

COMPUTATION OF A PLATE-KEY CLOSURE OF NATURAL GAS FILTER

JAN RYŚ

Institute of Mechanics and Machine Design, Cracow University of Technology
e-mail: szymon@mech.pk.edu.pl

In the paper analytical and numerical results of equivalent stresses in the closure of pressure vessel which is a main unit of a natural gas filter are compared. The closure unit consist of a circular plate, housings welded to a cylindrical shell of the vessel and blocking keys in the form of segment rings. The leakproof is achieved by a rubber gasket between the housing and circular plate.

The results for the gas filter made by Stalbud-Tarnow at nominal parameters $Dn = 1400$ mm; $p_0 = 5.5$ MPa; $t_0 = 50^\circ\text{C}$ and hydraulic test pressure $p_h = 9.9$ MPa are presented.

Key words: pressure vessel, gas filter, plate-key closure, nominal and test pressure

1. Introduction

The present study aims at comparison between the results of analytical and numerical (FEM) computations of the stress state under the conditions of local stress concentration in a pressure vessel.

The results of numerical and analytical computations presented in the paper demonstrate a good agreement outside the notch region. This means that analytical computations can be used in the case when the stress concentration factor in the notch region is assumed at the level of 1.30. The formulas given in [6] can be very useful at the initial stage of designing of this type of closure. When using the FEM, the results can be verified and made more precise, taking into account the possibility of local allowable plasticization of material during hydraulic tests.

When considering the structural strength of the vessel, the application of different methods to determination of the allowable stresses in the notch region

in comparison with the areas without a concentration (e.g., in the cylindrical part of constant-thickness vessel) should be emphasized.

The natural gas filter manufactured by Stalbud-Tarnów of the following parameters: vessel diameter $Dn = 1400$ mm, nominal pressure $p_0 = 5.5$ MPa, nominal temperature $t_0 = 50^\circ\text{C}$, and hydraulic test pressure $p_h = 9.9$ MPa has been used in numerical computations.

2. Description of the design

The closure of a pressure vessel working as a natural gas filter consists of the following three important components (see Fig.1a): cover plate (1), closure body in the form of a appropriately shaped ring connected with the cylindrical part of the vessel (2), and locking ring consisting of four segments used as keys (3) which should be analysed. Fig.1b shows how the keys (shaped as four half-rings) are mounted in the vessel body. After installing the cover plate, the keys are locked radially in a groove in the vessel body by means of a special locking mechanism. The tightness of this closure is ensured by a rubber seal between the body and the bottom surface of the cover plate.

Under internal pressure, the structure designed in this way is subjected to axially-symmetrical deformation involving four stress components: σ_θ , σ_z , σ_r , τ_{rz} , defined in the r, z, θ cylindrical co-ordinate system, where z is collinear with the vessel axis of symmetry, r is the radial direction, and θ is the circumferential direction.

According to the Huber-Misses theory (Kozłowski, 1968; [5]), the equivalent stress determining the limit of elastic strain for steel structures (failure criterion) can be written as

$$\sigma_H = \sqrt{\sigma_r^2 + \sigma_z^2 + \sigma_\theta^2 - \sigma_r\sigma_z - \sigma_\theta\sigma_z - \sigma_r\sigma_\theta + 3\tau_{rz}^2} < R_e \quad (2.1)$$

where R_e is the yield point of the material.

For the purposes of numerical computations it has been assumed that $R_e = 290$ MPa (material: 18G2A), strain hardening modulus for steel is $E_t = 20000$ MPa (according to Łączek, 1998), Young modulus and Poisson ratio for steel are $E_t = 2 \cdot 10^5$ MPa, and $\nu = 0.3$.

In the selected cross-sections of the considered structure some stress components given above can be neglected owing to their low values. This is helpful in the analytical method of computation. For example, in the cylindrical shell of the vessel, σ_r and τ_{rz} are neglected since they are small in comparison

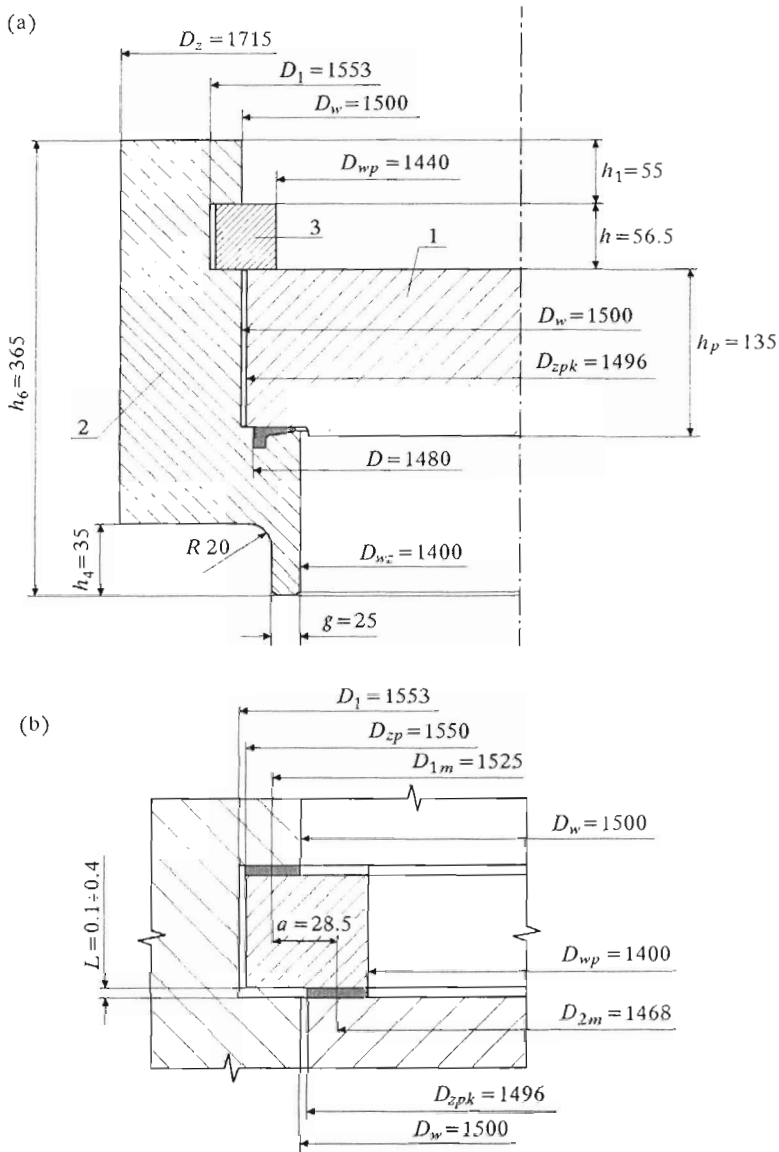


Fig. 1. Design of the gas filter closure; D – diameter of rubber seal, D_{wz} – inside diameter of cylindrical vessel, g – thickness of cylindrical vessel, D_w, D_z – inside and outside diameters of closure body, D_{zpk} – diameter of cover plate, h_p – thickness of cover plate, D_{wp}, D_{zp} – inside and outside diameters of locking ring, D_{1m}, D_{2m} – major diameters of the locking ring support, h – thickness of the locking ring

with σ_θ and σ_z . Similarly, σ_z and τ_{rz} can be neglected in a circular, axially-symmetrical plate (Timoszenko and Wpinowski-Krieger, 1959).

It should be noted that the closure body has small-radius notches, and local stress concentrations appear in these regions; as well as in the areas where the locking ring contacts the closure body and supports the cover plate. If, during the hydraulic test, the local strain region is small and covers small area around the notch (up to 5% of the cross-sectional area), the material will be locally hardened, and during the second hydraulic test the structure will experience elastic deformation only. As a result, the maximum strain in the notch region during the hydraulic test should be limited so as not to exceed more than several times the elastic strain, e.g.

$$\varepsilon < 3\varepsilon_s = 3 \frac{R_e}{E} \quad (2.2)$$

These conditions can be verified only by numerical methods, the Finite Element Method (FEM) being the most useful. In this study, the ANSYS program [4] (version 5.0) was used to solve the example with $Dn = 1400$ mm, and $p_h = 9.9$ MPa, for a material demonstrating linear and isotropic stress-hardening.

The following conclusions were drawn from the hydraulic tests at $p_h = 9.9$ MPa:

- The weakest parts of the structure (revealing the lowest strength) are the upper area of the groove in the closure body, and the cover plate edge.
- The structure will withstand the hydraulic test provided that $R_e \geq 290$ MPa.
- If the condition $R_e = 290$ MPa is fulfilled, the structure will experience limited plastic strain presented in Eq (2.2).

2.1. Allowable stresses in nominal conditions (p_0, t_0)

The stress concentration in the notch has a local nature, therefore the allowable stresses should be computed taking into account that the yield point can be slightly exceeded in the hydraulic test. In accordance with Eq (2.2), the allowable stresses in the notch region, for the nominal pressure and temperature, can be calculated as

$$k_k = \frac{R_e t_0}{x} \left(1 + 3 \frac{E_t}{E}\right) = \frac{290}{1.8} \left(1 + 3 \frac{2 \cdot 10^4}{2 \cdot 10^5}\right) = 209.4 \text{ MPa} \quad (2.3)$$

where x is the safety factor (determined by technical inspection regulations).

Similarly, according to [7], the allowable stress in the notch is assumed at a more liberal level in the sense of strength

$$k_k = \frac{R_e t_0}{x} 1.5 = 241.6 \text{ MPa} \quad (2.4)$$

and the local allowable contact pressure is

$$p_{allow} = R_e t_0$$

If we denote the length of the complete cross-section of the structure in the notch region by l_D , and the distance measured along the cross-section, starting from the notch, by x , then the strength criterion in this cross-section can be written as (for the nominal pressure and temperature)

$$\begin{aligned} \sigma_H &\leq k_k && \text{for } x \in < 0; 0.05l_D > \\ \sigma_H &\leq k && \text{for } x \in < 0.05l_D; l_D > \end{aligned} \quad (2.5)$$

where

$$k = \frac{R_e t_0}{x}$$

The conditions (2.5) and (2.3) or (2.4), respectively, should be applied if the computational method makes it possible to find the stress concentration state in the cross-section with the notch, and the actual contact pressure distribution. In the case of the considered axially symmetrical structure, it is necessary to apply the Finite Element Method (FEM), Boundary Element Method (BEM), or Finite Difference Method (FDM). If classical methods of strength analysis are used, which do not allow for stress concentration in the notch region, the failure criterion in the form

$$\sigma_H \leq k_k \quad k = \frac{R_e t_0}{x} \quad (2.6)$$

can be unreliable.

3. Results of numerical and analytical calculations

In order to compare the results of analytical and numerical computations, the distribution of stresses has been determined in the selected cross-sections

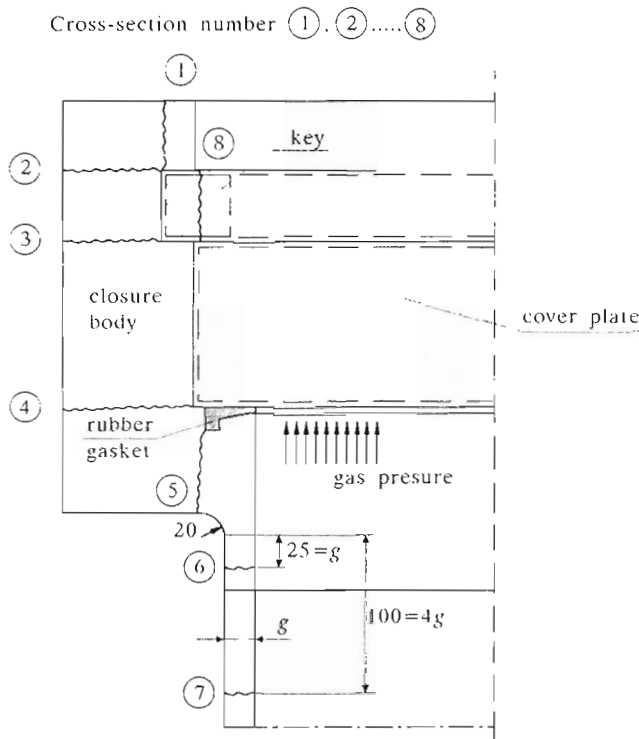


Fig. 2. Cross-sections to determine the state of stress

of the closure body. These cross-sections are marked in Fig.2. In each cross-section all stress components were determined, and the equivalent stress σ_H (to check the failure criterion) was computed.

The stress components and the equivalent stress at the cross-section number 5 are shown in Fig.3 (FEM) and Fig.4 (analytical method).

The concept of the analytical method presented in [6] consists in determining the internal forces in the area when the closure body is connected with a cylindrical part of the vessel, and in assuming a triangular contact pressure distribution on the key, with zero rotation angle owing to the locking action. Analytical computations were made using the *korppok.mcd* program developed for the company contracting the computations. The program allows the user to change the value of any design variable, and to obtain the results immediately.

In Table 1 the results of numerical (FEM) and analytical computations are compared. The results demonstrate a good agreement outside the notch vicinity. The results are satisfactory in the quantitative sense, and the design

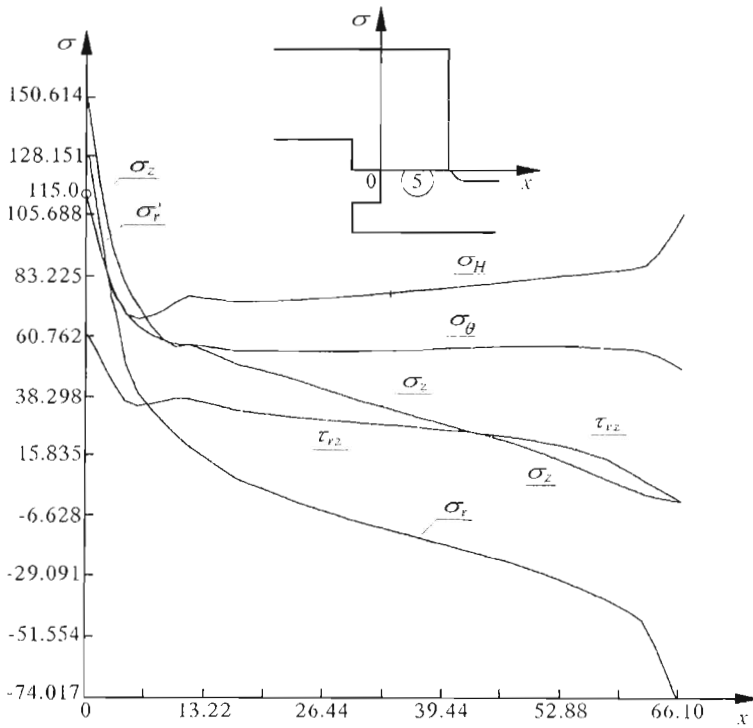


Fig. 3. Cross-sections number 5; the stress components determined by FEM

has been approved after the hydraulic test. Additionally, the difference between the stresses σ_θ and σ_z measured in the shell and in the closure body during the hydraulic test and the stresses computed using FEM software did not exceed 6%.

This means that using the analytical method developed is it possible to prepare a preliminary draft design of the structure, provided that the stress concentration factor in the notch is assumed at the level of 1.30. Furthermore, the analytical method gives us the possibility to establish optimisation procedure limiting the mass of such a closure.

An important result of computations is the possibility of changing the geometry of the cover plate and the closure body so as to reduce the closure weight by 10%, maintaining simultaneously the equivalent stresses level.

Polish Technical Inspection has approved this method of strength calculation.

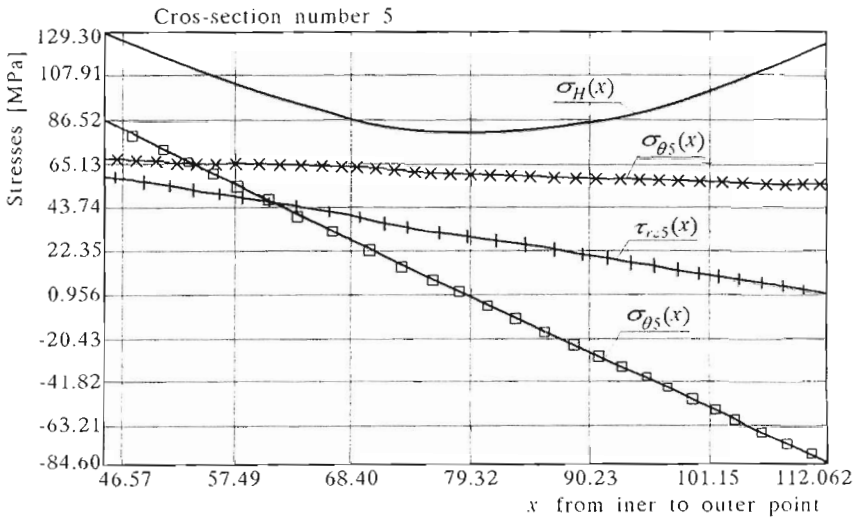


Fig. 4. Cross-sections number 5; the stress components determined analytically and equivalent stress $\sigma_H(x)$

Table 1. Comparison between the results of computations for the gas filter closure ($Dn = 1400$ mm, $p_0 = 5.5$ MPa, $t_0 = 50^\circ\text{C}$)

Cross-section number according to Fig. 2	Inner point of the cross-section		Central point of the cross-section		Outer point of the cross-section	
	Analytical method	FEM	Analytical method	FEM	Analytical method	FEM
7	125.9	136.9	131	141	153.7	160.9
6	115.9	104	120.8	114	154.6	144.3
5	129.3	115	79.2	78.5	121.7	115
4	99.6	133.9	40.2	19.5	108.5	62.4
3	134.2	182.5	41.4	50.2	83.8	65.8
2	134.2	166.2	41.4	40	83.8	74.8
1	152.9	185.1	88.8	57.9	69.1	40.2
	Plate-side point		Central point		Groove-side point	
8	151.4	177.1	87.5	102.2	98.7	90.5

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Obliczenia płytowo-wpustowego zamknięcia filtra gazu

Streszczenie

Celem pracy jest porównanie wyników obliczeń analitycznych i numerycznych MES stanu naprężeń w warunkach, gdy w konstrukcji występują lokalne koncentracje naprężeń. Przedstawione wyniki obliczeń analitycznych i numerycznych wykazują dobrą zgodność poza strefą karbów. Przyjmując w strefach karbów współczynnik koncentracji naprężeń na poziomie 1.3 można metodą analityczną wstępnie projektować konstrukcje tego typu o dowolnych wymiarach, przy zastosowaniu dowolnych materiałów i ciśnień, a stosując metodę MES można zweryfikować i uściślić wyniki. Ze względu na wytrzymałość konstrukcji na podkreślenie zasługuje inny sposób określenia naprężeń dopuszczalnych w strefie karbów w porównaniu ze strefami, gdzie taka koncentracja nie występuje, np. w walcowatej części zbiornika o stałej grubości. Obliczenia prowadzono dla filtra gazu o średnicy zbiornika $D_n = 1400$ mm na ciśnienie obliczeniowe $p_0 = 5.5$ MPa, $t_0 = 50^\circ\text{C}$ i ciśnienie próby hydraulicznej $p_h = 9.9$ MPa.