Industrial safety of pressure vessels – Structural integrity point of view

Aleksandar Sedmak[1], Mahdi Algool[2], Snezana Kirin[3], Branislav Rakičević[1], Ramo Bakić[4]

[1]*Faculty of Mechanical Engineering, University of Belgrade, Belgrade, Serbia*
[2]*University of Sirte, Sirte, Libya*
[3]*Innovation Center of the Faculty of Mechanical Engineering, Belgrade, Serbia*
[4]*Technical School, Tutin, Serbia*

Abstract

This paper presents different aspects of pressure vessel safety in the scope of industrial safety, focused to the chemical industry. Quality assurance, including application of PED97/23 has been analysed first, followed shortly by the risk assessment and in details by the structural integrity approach, which has been illustrated with three case studies. One important conclusion, following such an approach, is that so-called water proof testing can actually jeopardize integrity of a pressure vessel instead of proving it.

Keywords: pressure vessel safety, structural integrity, water proof testing, risk assessment.

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Pressure vessels, used in chemical industry, commonly have the form of cylinders, spheres, ellipsoids or some combination of these. In practice, vessels are usually composed of a pressure-containing shell together with covers and flange rings, typically connected by welding. Their main purpose is to contain a media under pressure and temperature to ensure safe and long life.

The most critical part of a pressure vessel is welded joint, because crack-like defects are inevitable, due to very nature of welding process [1]. Therefore, no wonder that there are numerous references dealing with weldment behavior in presence of crack [2–11]. This effect is also evident from many failures, as illustrated by three examples below.

The first example concerns the storage tank for liquid carbon dioxide (CO_2, produced of high strength micro-alloyed steel DIN St.E460, trade mark Nioval 47) (Steelworks, Jesenice), thickness 14 mm, with micro--cracks in ferrite-austenite welded joint, as shown in Fig. 1 [12,13]. Two new connections at the upper lid had been made to connect the outer freon unit to the inner built-in heat exchanger. Since Nioval 47 is not produced in form of tubes, the manufacturer used tubes and flanges of austenitic steel, 26.9 mm in diameter, 2.6 mm wall thickness. It was difficult to ensure quality of heterogeneous welded joints with large difference in thicknesses. Therefore, reinforcements of similar thicknesses were welded to the lid, using rutile electrode, alloyed with 29% Cr and 9% Ni, and then the connecting tubes were welded to the reinforcement. Anyhow, due to metallurgical problems with dissimilar base metals, micro-cracking appeared in HAZ, as shown in Fig. 1, leading to the leakage of this pressure vessel.

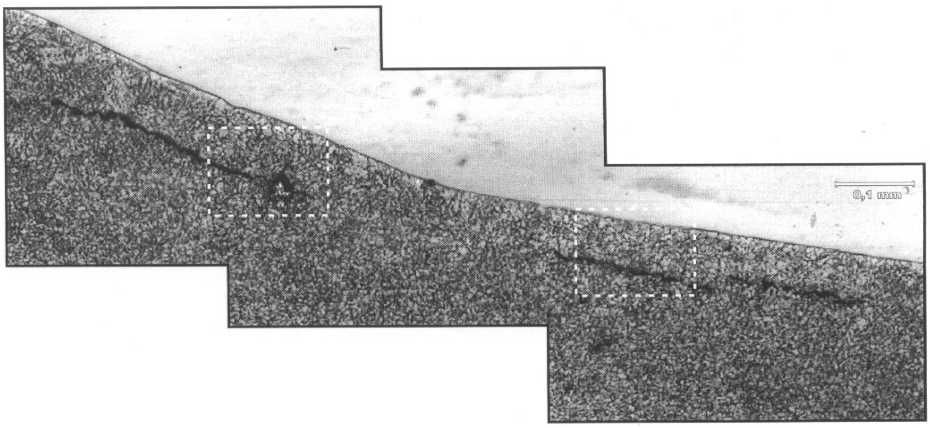

Figure 1. Spherical pressure vessel for ammonia [12].

Correspondence: A. Sedmak, Faculty of Mechanical Engineering, University of Belgrade, Belgrade, Serbia.
E-mail: asedmak@mas.bg.ac.rs
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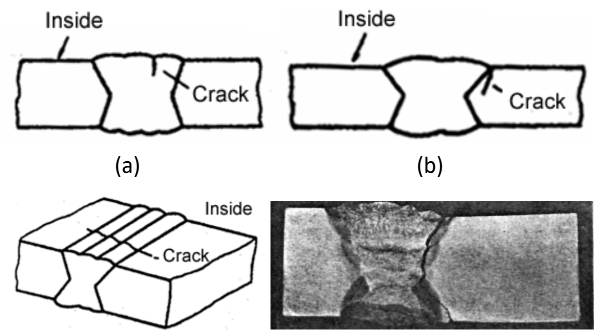
As the second example, leakage of large spherical tank, used for storage of ammonia, is briefly presented. It was caused by undetected micro-cracks in welded joint, which have grown through the thickness during proof testing (cold-water test with pressure up to 50% above the operating pressure) [12]. The testing of storage tanks before and after inspection has clearly shown the adverse effect of proof test in service, since it has indicated large number of new cracks in the positions of the "old" ones. Typical positions of longitudinal and transverse cracks on inner vessel wall of tank are shown in Fig. 2a–c, whereas the macroscopic view of through crack causing leakage is given in Fig. 2d.



(a) (b) (c) (d)

Figure 2. Micro-cracks in HAZ in NIOVAL 47 steel welded joint [13]; a–c) typical positions of longitudinal and transverse cracks on inner vessel wall of tank; d) the macroscopic view of through crack causing leakage.

As the third example the leakage in the tube-to--tubesheet connection is shown, causing failure of feed water heater components in ammonia converters in the Chemical Industrial Plant (HIP) in Pančevo.

Having in mind the complexity of presented problem, the basic aim of this paper is to present different aspects of pressure vessel safety in the scope of industrial safety, focused to the chemical industry. Quality assurance, including application of PED97/23 has been analysed first, followed shortly by the risk assessment approach. More attention is given to the structural integrity approach, which is described in some details and illustrated with three case studies.

QUALITY ASSURANCE - PED 97/23

In order to avoid failure of pressure equipment the defects and heterogeneities have to be under strict control and inspection, especially welded joints [12]. The basic approach is to apply quality assurance system, including codes and rules defined for that purpose, like Pressure Equipment Directive (PED 97/23/EC) [14–17], which are applicable only to new products. The second, more advanced approach, can be the risk based inspection, maintenance and control, as described in [18–21]. It is relatively new approach, still developing and potentially the most important one, especially if combined with other approaches. Finally, the third approach, which will be analysed and discussed in details here, the structural integrity assessment, applicable also in the case of damaged pressure equipment, when the decision of its further exploitation is possible under given condition and only after careful and detailed consideration.

In any case, the operational safety of welded pressure vessels primarily depends on weldments behavior. The approach, accepted in standards for weldment design, is to define the acceptable defect size. All efforts in material production and improvements in welding and non-destructive testing techniques, together with strict requirements in quality assurance, cannot exclude the appearance of crack-like defects during fabrication, stress relieving, hydrostatic proof tests or operational service [1,12]. Furthermore, in real welded pressure vessels stress concentration caused by geometrical changes, including inevitable weldment imperfections, such as angular distortion or misalignment, causes local plastic strains. In these circumstances the question arises of how cracked welded joint will behave.

Full scale tests of welded pressurized equipment are the most informative when safety is considered [12]. In some cases they are inevitable despite high cost because they can give realistic answers relating the service behavior of welded joints. Hydrostatic pressure proof test can be classified as the full scale test.

Directives for technical standards for stationary pressure vessels prescribed that the regular periodic proof pressure test of vessels should be performed before six years in service, if not otherwise required by the regulations on the technical standards for certain type of pressure vessels and stored substances.

Hydrostatic pressure for proof test is often calculated using the formula $p_i = 1.3p_r$, where p_i is proof test and p_r is the design pressure. The logic behind this approach is that once a pressure vessel has withstood proof test, it will be safe in the exploitation under design pressure. Nevertheless, experience indicates more complex situation, as was the case with number of large pressure vessels, used in chemical industry. Already mention here, where the proof test has provoked cracking and leakage, instead [12].

RISK BASED APPROACH

The Extensive European project RIMAP, from 2001 until 2004, was introduced to offer a European standard for risk based management (RBM) [18–21]. It has produced four industry specific workbooks (petrochemical, chemical, steel and power generation industries), aimed to provide more specific guidance on how to apply the RIMAP approach. However, this approach is too complex, and will not be considered here. Instead, we present here only the risk matrix approach, as illustrated in Table 1. This approach uses well-known definition of risk being the product of the probability and the consequence [18–21].

In the risk matrix shown in Table 1, consequences are categorized, based on several parameters (health, safety, environment, business, security) as A to E; A indicates low, almost negligible consequences, and E refers to fatal and serious consequences. Probability categories are graduated 1 to 5, starting with very unlikely event, once in over a 100 years (1×10^{-4}), ending with highly probable event occurring at least once in a year (1×10^{-1}).

Table 1. Scheme for risk-based qualitative evaluation of maintenance

Probability category	Consequence category				
	A	B	C	D	E
5					Very high
4				High	risk
3			Medium	risk	
2		Low risk	risk		
1	Very low risk				

It is self-evident that consequences of failure of pressure vessels used in chemical industry can be extremely serious, even catastrophic, as in the case of Bopal disaster, indicating category E, as the rule. For safe and reliable use of pressure vessel in chemical industry it is of utmost importance to assure extremely low frequency of such events, classified as probability category 1, since only in this case risk is not bigger than medium. This can be achieved by special measures in all steps of design, construction and operation, including structural integrity assessment. The application of risk matrix to ensure safety of critical components in petrochemical industry, mostly pressure vessels, is illustrated and explained in more details in [18–21].

It is also interesting to note that pressure vessels are treated in somewhat similar way by PED 97/23, since they have to be categorized from 0 to 4, according to pV value (p stands for pressure, V for volume). Although this looks like risk based approach, one should notice that consequence and probability cannot be separated in this approach, making it one-dimensional, or let say risk vector approach.

Finally, one should notice that the probability of event is actually increased by water proof test, "pushing up" risk along the vertical axis of the risk matrix. Therefore, this approach clearly, even graphically, indicates possible problems with water proof testing.

STRUCTURAL INTEGRITY ASSESSMENT

In-service behavior of many structural components revealed that cracks lead to the fatal failure. One possibility to prevent such a scenario is to use quality assurance system. However, it cannot cover all situations in which pressure equipment can operate, and this system is not applicable completely for pressure equipment in operation. Problem might be solved by applying fitness-for-purpose approach and fracture mechanics analysis for a cracked component, in the scope of its structural integrity assessment.

This approach was first used in Alaska pipeline in USA [22]. After final non-destructive testing (NDT), before pipeline introduction in exploitation, a large number of shallow cracks has been detected, not acceptable according to standards. The repair of defective pipeline would have been too expensive, and possible being more harmful than beneficial (if provoking new cracking), so additional investigation has been performed by the National Institute of Standards and Technology (NIST) to assess cracks significance by fracture mechanics approach. It was found that less than 5% of detected cracks had to be repaired according to structural integrity approach, and 95% of them were accepted without affecting structural integrity and in-service safety [22]. This has been accepted by the National authorities as the permissible exemption from the standard procedure, since it has proved, in reasonable and conservative manner, that the pipeline safety will not be jeopardized.

The practical application of fracture mechanics is from the very beginning based on twofold interpretation of its parameters: they represent loading and structural geometry, including a crack, on one hand side, whereas their critical values represent material properties and crack resistance, on the other hand side. Engineering practice had been changed adopting fracture mechanics criteria instead of traditional and rigorous standards on admissible defects regarding necessity of repair. This enabled acceptance of fracture mechanics analysis as a sound base for allowable exclusions from existing standards under certain circumstances, *i.e.*, if such analysis results in justified and conservative (safe) assessment of the integrity of a structure.

Fracture mechanics has connected three variables (stress, defect dimensions and fracture toughness), as shown in Fig. 3, enabling evaluation of the third value, based on two known variables. Based on this, several procedures were developed in order to assess structural integrity [23–28]. Here, the comparison of material crack resistance (expressed by J integral in experimentally obtained J–R curve) and crack driving force (CDF), obtained by analytical or numerical model, will be used.

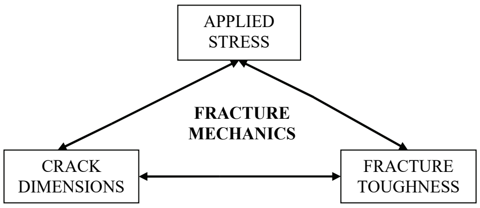


Figure 3. Fracture mechanics triangle.

It is possible to establish criterion for fracture prediction, applying crack growth resistance curve, expressed by J-R curve, and CDFs, as shown in Fig. 4 for five different levels of load (stress). The simple explanation of this approach is that crack of length a_0 will grow in stable manner under certain load, in the case presented in Fig. 4 it is stress σ_4, until it reaches its critical value (point A), *i.e.*, length $a_0+\Delta a$, when the growth will become unstable, leading to the fracture of a component. It is very important to understand fully such a behavior and eventual consequences of stable to unstable crack growth for the safe exploitation of a pressure vessel.

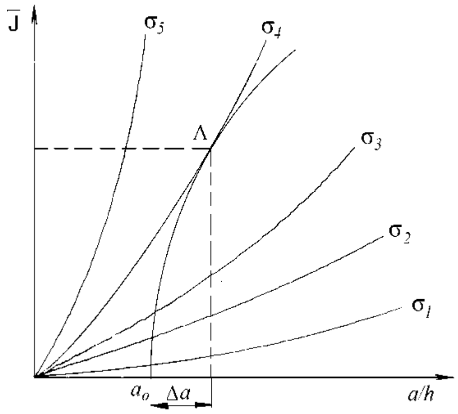


Figure 4. Procedure for fracture prediction based on crack resistance J–R curve.

CASE STUDIES

Leakage of CO_2 storage tank [12,13]

As already mentioned in the introduction, during the water proof testing of storage tank for liquefied carbon dioxide droplets of water had been revealed on the outer wall of its manhole [12,13]. The storage tank is of cylindrical form, thermally insulated, 12.5 m^3 in volume. The mantle and tank lids are produced of high strength micro-alloyed steel, 14 mm in thickness. The lowest operating temperature of the tank is –55 °C, the highest operating pressure 30 bar, the proof pressure test 39 bar. The storage tank is classified as II class of pressure vessels, according to *pV* criterion. A general view of tank and position of a manhole is presented in Fig. 5.

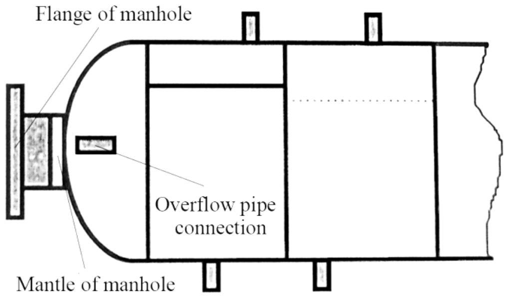
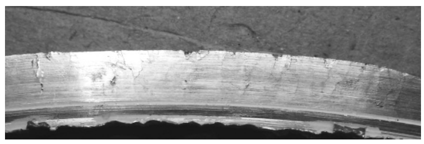


Figure 5. General view of tank for liquefied CO_2 storage [13].

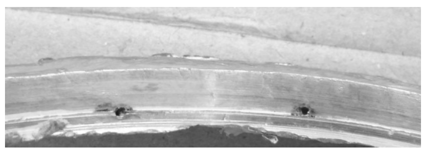
After removing the thermal insulation from the manhole, the moisture is located around the welded joint between the mantle and flange neck. The manhole consisted of mantle, produced of the same high strength micro-alloyed steel, 10 mm thick, and a flange casting of high alloyed austenitic steel [12,13]. The flange and the manhole are welded by shielded manual arc welding (SMAW), using high alloyed austenitic consumable [12,13]. Detailed analysis of austenite-ferrite weldment behaviour is given in [29–31].

Between the flange and manhole mantle a butt welded joint was made. Flange necking towards the welded joint ended by a cylindrical part of diameter and thickness that are equal to the diameter and thickness of the manhole mantle of width 30 mm. Cracks have been detected in the cylindrical part of the flange neck, in a limited zone, with lengths typically between 25 and 27 mm. In the zone of individual cracks, only short cracks, 1 to 2 mm long, were detected.

After emptying the tank and opening the manhole, its inner side was tested. On the inner side two larger pores are found, as well as the individual cracks, 1 to 2 mm in length. Figure 6 presents the cross section through the center of cylindrical part of flange neck, with the zone with the highest cracking density shown in Fig. 6a, and pores in Fig. 6b. A large number of cracks, approximately perpendicular to the flange surface, are visible. Testing by dye penetrants indicated a large number of cracks penetrating to different depths, some of them through the thickness.



(a)



(b)

Figure 6. Cross section views: a) network of cracks, b) pores on inner side [13].

In Fig. 6b the cross-section close to the pores detected on the inner side of a manhole is presented. Two pores, which are in fact the continuation of pores detected on the inner side of the manhole, are clearly visible. Most intensive leakage on the outer side of the manhole is revealed just opposite of these pores, caused by cracks, penetrating to the outer surface of the tank. It was concluded that the detected cracks affect the safety operation of the tank, and hence the complete structural integrity assessment was performed, as follows [12,13].

First, SEN(B) specimens have been tested to obtain J–R curve, as the material property related to the crack resistance, in accordance to the Standard ASTM E1820, both at ambient temperature, 20 °C and at operating temperature, –60 °C. From this curve it is possible to determine critical J_{Ic}, a measure of fracture toughness, convert it to plane strain fracture toughness K_{Ic}, and verify crack significance in respect to brittle fracture. The complete J–R curve is even more useful, enabling determination of stress level for initiation of stable crack growth, in combination with Crack Driving Forces (CDF) [12].

As an example of this procedure, the J–R curve obtained for the heat-affected-zone (HAZ) is presented in Fig. 7, and applied to the CDFs diagram to assess structural integrity for cracked storage tank, as shown in Fig. 8.

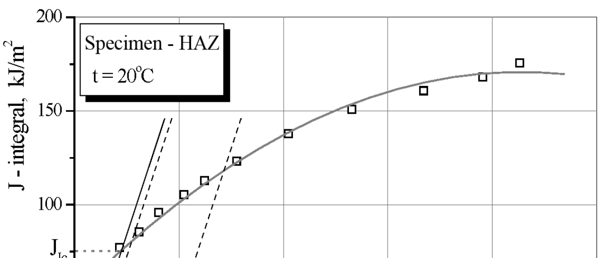


Figure 7. J–R curve and determined value of J_{Ic}, for specimen in HAZ [13].

One can see from Fig. 8 (the critical point) that the crack can grow from 5.76 mm (0.4 mm×14 mm) up to 7.7 mm in stable manner under the pressure 306 MPa (72% of yield strength). Once crack has reached the critical value, conditions for unstable crack growth, *i.e.*, brittle fracture, are fulfilled. It is very important to notice that such a scenario is also possible in real pressure vessel, especially if maximum stress approaches the Yield Stress, which is typical situation during water proof test. From Fig. 8, one can see that stresses less than 72% of yield strength will not provoke crack growth. Using this example, one can think of situation in which the design stress is safely below the critical one, let say 60% of yield strength, but 30% of additional water proof stress will not be, since it would reach 78% of yield strength.

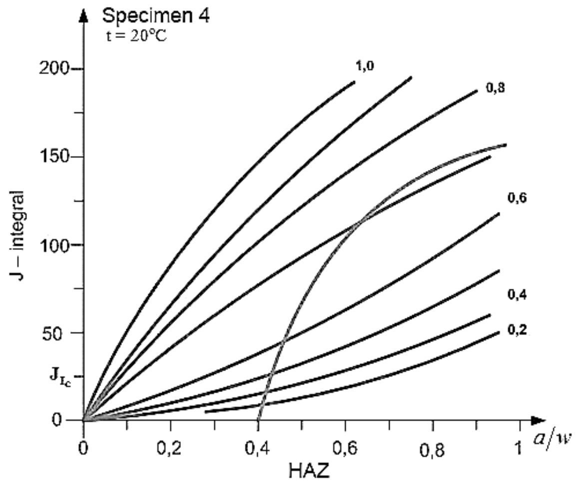


Figure 8. Structural integrity assessment for storage tank [13].

Leakage of large spherical storage tank [32,33]

Experimental and numerical analysis of weldments fracture behavior is performed to assess the structural integrity of spherical storage tanks in Macedonia, [32,33]. The analysis was performed on two spherical storage tanks for ammonia storage (volume 1000 m^3, diameter D = 12400 mm and wall thickness t = 30 mm, Fig. 9). The operating pressure was p = 16.5 bar and proof test pressure p = 25 bar was applied in 1998 and 1999 together with NDT. The tanks have been constructed in 1979 using microalloyed steel DIN St.E460, (yield strength $R_{p0.2}$ = 460 MPa, ultimate tensile strength R_m = 600 MPa, elongation A_5 = 28%), welded by electric arc procedures with basic consumables.

Figure 9. Spherical pressure vessels in Macedonia [33].

Different NDT methods (ultrasonic, dye penetrant and magnetic particles) have been used to test welded joints. The last one, in combination with fluorescent light, turned out to be the most efficient for detection of surface and small subsurface cracks. A large number of transverse cracks in weld metal and longitudinal cracks in HAZ along fusion line of inner welded joints have been detected, like one shown in Fig. 10 [33]. The longitudinal cracks were considered as more dangerous due to their size (highest length up to 300 mm, depth up to 13.5 mm) and position. The cracks were caused by improper welding, residual stresses effect and stress corrosion due to hostile environment [33].

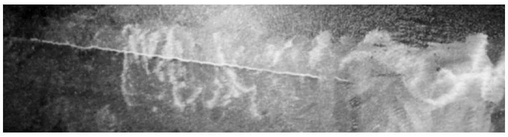


Figure 10. Cracks along fusion line [33].

For detailed examination tensile specimens (width W = 24 mm, thickness B = 20 mm, length L = 300 mm) were machined out of the spare trial welded plate (400 mm×400 mm).In addition, plates 500 mm×500 mm were cut out from both tanks to assess crack significance and structural integrity, using specimen for chemical analysis, mechanical properties and metallographic investigation.

Fracture mechanics testing for determination of crack resistance in the form of J–R curves and critical J integral value, J_{Ic}, was performed using tensile, single edge notched specimens. Cracks in specimens had been produced by electrical discharging procedure, since by standard fatigue procedure it was not possible to locate the crack tip in the very narrow HAZ regions (fine grain – FGHAZ or coarse grain – CGHAZ). To produce very sharp crack tip (radius less than 0.05 mm, Fig. 11) the amperage of 1 A had been applied.

Once again the HAZ is the most critical region [33], so its J–R curve was plotted against CDFs to define instability (critical) point, as the tangent point of corresponding CDFs, Fig. 11. Considering the specimen with largest longitudinal crack along fusion line, 300 mm long and c = 13.5 mm deep, with the ratio c/h = 0.45 for thickness h = 30 mm, the critical pressure for unstable crack propagation reached 30.8 bar, Fig. 12, above prescribed proof test pressure, 25 bar, proving safety of pressure vessel and its structural integrity. Anyhow, once again it is easy to think about situation in which the proof test shifts the critical point into the unsafe region.

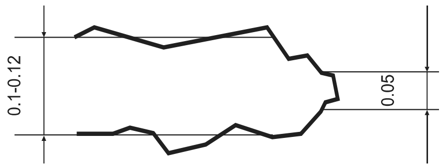


Figure 11. Crack tip obtained by electro-erosion [33].

Leakage of feedwater heater [34,35]

One of the frequent problems in ammonia converters is the leakage in the tube-to-tubesheet connection, causing failure of feedwater heater components. Such a problem occurred in the production of ammonia in the chemical industrial plant (HIP) in Pančevo [34,35]. In-service inspection revealed leakage in several tube-to-tubesheet joints of the feedwater heater, requiring not only repair, but also redesign. The original joint design consisted of tubes rolling into tubesheet and their mechanical expansion (Fig. 13), considering only the sealing, and neglecting thermal and mechanical loads. After a short service time, leakage occurred due to displacements of tubes in the tubesheet, caused by thermal and mechanical overload. To prevent the leakage, the tube-to-tubesheet joints were redesigned and the heater repaired.

The header and tubesheet are both made of carbon steel, and the tubes of an alloyed, Cr1/2Mo steel. Original design of tube-to-tubesheet connections required a pressed joint along 90% of sheet thickness (133 mm)

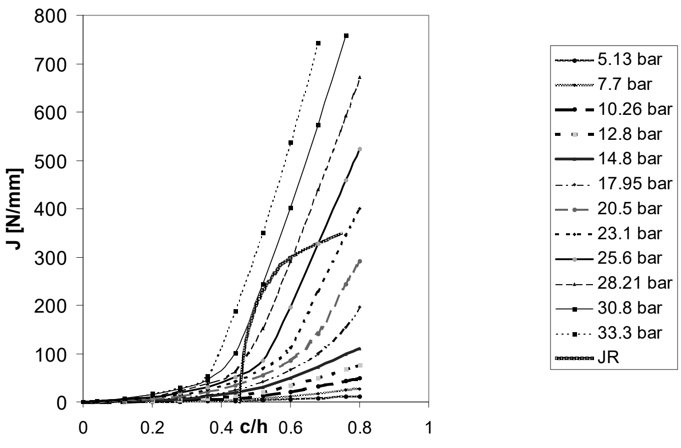


Figure 12. J–R curve positioned in CDFs [33].

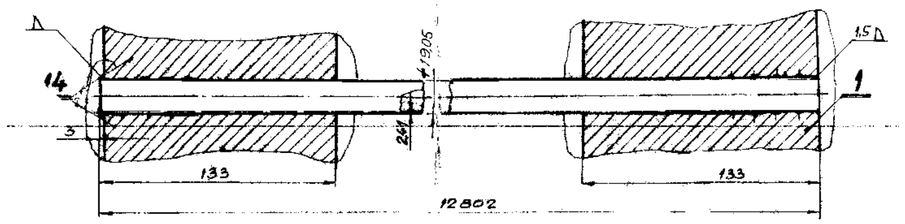


Figure 13. Original design of tube-to-tubesheet connection [35].

and provided sealing in the initial operating stage. Joints were performed by rolling tubes into the tubesheet, followed by mechanically expanding. New design introduced welding the tube-to-tubesheet connection at all tube ends in addition to the former design that required only rolling and mechanical expansion [35]. The accepted new tube-to-tubesheet joint design with additional welding is presented in Fig. 14. It is performed by applying additional surfacing with a soft layer, using Inconel consumable.

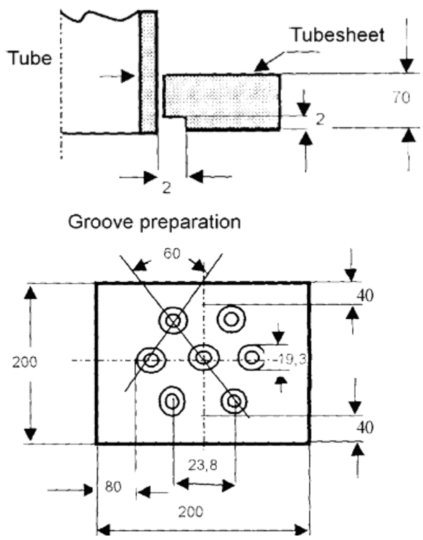
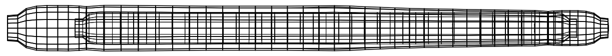


Figure 14. Design of plate preparation and dimensions of sample for welding procedure qualification with tube distribution [35].

Numerical models and results of analysis [35]

Three different models have been developed for numerical analysis. The water heater is modeled first in order to define the state before the repair. Main components of the heater (cylindrical mantle, tubesheet and reinforcements) are modeled as plates, as described in more details in [35]. In this model only several tubes are involved, in form of beams, defining behavior of all tubes [35]. Distribution of displacements due to inner pressure and thermal loading of the heater is shown in Fig. 15. The maximal displacement of the heater was 1.13 cm, enabling to conclude that tubesheet deformation allows for the heater repair.



f_{max} = 1.13 cm

Figure 15. Distribution of displacements on the heater model [35].

The mechanical and thermal displacements were calculated separately in the second model in order to consider their individual effects, since it was not possible to separate them in total displacements scheme. Stress intensities are given in Table 2.

The analysis of stress distribution revealed that maximum stress corresponds well to the hoop stress ($\sigma_t = pR/t$; p – pressure, R – inner radius, t – wall thickness). The upper side of tubesheet (in position of tube welding) is exposed to tension, that could be inconvenient for sealing between tube and tubesheet, making it the most probable cause of leakage. Both maximum stresses, in mantle (121.5 MPa) and in tubesheet (90.7 MPa) are well below yield strength. Anyhow, tensile stress in tubesheet is inconvenient for tubes sealing, contributing to the leakage.

The third model represents upper heater part, as generated for the analysis of local heating in welded

Table 2. Stress distribution in radial and hoop directions in the region of upper tubesheet, [35]

Stress components	Contribution to radial stress (MPa) from			Contribution to hoop stress (MPa) from		
Position	Pressure	Temperature	Total	Pressure	Temperature	Total
Mantle, remote of tubesheet, up	52±2	≈0	52±2	109±0	≈0	109±0
Mantle, close to tubesheet, up	52∓44	0±32	52∓12	52∓13	17±10	69∓3
Tubesheet – centre	23∓30	–10±56	13±26	23∓30	–10±56	13±26
Tubesheet – close to mantle	23∓2	3±57	26±55	23±19	–8.3±57	14.7±76
Mantle, close to tubesheet, down	78∓1	8∓2	86∓3	54∓3.5	0∓6	54∓9.5
Mantle, remote of tubesheet, down	54.3±8.4	≈0	54.3±8.4	119±2.5	≈0	119±2.5

region during the post-weld heat treatment. Model of 1299 nodes, defining 736 solid finite elements for substructures of mantle, tubesheet and tubes [35], as well as the 3D model of selected heater segment and its projections.

A program for nonstationary nonlinear heat conduction was used to obtain temperature fields during heat treatment. Heat source of 4 kW per meter of perimeter is simulated on the upper mantle part, based on experience [35]. Physical properties of steel are used, corresponding to expected heating temperature. Temperature field is calculated every 30 min up to 15 h, when 680 °C is achieved, as required (Fig. 16).

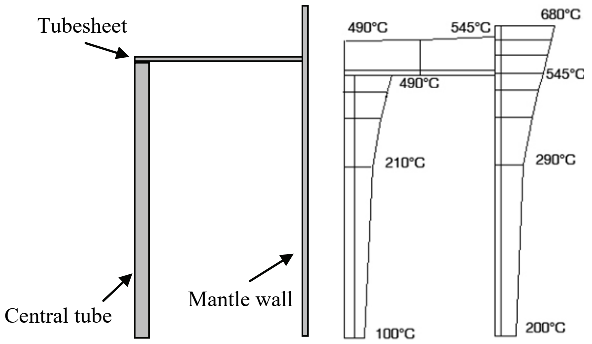


Figure 16. Temperature field simulating post-weld heat treatment [35].

The calculation was performed using the second model, but for a different thermal field. Displacements and stresses were calculated for given temperature field and their distribution is shown in Fig. 17. Results indicate low stress level, enabling to conclude that the heater is fit for service.

Therefore, the leakage problem was solved by additional welding using austenitic electrode at tube ends and tubesheet, requiring cutting the heater header and its welding after repair welding of tube ends and tubesheet. It has been proved that surfacing the tubesheet with austenitic electrodes was beneficial for its stress state, introducing compressive stress. Stress state during post-weld heat treatment (tubesheet and mantle) was acceptable, since the maximal stress in the redesigned heater was bellow Yield Strength.

CONCLUSIONS

Based on the results and discussion presented in this paper one can conclude the following:

- Fracture mechanics principles, applied for structural integrity assessment, provide better approach for safety, because can evaluate the significance of crack presence, predict crack growth and provide fitness-for-service, as very important engineering tool. It can be used even for non-existing cracks in the design phase of pressure equipment.
- Chemical engineers, in charge for pressure equipment, should learn how to live with cracks, rather than to assume that weldments are defect-free and provoke their growth, with unnecessary proof testing.

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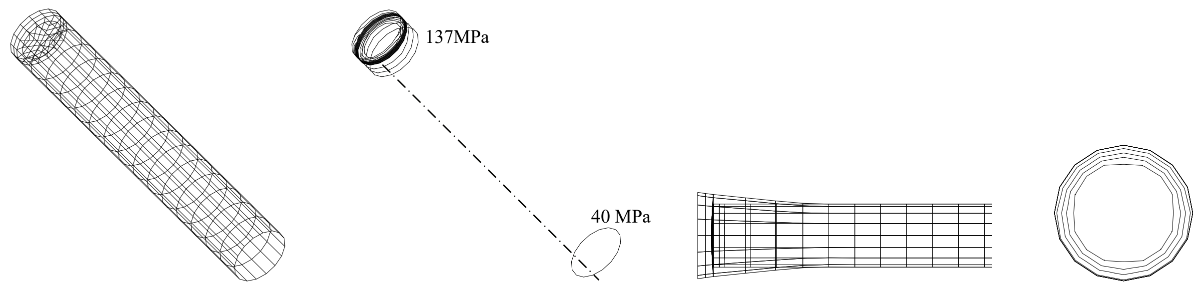

3D shell model of mantle with tube bundle | Stresses range between 40 and 137 MPa | Deformed state, f_{max} = 0.75 cm | Iso-stress lines at tubesheet periphery

Figure 17. Model, displacement and stress fields in feedwater heater after 15 hour heating for stress relaxation [35].

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IZVOD

INDUSTRIJSKA BEZBEDNOST POSUDA POD PRITISKOM – TAČKA GLEDIŠTA INTEGRITETA KONSTRUKCIJA

Aleksandar Sedmak[1], Mahdi Algool[2], Snezana Kirin[3], Branislav Rakičević[1], Ramo Bakić[4]

[1]*Mašinski fakultet Univerziteta u Beogradu, Beograd, Srbija*
[2]*Univerzitet u Sirtu, Sirt, Libija*
[3]*Inovacioni centar Mašinskog Fakulteta, Beograd, Srbija*
[4]*Tehnička škola,Tutin, Srbija*

(Pregledni rad)

Ovaj rad prikazuje različite aspekte problema sigurnost posuda pod pritiskom, sa fokusom na hemijsku industriju. Obezbeđenje kvaliteta, uključujući primenu PED97/23, je prvo analizirano, zatim je dat kratak osvrt na procenu rizika, a detaljnije prikazan pristup zasnovan na proceni integriteta konstrukcije, koji je ilustrovan opisom tri primera iz prakse. Prvi primer se odnosi na procurivanje zavarenih spojeva cilindričnog rezervoara za CO_2, kod koga su priključci napravljeni od austenitnog čelika, dok je omotač od feritnog, niskolegiranog čelika povišene čvrstoće. Drugi primer se odnosi na velike sferne rezervoare za amonijak, napravljene takođe od niskolegiranog čelika povišene čvrstoće, za koje je praksa pokazala da su osetljivi na nastanak i rast prslina, što dolazi do izražaja posebno pri reparaturnom zavarivanju i hladnom probnom pritisku. U trećem primeru je analiziran problem procurivanja na spoju cevi i cevne ploče na ulazu u zagrejač vode u HIP Azotara, što je zahtevalo rekonstrukciju i popravku dodatnim zavarivanjem cevi. Postupak popravke je predvideo isecanje glave zagrejača po zavarenom spoju, pripremu i zavarivanje cevi za cevnu ploču, i ponovno zavarivanje glave. Kao važan zaključak koji sledi iz ovih primera, navodi se kontraverza u vezi hladne probe pritiskom, koja može da napraviti više štete nego koristi, posebno kod čelika osetljivih na prsline.

Ključne reči: Bezbednost posuda pod pritiskom • Integritet konstrukcija • Probni vodeni pritisak • Procena rizika