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MISKAR COMPRESSION MODULE SEAWATER COOLING PIPEWORK IMPROVEMENT

POBOLJŠANJE KOMPRESIONOG MODULA RASHLADNOG CEVOVODNOG SISTEMA MORSKE VODE MISKAR

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Keywords
- crack
- vibration
- fatigue
- failure
- numerical solution
- Whöler curve

Ključne reči
- prslina
- vibracije
- zamor
- lom
- numerička rešenja
- Velerova kriva

Abstract

This paper consists on studying root causes of seawater pipeline network failures. The pipe network is installed on the Miskar offshore of the British Gas Tunisia. It provides cold seawater for heat exchangers of gas pipework. The Engineering department cannot ignore these failures since the closure of this network will cause production shutdown. One of identified root causes is the flow induced vibration that provokes excessive vibrations. A numerical simulation is done to ascertain flow turbulence in critical areas. This simulation, based on the CAESAR II pipe stress analysis, proves that maximum of turbulence intensity and velocity are located at welded areas and at points with excessive vibrations. It is noted that the higher the diameter the more likely failure to occur. By calculating stress components in critical areas and points, as tees and elbows, one can estimate the theoretical lifetime of crack appearance in structural supports. Three methods are applied here to estimate the fatigue life of the elbow support: ANSYS Inc software, ASME tables and Wöhler curves for stainless steel material. This lifetime is compared with theoretical lifetime expected by the office who had studied and constructed this compression module.

Izvod

Ovaj rad predstavlja studiju uzročnika otkaza cevovodne mreže morske vode. Cevovodna mreža je instalirana na morskoj platformi Miskar u svojstvu British Gas Tunisia. Ona obezbeđuje hladnu morsku vodu za toplotne izmenjivače cevovodne mreže. Ovi otkazi se ne mogu ignorisati, jer se zatvaranjem ove mreže gasi proizvodnja. Jedan od identifikovanih uzročnika jeste tok indukovanih vibracija koje prouzrokuju dodatne vibracije. Izvedena je numerička simulacija za sračunavanje turbulentnih strujanja na kritičnim mestima. Simulacija zasnovana na CAESAR II analizi napona u cevovodu dokazuje da se najveći intenzitet turbulencija i brzina nalazi na mestima zavarenih spojeva i u tačkama sa najvećim vibracijama. Primećuje se da, što je veći prečnik, to je veća mogućnost za pojavu otkaza. Izračunavanjem komponenata napona na kritičnim mestima i tačkama, kao što su račve i kolena, može se proceniti teorijski radni vek do pojave prsline na osloncima konstrukcije. Ovde su primenjene tri metode za procenu zamornog veka oslonca kolena: ANSYS Inc software, ASME tabele i Velerove krive za nerđajući čelik. Ovaj radni vek je upoređen sa teorijskim vekom, očekivanim od strane izvođača, koji je proučio i konstruisao ovaj kompresioni modul.

INTRODUCTION

Failure in pipe network due to fatigue is one of the most important problems encountered in the oil and gas field. This failure is caused generally by alternating stress which generates the largest failure issue which is vibration. In order to control this type of failure, oil and gas companies give much more importance to this problem than to others, because it has a great influence in the commanding process. British Gas Tunisia is one of the largest companies producing natural gas in Tunisia and in the world. It supplies more than 60% of Tunisia's domestic gas production through Miskar and Hasdrubal offshores. In 2005, the company has added a compression module to the Miskar offshore in order to increase gas pressure that will be sent in a pipeline from Miskar offshore to the "Hannibal" plant /1/. This module contains heat exchangers and strainers. A seawater pipeline network is installed to provide cold seawater to use it for heat exchangers of gas. For this, British Gas Company tries to evaluate every symptom of cracks or rupture and identify issues causing this problem.

In 2008, failures and leaks have been detected in this network. These failures are caused generally by alternating stress which generates vibrations.

In this work, fatigue problems and vibration issues are carried out in detail. An investigation visit to the Miskar offshore of the BG Company is done to evaluate the present situation and identify vibration, corrosion, leaks and cracking root causes. After root causes identification, a fatigue assessment is done to estimate fatigue life of an elbow support before cracking, using “ANSYS” software and “ASME B& PV division 2” code. Finally, some solutions and recommendations are revealed to decrease the failure level.

SEAWATER PIPE NETWORK

Miskar contains three attached offshores (A, B and C). All production process is located in A offshore, however C contains the torch and B contains the accommodation and control rooms (Refer to Fig. 1). Gas from the Miskar field is processed at the BG Group-operated Hannibal plant and sold into the Tunisian gas system. Miskar A offshore contains production equipment as heat exchangers, compressors, pumps and certainly gas wells. It also contains three decks; cellar deck, main deck and a sub-cellar deck that contains the seawater pipework with pumps.

The piping & instrumentation diagram (P & ID) describing this pipework is shown in Fig. 2 with line numbers and three seawater lift pumps.

Figure 1. Miskar offshore.
Slika 1. Miskar platforma

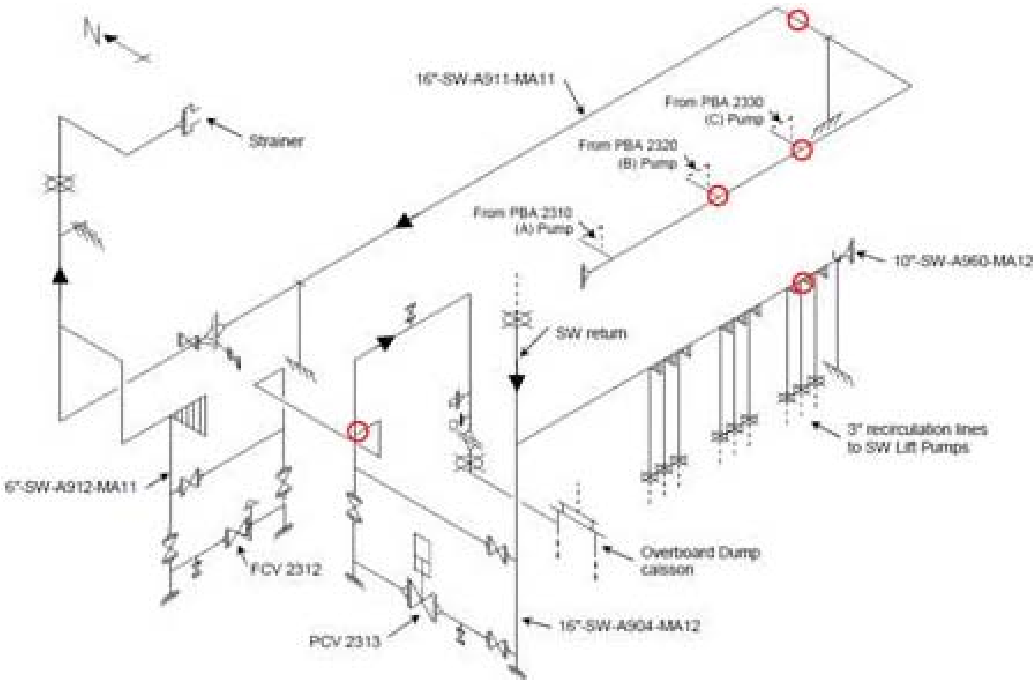

Figure 2. Schematic layout of seawater pipework in sub-cellar deck
Slika 2. Shema cevovodne mreže u podćelijskom nivou

Lift pumps (PBA 2310-2320-2330) bring seawater at a rate 2010 m^3/h from the sea through line number 16”-SW-A911-MA11 under a pressure of 3 bar until arriving to the strainer. From the strainer, the seawater line is divided into many lines to supply heat exchangers, compressors, and the turbine.

After passing through this equipment, the seawater returns to the sub-cellar deck through line number 16”-SW-A904-MA12 (bypass) and then directly to the overboard. The pumps operate at a speed of 1480 rpm (24.7 Hz). For the seawater pipework, the piping material is stainless steel (SS) A312-S31254 (254 SMO). In this plant it is used to support a temperature of 100°C and 20 bar pressure. It is austenitic stainless steel and is designed for maximal resistance to pitting and crevice corrosion. 254 SMO is substantially stronger than the common austenitic grades. It is also characterized by high ductility and impact strength. 254 SMO is readily fabricated and welded. For all pipework, the allowable corrosion is 1.5 mm.

INVESTIGATION VISITS

Visits have been done to Miskar offshore platform in order to investigate pipework failures. Many of them have been identified such as corrosion, vibration, and leaks. In the control room during the investigation, abnormal phenomena are identified such as alternating flow rate and pressure in the pump discharge. This information is plotted in trends so to see the history of this phenomenon. Figure 3 shows curves (from top to bottom) concerning alternating pump speed, discharge pressure, overboard return flow and condensate pump flow control. There is a dangerous alternating flow rate in the seawater pipework. This alternating flow rate comes certainly from alternating rotation speed of the pump regulated by a flow control valve. Some seawater pipework locations, where vibration can exceed allowable levels and cause fatigue damages to pipes and supports, are repaired (Fig. 4).

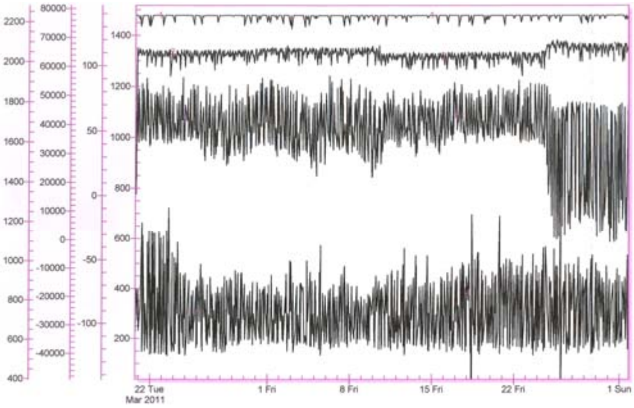

Figure 3. Pump speed, discharge pressure, overboard return flow and condensate pump flow.
Slika 3. Brzina pumpe, pritisak pražnjenja, povratni tok i protok kondenzatne pumpe

Figure 4. Support failure.
Slika 4. Otkaz oslonca

PREDICTION OF VIBRATION MODES

A detailed pipework model is created using CAESAR II pipe stress analysis. This includes 16”, 10” and 6” pipework and suitable elements used in the pipe stress analysis (PSA) to model the seawater pipework for the purpose of improved conditions. Using CAESAR II, incorporating a wide range of capabilities, the system presented in Fig. 5 is studied.

The pipework and components are of stainless steel Class 150 rated. Table 1 gives details on these components, the pipework and bends. The mechanical properties for 16”, 10” and 6” diameter pipework in the model are those conforming to ASME B&PVII Part D, /2/, code UNS31254 material, i.e. the pipe material (ASTM–A312) has a specified minimum yield strength of 303.4 MPa and an ultimate tensile strength of 648.1 MPa. The allowable stress for this material according to specification of ASME B31.3, /3/, is 157.1 MPa.

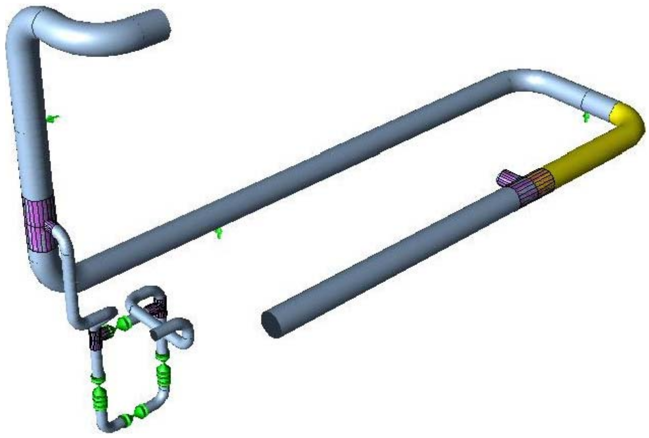

Figure 5. Layout of seawater cooling pipework.
Slika 5. Sklop cevovodne mreže rashladne morske vode

Table 1. Pipework and bends details.
Tabela 1. Elementi cevovodne mreže i kolena

Nominal size	Outside diameter (mm)	Wall thickness (mm)	Bend 90° outside diam. (mm)	Centre to end dimension (mm)
6”	168.3	3.4	168.3	229
10”	273	4.19	273	426
16”	406.4	6.35	406.4	610

Valves and flanges, insulation joints, and filters are modelled as rigid connections. Additional information regarding the weights of these components is taken from the CAESAR II database of components.

Analysis of the static loading conditions is required when non-linear restraints are present in order to determine whether the restraints are classed as active or inactive. Analysis of the static cases is also required to ensure that any proposed new designs are still code compliant.

The following standard static load cases are created and analysed to ASME B31.3:

L1 (OPE) W+T1+P1 = Max Design Case (Design pressure 10.3/3.5 bar and design temperature 150°C)

L2 (OPE) W+T2+P1 = Operating Case (Operating pressure and temperature)

L3 (OPE) W+T3+P1 = Min Design Case

L4 (SUS) W+P1 = Sustained Case for ASME B31.3 static code stress check

L5 (EXP) L1-L3 = Expansion Case for ASME B31.3 static code stress check

L6 (EXP) L2-L3 = Expansion Case for ASME B31.3 static code stress check

where,

WW = Pipe & water weight (for hydrostatic test)
W = Pipe & fluid weight
D1 = Condenser displacement

T1 = Maximal temperature thermal gradient with reference to ambient temperature

T2 = Minimal temperature thermal gradient with reference to ambient temperature

P1 = Internal design pressure

In this model the temperature and pressure are fixed in all nodes (T = 20°C and P = 3 bar)

The results of the load cases are presented in Table 2, which shows high code stress for three different load cases:

Table 2. Static results.
Tabela 2. Rezultati statike

High stresses (kPa)	(OPE) W+T1+P1	(SUS) W+P1	(EXP) L2–L3
Code stress ratio	0.0%@N885	12%@N860	53%@N885
Code stress	184980.4	16956.9 A:137892	180448 A:340605
Axial Stress	16938.3@N865	4738.8@N48	13343@N865
Bending Stress	180373.9@N885	13488.2@N860	179943@N885
Torsion Stress	6745.2@N885	1408.9@N55	6745.2@N885
Hoop Stress	9290.5@Node10	9366.6@N10	0.0@N10

A: Allowable stress, (max stress)@(node number)

This table gives us an idea for maximum stress and the maximum stress ratio for specific node locations and even for different load cases.

Through this analysis, we can deduce that stresses due to sustained loads are all allowable.

CAESAR II is a great way to analyse displacements of a pipework, as Fig. 6 shows; we can observe nodes or parts of the pipework that are more exposed to displacement.

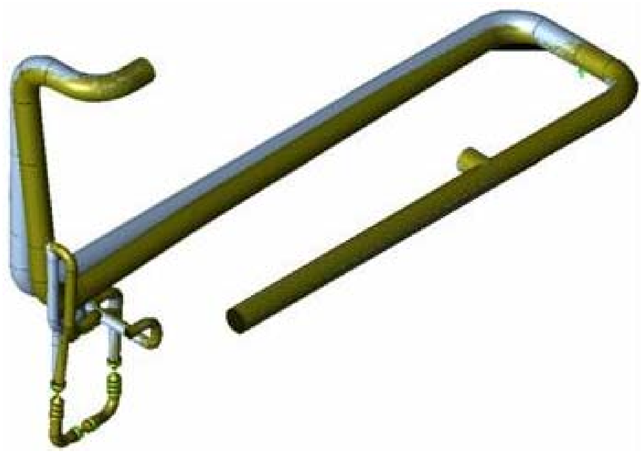

Figure 6. Displacement growth, 3D view.
Slika 6. Razvoj pomeranja, 3D prikaz

The highest code stress ratio (the predicted code stress over the allowable stress) is predicted to be 12% of the allowable at node 860 which represents the 6” elbow as shown in Fig. 7. At the same time this node represents the maximum displacement and force applied (4786 N).

The highest code stress ratio is predicted at 53% of the allowable stress in node 885 which represents the second 6” elbow. This elbow is anchored by a support that will be studied for fatigue life case.

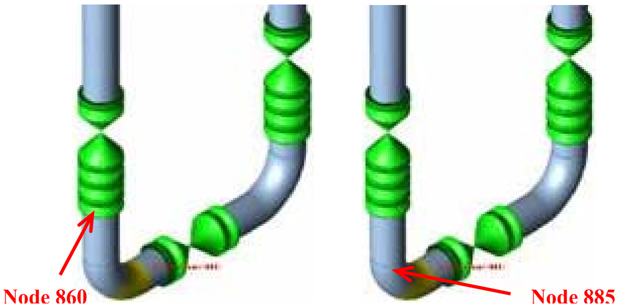

Figure 7. Maximum load force in node 860 and stress in node 885.
Slika 7. Najveća sila u čvoru 860 i napon u čvoru 885

FAILURES ROOT CAUSES IDENTIFICATION

The likelihood of failure (LOF) is a form of scoring to be used for scoring purpose, /4/. The likelihood of failure is not an absolute probability of simplified models to ensure ease of application and conservative models are necessary. A quantitative main line LOF assessment is undertaken because of the lack of technical material.

This result is an LOF score for each main line in the system and for each identified excitation mechanism. The LOF scores for some excitation mechanisms are pipe diameter and wall thickness dependent (e.g. flow induced turbulence).

The likelihood of failure for flow induced turbulence is then determined by the following equation:

$$\text{Flow induced turbulence LOF} = \left(\frac{\rho v^2}{F_v}\right) FVF$$

This quantitative assessment will include three diameters of flow lines 6”, 10” and 16”. Table 3 shows the results for three internal and external pipe diameters.

Table 3. Likelihood of failure for different diameters.
Tabela 3. Izgledi za otkaz za različite prečnike

Nominal diameter	6”	10”	16”
Internal diameter (mm)	164.9	273.81	400.05
External diameter (mm)	168.3	273	406.4
V (m/s)	25.4	9.66	4.5
ρv^2 (N/m^2)	656127.72	95090.56	20594.25
FVF	1	1	1
$F_v = \alpha(D_{ext}/t)^\beta$ (N/m^2)	19283.54	20955.2	28470.2343
LOF	0.03	0.22	0.723

One can note that the higher the diameter the more is the likelihood of failure become higher, and this explains the failures of 16” more than in the other pipework diameters.

SUPPORT FATIGUE ANALYSIS

One major factor responsible for the support failure presented in Fig. 4 are the excessive vibrations. As it has been mentioned, they are obtained by alternating flow rate which causes alternating forces and stresses. These alternations conduct to a stress range that will cause a fatigue problem for this support /5, 6/. Three methods are presented to calculate the support fatigue life.

1. Numerical method using ANSYS Inc software

The boundary conditions are faces from where the support will be fixed and face where the calculated force will be applied. The support will be exposed to the elbow force which varies with flow alternations. The boundary conditions are presented in Fig. 8.

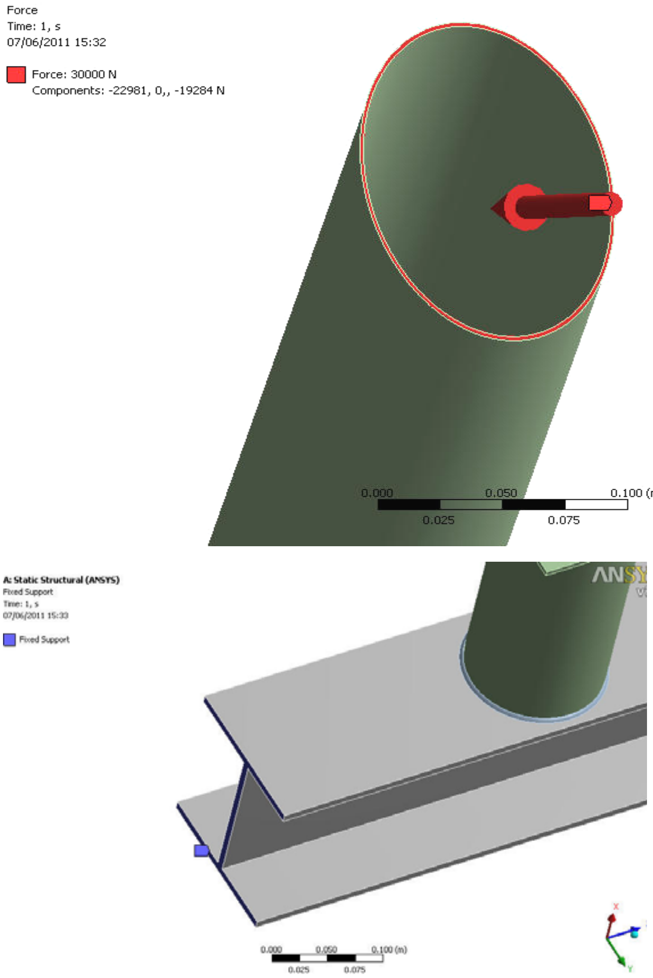

Figure 8. Boundary conditions (elbow force K = 31 kN and support fixed end).
Slika 8. Granični uslovi (sila u kolenu K = 31 kN i nepokretni oslonac)

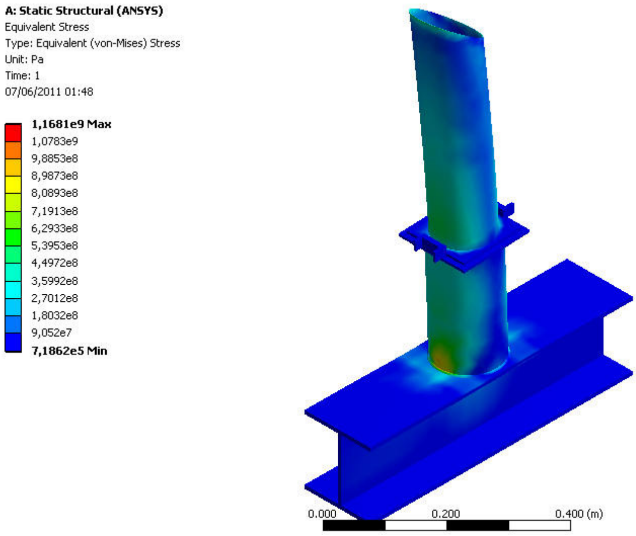

Figure 9. Equivalent (Von-Mises) stress distribution.
Slika 9. Raspodela Mizes ekvivalentnih napona

Figures 9 and 10 show clearly the maximal and minimal stress and strain values provided by ANSYS software, /7/. It can be noted that the maximal stress value ($1.168\cdot10^9$ Pa) is situated in the area between vertical and horizontal support, meaning very close to the weld zone. We can not ignore the minimal stress value ($7.18\cdot10^5$ Pa) which is very important.

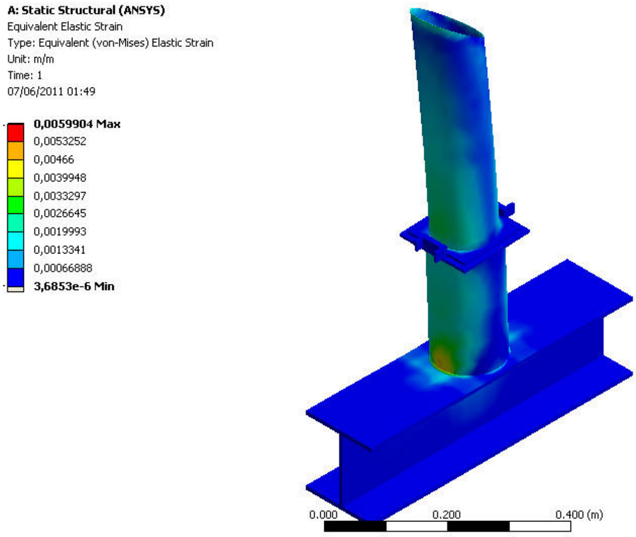

Figure 10. Equivalent (Von-Mises) stress distribution.
Slika 10. Raspodela Mizes ekvivalentnih napona

As it is shown, the maximal strain values vary between 0 and 0.006. Maximal strain is situated in the area which represents the intersection between elbow and support.

ANSYS software can perform fatigue calculations for either constant amplitude loading or proportional non-constant amplitude loading. For our case, we shall apply constant amplitude loading. 0.2 is the stress ratio that will be applied for alternating loading.

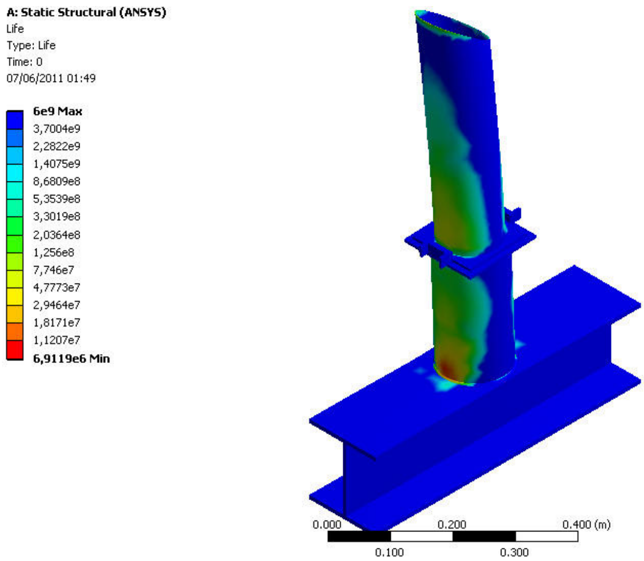

Figure 11. Fatigue life distribution.
Slika 11. Raspodela zamornog veka

Fatigue analysis results are shown in Fig. 11. This last one demonstrates fatigue life for every stress variation in the model. The critical point or area is located between parts intersection which means next to the welding zone. This area contains the maximal equivalent stress ratio, so it will be evident that it presents the first area to be affected by failure (breaking-cracking). $6.9\cdot10^6$ cycles is the minimal fatigue life value equal to 7 years (10^6 cycles = 1 year).

2. ASME code; ASME B & P VII Part D

Smooth bar design fatigue curves are provided for materials in terms of a polynomial function and the design cycles, N, can be computed from the equation, /8/:

$$N = 10^x$$

or from Table 4, based on the stress amplitude S_a. Where,

$$x = \frac{C_1 + C_3Y^2 + C_5Y^3 + C_7Y^4 + C_9Y^5 + C_{11}Y^6}{1 + C_2Y + C_4Y^2 + C_6Y^3 + C_8Y^4 + C_{10}Y^5},$$

$Y = (S_a/6.9)\cdot(E_{FC}/E_T)$, and $C_1 \rightarrow C_{11}$ are equation constants.

E_{FC} – modulus of elasticity used to establish the design fatigue curve; E_T – modulus of elasticity of the material under evaluation at the average temperature of the cycle being evaluated.

By referring to Fig. 3, we can determine the maximal and minimal stress values using matching maximal and minimal flow rate and force values. By calculating these we obtain a stress amplitude of 126 MPa, which means that we shall use the table containing $S_a \leq 195$ MPa.

Table 4. Data for fatigue.
Tabela 4. Podaci za zamor

2010 SECTION VIII, DIVISION 2
Table 3.F.10– Data for Fatigue Curves in Table F.1 Through F.9

Number of Cycles	Fatigue Curve Table (in ksi)					
	3.F.1	3.F.2	3.F.3	3.F.4 Curve A	3.F.4 **Curve B**	3.F.4 Curve C
1E1	580	420	708	–	–	–
2E1	410	320	512	–	–	–
5E1	275	230	345	–	–	–
1E2	205	175	261	–	–	–
2E2	155	135	201	–	–	–
5E2	105	100	148	–	–	–
8.5E2 (1)	–	–	–	–	–	–
1E3	83	78	119	–	–	–
2E3	64	62	97	–	–	–
5E3	48	49	76	–	–	–
1E4	38	38	64	–	–	–
1.2E4 (1)	–	43	–	–	–	–
2E4	31	36	55.5	–	–	–
5E4	23	29	46.3	–	–	–
1E5	20	26	40.8	–	–	–
2E5	16.5	24	35.9	–	–	–
5E5	13.5	22	31.0	–	–	–
1E6	12.5	20	28.3	28.2	28.2	28.2
2E6	–	–	–	26.9	22.8	22.8
5E6	–	–	–	25.7	19.8	18.4
1E7	11.1	17.8	–	25.1	**18.5**	16.4
2E7	–	–	–	24.7	17.7	15.2
5E7	–	–	–	24.3	17.2	14.3
1E8	9.9	15.9	–	24.1	17.0	14.1
1E9	8.8	14.2	–	23.9	16.8	13.9
1E10	7.9	12.6	–	23.8	16.6	13.7
1E11	7.0	11.2	–	23.7	16.5	13.6

As we can see 18.5 ksi is the nearest value to 18.34 ksi. 18.5 ksi corresponds to $N = 10^7$ cycles (see Table 4) which is equivalent to 10 years before failure and cracking.

3. Wöhler curve method

Figure 12 presents the Wöhler curve of an austenitic stainless steel for temperatures not exceeding 371°C and for physical properties identical to those of A312 material, /9/.

Using 126.8 MPa of stress amplitude value, the ASME code curve shows a cycles number of $2\cdot10^6$. It presents an abnormal number because the support has been changed before 3 years, and because of obtaining of this value is that conditions of experience are certainly different from experience done by ANL. This last one has made a curve that gives as a result of $9.5\cdot10^6$ which is near to ANSYS fatigue analysis and ASME B & P VII method.

CONCLUSION

Three methods are applied in this paper to estimate the fatigue life of the elbow support of 6”-SW-A912-MA11 line number. The result changes from one method to another but is still too close even if we have obtained in some cases abnormal results. Seven to ten years is the estimated fatigue life of the support before failure or cracking. This result can change because of the addition of many other conditions as fatigue corrosion and stress corrosion crack growth, or increase/decrease of water flow rate which causes force variation. It is absolutely necessary to say that this analysis has been done in the case of alternating stress and the line number pipework line did not stop working, but in the real case many shutdowns may occur or the line work can be deactivated.

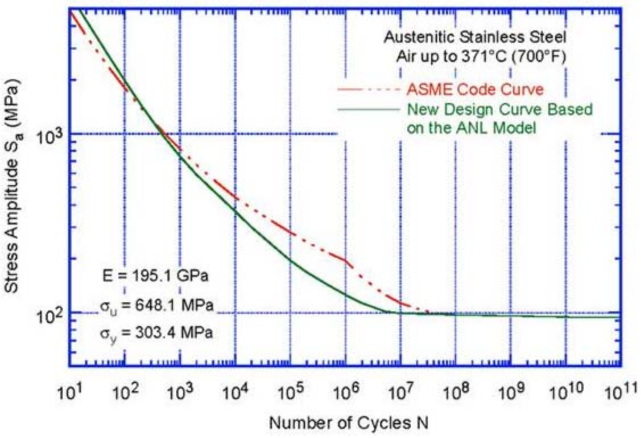

Figure 12. S-N curve for austenitic stainless steel not exceeding 371°C.
Slika 12. S-N kriva za austenitni nerđajući čelik, kod kojeg temperature ne prelazi 371°C

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