and separately with axial misalignment (Fig. 1).



**Figure 1.** Scheme of the axial and angular misalignments:  $R$  – the average shell radius,  $h$  – the shell wall thickness,  $e$  – the absolute displacement of the weld edges,  $\theta$  – angle misalignments

**Basic mathematical dependencies.** The equation of the cylindrical shell equilibrium is well-known [7]:

$$\frac{\partial N_x}{\partial x} + \frac{\partial L}{R \partial \varphi} = 0, \tag{1}$$

$$\frac{\partial N_\varphi}{R \partial \varphi} + \frac{\partial L}{\partial x} + \frac{Q_\varphi}{R} = 0, \tag{2}$$

$$\frac{\partial Q_x}{\partial x} + \frac{\partial Q_\varphi}{R \partial \varphi} - \frac{N_\varphi}{R} = 0 \tag{3}$$

where  $N_x, N_\varphi$  – are axial and circular forces,  $L$  – is tangential force,  $Q_x, Q_\varphi$  – are lateral forces.

It is shown in paper [1, 8] that general solution (1)–(3) can be obtained by dividing the system into two separate ones: short solution, which rapidly damps along the axial coordinate and long solution which is slowly damped in axial direction. In case of axial and angular misalignment analysis under internal pressure action it is enough to have only short solution.

The short solution is characterized by four main parameters: radial displacement  $w$ , rotation angle  $\gamma_x$ , axial bending moment  $M_x$ , and lateral force  $Q_x$ . The final dependencies of the short solution [1, 8] required in our analysis are given below:

$$w = (C_1 F_1(x) + C_2 F_2(x)) \cos(n\varphi) \tag{4}$$

$$\gamma_x = - (C_1 (-aF_1(x) - bF_2(x)) + C_2 (-aF_2(x) + bF_1(x))) \cos(n\varphi) \tag{5}$$

$$M_x = H \delta \left( \left( C_1 \left( F_1(x) \left( (a)^2 - (b)^2 \right) + 2abF_2(x) \right) + \right. \right. \\ \left. \left. C_2 \left( F_2(x) \left( (a)^2 - (b)^2 \right) - 2baF_1(x) \right) \right) - \mu \frac{n^2}{R^2} (C_1 F_1(x) + C_2 F_2(x)) \right) \cos(n\varphi) \quad (5)$$

$$Q_x(x, \varphi) = D \left( C_1 \left( (3cd^2 - c^3) F_1(x) + (d^3 - 3dc^2) F_2(x) \right) + C_2 \left( 3cd^2 - c^3 \right) F_2(x) + \right. \\ \left. + (3dc^2 - d^3) F_1(x) - \frac{n^2}{R^2} \left( C_1 (-cF_1(x) - dF_2(x)) + C_2 (dF_1(x) - cF_2(x)) \right) \right) \cos(n\varphi) \quad (7)$$

$$N_x(x, \varphi) = \frac{n}{(a^2 + b^2)R} \left[ \left( C_1 (E_1a - E_2b) + C_2 (E_1b + E_2a) \right) F_1(x) + \right. \\ \left. + \left( C_2 (E_1a - E_2b) - C_1 (E_1b + E_2a) \right) F_2(x) \right] \cos(n\varphi) \quad (8)$$

The following functions are introduced here  $F_1(x) = e^{-a_n x} \cos(b_n x)$ ,  $F_2(x) = e^{-a_n x} \sin(b_n x)$ , Де  $\rho \cos \psi = a_n$ ,  $\rho \sin \psi = b_n$ ,  $\rho = \sqrt[4]{B}$ ,  $\psi = \frac{1}{2} \arccos \left( \frac{A}{\sqrt{B}} \right)$ :

$$A = \mu \frac{n^2}{R^2}, B = \left( \frac{n}{R} \right)^4 + 12 \frac{(1 - \mu^2)}{R^2 h^2} \quad (9)$$

If  $n=0$  then the short solution is the well-known axi- symmetrical end effect in cylindrical shells, here  $L = N_x \equiv 0$ .

In case of axial misalignment constants  $C_1$ ,  $C_2$ ,  $C_3$  and  $C_4$  are determined from the conditions of section conjugation at the point of their joining:

$$w_1 = w_2, \gamma_{x1} = \gamma_{x2}, Q_{x1} = Q_{x2}, M_{x1} - Ne = M_{x2} \quad (10)$$

If  $n=0$  from (9) we get that  $a = b = \alpha$ , where notation  $\alpha = \sqrt[4]{\frac{3(1 - \mu^2)}{R^2 h^2}}$  is introduced, then from (4), (5)–(8) and (10) it is easy to obtain the constants values:

$$C_1 = 0, C_2 = -\frac{1}{4} \frac{Ne}{H \delta \alpha^2}, C_3 = 0, C_4 = \frac{1}{4} \frac{Ne}{H \delta \alpha^2}, \quad (11)$$

here  $H = \frac{Eh}{1 - \mu^2}$ ,  $\delta = \frac{h^2}{12}$ ,  $N = -pR / 2$ . Knowing the constants  $C_1, C_2, C_3, C_4$  from (4)–(8) it is easy to get all force components, it should be noted that in this case  $M_\varphi = \mu M_x$ ,  $N_x \equiv 0$ :

$$N_\varphi = \pm Ne \alpha^2 R F_2(x) \quad (12)$$

$$M_x = \pm \frac{Ne F_1(x)}{2}, M_\varphi = \pm \frac{Ne \mu F_1(x)}{2}, \quad (13)$$

here sign «+» refers to the pipe with smaller diameter, and sign «-» with larger one.

Then from (12–13) it is easy to derive the concentration coefficients,  $K_x$  estimates the stress concentration in axial direction,  $K_\varphi$  – in circumferential direction:

$$K_x = \frac{\sigma_x}{\sigma_x^{HOM}} = 1 \pm \frac{6Ne}{pRh} F_1(x), \quad (14)$$

$$K_\varphi = \frac{\sigma_\varphi}{\sigma_\varphi^{HOM}} = 1 \pm \frac{\left( \frac{Ne\alpha^2 R F_2(x)}{h} + \frac{3Ne\mu F_1(x)}{h^2} \right)}{pR} h \quad (15)$$

In case of angular misalignment under internal pressure only two forces:  $N_x^b = \frac{-pR}{2}$  and  $N_\varphi^b = -pR$  act on the straight pipe section. Then the following boundary conditions should be used [1]:

$$L' \approx \frac{-pR}{2} \sin \theta \cos \varphi; Q' \approx \frac{pR}{2} \sin \theta \sin \varphi \quad (16)$$

Taking into account the symmetry, the boundary condition  $\gamma_x|_{x=0} = 0$  should be met. In short solution at  $n=1$  it is necessary to meet the boundary conditions for  $\gamma_x, Q_x$ , as they are the main variable. Analyzing expression (9) for thick shells  $R/h > 10$  at  $n=1$  we can easily understand that  $\left(\frac{n}{R}\right)^4 \ll 12 \frac{(1-\mu^2)}{R^2 h^2}$ , and value  $\psi \approx \frac{\pi}{4}$  as  $\frac{A}{\sqrt{B}} \approx 0$ . Thus, we can assume that  $a = b = \alpha$  similarly as at  $n=0$ . Hence from (5), (7) and (16) we derive two constants  $C_1, C_2$ :

$$C_1 = \frac{1}{8} \frac{pR \sin(\theta)}{H \delta \alpha^3}, C_2 = \frac{1}{8} \frac{pR \sin(\theta)}{H \delta \alpha^3} \quad (17)$$

If constants  $C_1, C_2$  3 (4)–(8) are known we can easily get all force components, and it should be noted that  $M_\varphi = \mu M_x, N_x = 0$ .

$$N_\varphi = -\frac{1}{2} \alpha pR^2 \sin(\theta) \cdot (F_1(x) + F_2(x)) \quad (18)$$

$$M_x = \frac{1}{4\alpha} pR \sin(\theta) \cdot (F_2(x) - F_1(x)) \quad (19)$$

Then from (18) and (19) we derive the stress concentration coefficients:

$$K_x = \frac{\sigma_x}{\sigma_x^{HOM}} = 1 + \frac{3}{\alpha h} \sin \theta (F_2(x) - F_1(x))$$

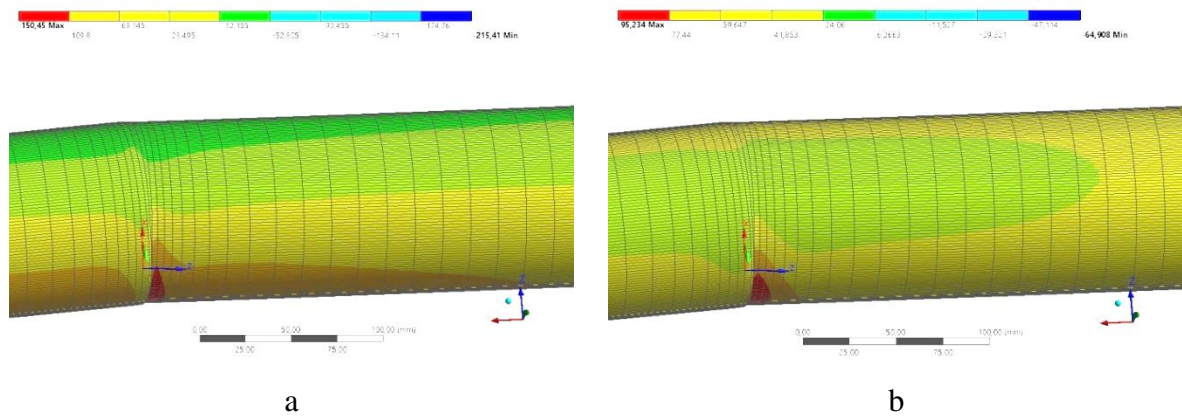
$$K_\varphi = \frac{\sigma_\varphi}{\sigma_\varphi^{HOM}} = 1 - \frac{1}{2} \alpha R \cdot \sin \theta (F_1(x) + F_2(x)) + \frac{3\mu \sin \theta}{2\alpha h} (F_2(x) - F_1(x)) \quad (20)$$

**Numerical verification.** The developed analytical approach for SS determination in the weld defect areas was tested by means of numerical modeling. The numerical modeling of axial and angular misalignment was carried out by finite elements (FE) method using ANSYS software. This problem was considered under internal pressure action.

In order to solve the given problem the shell FE model with idealized angular and axial misalignment was constructed (see Fig. 2). The model mid-radius is 374,65 mm, angle  $\theta = 4^\circ$ , wall thickness  $t = 12,7$  mm,  $e = 0,8$  mm.

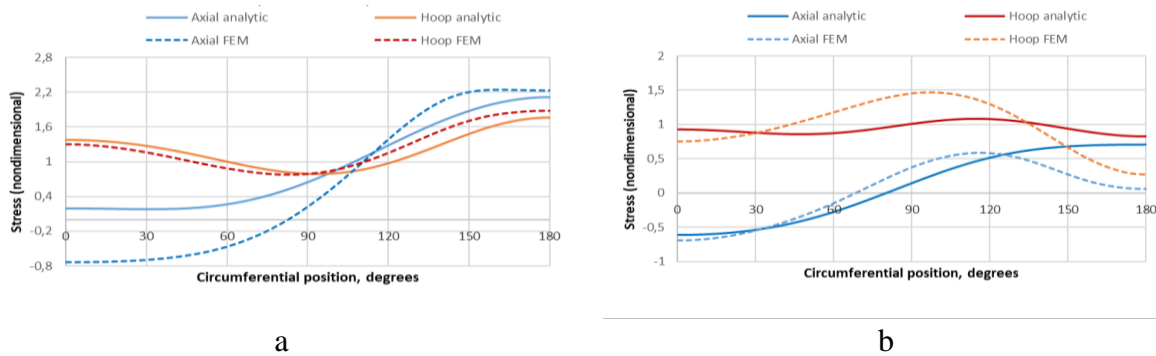
The boundary conditions providing the fixing of one pipe end and application of 24390H force to another end as well as 3,8 MPa internal pressure are used for shell stresses determination. Stress fields occurring in the model with complex defect under pressure are shown in Fig. 2. The calculations proved that the largest stresses occur in the weld area.

The distribution of axial and circumferential forces on the outer and inner surfaces calculated by analytical dependences (12), (13), (18), (19) is represented in Fig. 3. The numerical results are shown in Fig. 3 for comparison.



**Figure 2.** Normal stresses under pressure: a) in axial direction; b) in circular direction

The stress distribution curves are similar in character, their divergence is related to additional moments occurring under numerical model loading but analytical expressions provide convenient tool for estimation of stresses in complex defect.



**Figure 3.** Distribution of axial and circumferential stresses: a – on the outer surface, b – on the inner surface

**Conclusions.** Concentration coefficients separately for angular and axial misalignment are obtained and SSS of complex defect as superposition of simple ones is proposed in this paper. The carried out numerical modeling of the shell under internal pressure focused on testing the results of analytical solution (where we confined ourselves to short solution) revealed certain results discrepancy in the stresses concentration areas. The curve distribution character is similar and the results discrepancy can be explained by smoothing joint area displacement and angular misalignment in numerical model as well as occurrence of certain

local ones. However the obtained expressions for stress concentration coefficients are convenient for using in practical calculations.

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УДК 539.4

## ЛОКАЛЬНИЙ НАПРУЖЕНИЙ СТАН ТРУБОПРОВОДУ З ОСЬОВОЮ ТА КУТОВОЮ НЕСПІВВІСНІСТЮ ЗВАРНОГО ШВА

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**Резюме.** Присвячено оцінюванню напруженого стану оболонки, яка містить в собі одночасно два поширених дефекти форми зварного шва – кутову та осьову неспіввісність.

Не зважаючи на достатню кількість публікацій за темою дефектів зварних швів, мало уваги приділялося такому складному дефекту зварного шва, як комбінація кутової та осьової неспіввісності. В попередніх роботах було проаналізовано напружений стан труби окремо з кутовою неспіввісністю та осьовою неспіввісністю (різновісністю зварювальних труб) на основі оболонкових рішень і отримано аналітичний метод визначення силових та деформаційних параметрів для аналізу напруженого стану трубопроводів з недосконаlostями зварних швів.

Розроблено інженерний аналітичний метод визначення напруженого стану трубопроводу зі складним дефектом форми, який може бути представлений як комбінація простих. Тому отримано вирази для силових компонент й коефіцієнти концентрації напружень окремо для осьової та кутової неспіввісностей. Аналіз ґрунтується на раніше розробленому понятті «короткого» рішення. Для верифікації запропонованого аналітичного методу побудовано скінченноелементну модель комбінованого дефекту зварного шва та виконано чисельний аналіз при дії внутрішнього тиску з використанням розрахункового комплексу ANSYS. Чисельне моделювання у порівнянні з аналітичною процедурою демонструє розходження результатів у зоні концентрації напружень, в околі зварного шва, за рахунок виникнення локальних моментів у чисельній моделі. Проте запропоновані коефіцієнти концентрації напружень зручно використовувати у практичних розрахунках вказаних типів дефектів поперечно зварного шва, для швидкого оцінювання допустимості дефектів. Детальніше оцінювання напружено-деформованого стану дефекту, у випадку невиконання умов міцності за попереднім розрахунком, вимагає побудови чисельної моделі.

**Ключові слова:** осьова неспіввісність, кутова неспіввісність, коефіцієнт концентрації напружень, метод скінченних елементів, циліндрична оболонка.

[https://doi.org/10.33108/visnyk\\_tntu2019.02.051](https://doi.org/10.33108/visnyk_tntu2019.02.051)

Отримано 03.05.2019