FINITE ELEMENT METHOD ANALYSIS OF PIPE MATERIAL TEMPERATURE CHANGES INFLUENCE ON LINE EXPANSION LOOPS IN HYDRAULIC INSTALLATIONS ON MODERN TANKERS

by

Andrzej BANASZEK [a*], ***Radovan PETROVIĆ*** [b], ***and Bartlomiej ZYLINSKI*** [c]

[a] Faculty of Maritime Technology, West Pomeranian University of Technology, Szczecin, Poland
[b] Faculty of Mechanical Engineering Kraljevo, University of Kragujevac, Serbia
[c] Strata Marine & Offshore AS, Oslo, Norway

Original scientific paper
UDC: 673.6:620.16:66.011
DOI: 10.2298/TSCI1101081B

Finite element method analysis of main lines of hydraulic central loading system installation expansion loops mounted on product and chemical tankers has been presented in the paper. The axial forces problem in installations mounted along the ship's open decks executed from hull deformations on waves and thermal stresses is given. Use of "U" type expansion loops is described. Results of forces in anchor points and stresses of Mises due to expansion loop deformations are shown. Calculations were made by ABAQUS Ver.6.7 FEM computer program.

Key words: *expansion loops, hydraulic installation, anchor points, Misses stresses, finite element method analysis, product and chemical tanker*

Introduction

Product and chemical tankers are ships intended to the transport of dangerous, petroleum-based liquid cargoes and chemicals (see fig. 1). Therefore many types of pipes are installed on decks of the product and chemical tankers as ships intended to the sea carriage of liquid cargoes. One of the most important installations are hydraulic installations of central supplying systems, intended to the drive hydraulic submerged cargo pumps [1-5]. Working conditions under which these hydraulic installations are exploited at sea are extremely difficult. This is the result of specificity of prevalent circumstances on board of sea-going vessels. One of the most important problems is influence of changes of the ambient temperature, changes of the degree of solar radiation, and temperatures of transfluent hydraulic oil on stresses and tensions in pipes of hydraulic installations.

It takes root that the change of the length of the line is proportional to the change of the temperature of line material [6]:

$$l \quad l_0(1 \quad \alpha_L \Delta T) \tag{1}$$

* Corresponding author; e-mail: a.banaszek@ps.pl

where l is the length of the line after the temperature rise of pipe material, l_0 – start length of the line, ΔT – temperature rise of pipe material, and α_L – coefficient of linear expansion of pipe material.

In tab. 1 are presented values of the linear expansion coefficient for materials most often used under construction of the hydraulic drives installation on ships.

Equation (1) can be presented in folowing form, showing the elongation value of pipeline after the temperature rise of pipe material:

$$\mathrm{d}x \quad l \quad l_0 \quad l_0\alpha_L\Delta T \tag{2}$$

Table 1. Values of linear expansion coefficients for chosen pipe materials

Name of material	The coefficient of linear expansion α_L, [K^{-1}]
Carbon steel St37.0	16.0 10^{-6}
Stainless steel AISI 304	16.0 10^{-6}
Stainless steel AISI 316	17.3 10^{-6}
Copper	16.6 10^{-6}
Iron	11.0 10^{-6}
Inwar	1.0 10^{-6}

Source: The Vademecum of hydraulics,Vol. 3, Mannesmann Rexroth [7]

In case of fixed two ends of pipeline and lack of expansion loops (compensators) in installaton structure above elongation can leads, according to Hook's to followng axial type stresses in pipeline:

$$\mathrm{d}x \quad \frac{\sigma l_0}{E} \quad l_0\alpha_L\Delta T \tag{3}$$

$$\sigma \quad E\frac{\mathrm{d}x}{l_0} \quad E\alpha_L\Delta T \tag{4}$$

where σ is the tensile stress in pipeline, dx – the elongation of

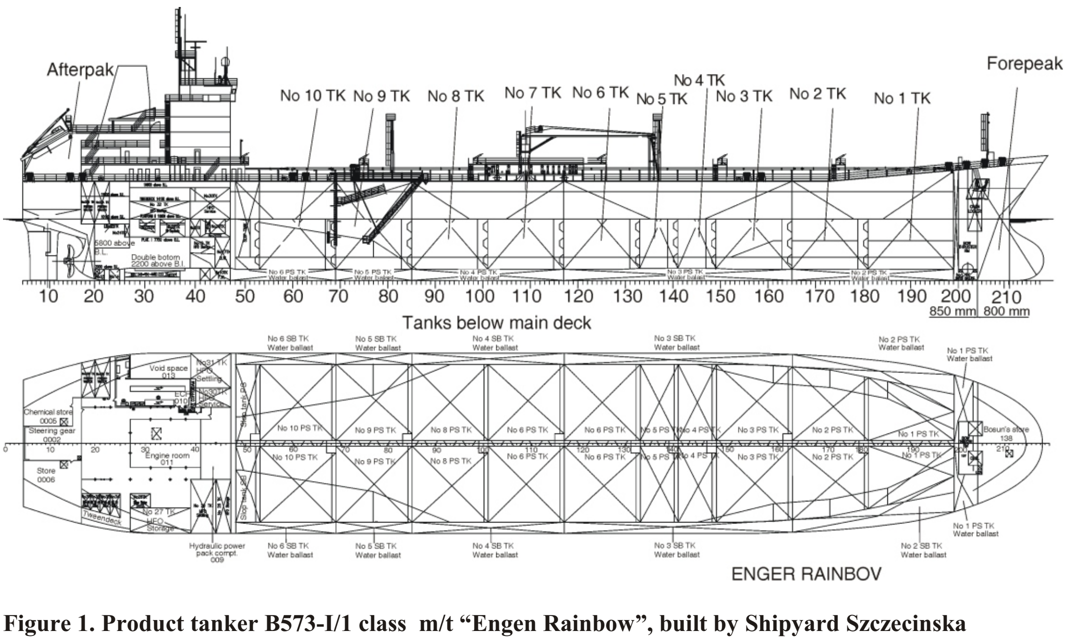

Figure 1. Product tanker B573-I/1 class m/t "Engen Rainbow", built by Shipyard Szczecinska S. A., Poland, for the ship-owner Unicorn Tankers from Republic of South Africa

pipeline determined by temperature rise of pipe material, and E – Young's modulus of pipe material.

Normal length of main lines of hydraulic central loading systems, mounted on board of modern product and chemical tankers can reach 150 m. System can be exploited from temperature (in winter) 253 K, and temprerature of hydraulic oil can reach 343 K [8].

Therefore, this situation can created, according to eq. (4), dangerous high, axial stress in hydraulic pipes. One of the ways to avoid described problems is the assembly of line expansion compensations of the "U" type. The lack of above mentioned compensations in case of appearing of axial tensions can lead to bursting and lack of hydraulic lines tightness. Accidents with described problems were noticed on board of many ships, especially tankers, to example on product tankers B560 type m/t "Boris" and "Anatoli" (Liberia), during sea trials. Compensators of "U" type in hydraulics are most often used. It is result of experiences at sea, during exploitations on ships, because they are the most safe. Other type compensators, with flexible elements (*i. e.* hoses), membranous, follicular, very often damaged in hard sea atmosphere or are not enought strength on high pressure oil inside (300 bar). Therefore compensators "U" executed directly from own pipe material are so popular in execution of high pressure hydraulics in shipbuilding. Expansion loops, with other shape, as to example "V" type, are not exploited in practice. It is result of problems with hydraulic pipe bending. In hydraulics are strictly allowed only "cold" type bending procedure. In case of pipe bending with degree bigger than 90 degree, at relative big pipe thickness refering to pipe diameter (to example size of pipe 130 13), create problems with reduction of material thickness, breaking or folding in pape, in bending area. Therefore "V" type expansion loops, or equivalent, are not used in practice on ships.

The above events are unprofitable for the shipowner because they can cause damages of hydraulic drive systems and environmental pollution by leaking of hydraulic oil. This can result in heavy financial penalties payments, especially when the event takes place directly in the port. Described problem is important not only from practice, but from scientistic side too. In scientist literature is lack of publications referring to calculations of expansion loops in high pressure hydraulic systems, especially mounted on ships. Some papers described finite element method (FEM) analyse of pipes mounted in low pressure installations, especially in heat systems (Koorey [9], Fedorov, [10], Jo., *et al.* [11], *etc.*). Mahmood, *et al.* [12] are shown study of performance limitation in micro heat pipes with diammeter to only 3 mm. Next authors concentrated on analysis of thermal efects in simple straight pipes (*i. e.* Rajeev *et al.* [13], Madenci, *et al.* [14], Deng Zhi-Wei, [1-6, 9]). Important analysis of fastening of pipes was presented by Vakharia, *et al.* [15]. Lack of same publications in area of high pressure oil hydraulic systems, required own analyse of problems, especially referring to forces in anchor points and stresses created during activity of expansion loops as result of pipeline squeezing deformations.

As result of these investigations, the FEM analysis of expansion loops "U" type, mounted on hydraulic installations of central loading systems on modern product and chemical tankers, has been presented in the paper. Results of above mentioned calculations were used in Shipyard Szczecinska S. A. in practice.

Idea of line expansion loops "U" type

As already mentioned in the introduction, in order to avoid bursting of pipe installations mounted along the ship's open deck it is necessary to install suitable quantity of line expansion compensators. In general, compensators of follicular or pipe-sliding types are used in low-pressure ship's installations. Because of the limited range of pressures they can not be used in hydraulic systems. Therefore the compensators used in main lines of hydraulic central loading systems or other hydraulic installations mounted along ship's open decks are pipe expansion loops "U" type. The idea of above mentioned compensators is presented in fig. 2.

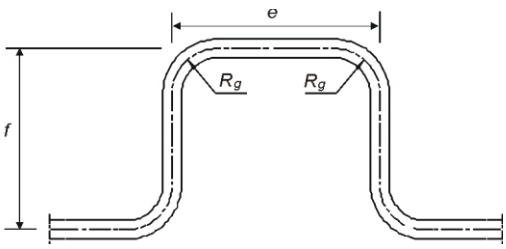

Figure 2. Geometry of line expansion loops "U" type [8, 16]

The compensators are properly bent pipe in the shape of the "U" alphabet-letter. If postaxial forces appear in the installation, this kind of compensator is properly deformed. Its shape makes easy this type of deformation and limits the height of tensions in the remaining part of piping installation. For the high pressure main line with size of 130 13 mm from the hydraulic central loading system, standard dimensions of the expansion loops are given in tab. 2.

Table 2. Dimensions of standard expansion loop "U" type for high pressure hydraulic pipe *P* 130 13 mm

Nominal diameter	Pipe size d_z g [mm]	Material	Max.working presure [MPa]	Bending radius, R_g [mm]	Dimension, *e* [mm]	Dimension, *f* [mm]
110	130.0 13.0	St52.4	32.0	325	1500	1400

Source: [8, 16]

Presented (in tab. 2) line expansion loops, made of carbon steel St52.4 (see tab. 3) were installed on board of product tanker B578-I/1 m/t "Helix" built by Szczecinska S. A. Shipyard for the Shell ship-owner from Australia. Main pressure line *P* 130 13 mm was a main supply hydraulic line for the team of cargo pumps equipped into hydraulic drive [1, 2, 4]. The overall length of the ship was 183 m and above mentioned hydraulic line was installed along the whole open deck, from the superstructure to the bow.

Table 3. Mechanical properties of carbon steel St52.4/DIN 1630

Steel symbol	Material Nr/DIN standard	Stretch endurance (min.) [MPa] R_m	Yield strength (min.) [MPa] $R_{p0.2}$	Elongation at breaking (min.) [%] A_5	Endurance coefficient *K* [MPa] acc, instr. AD W4 at temperature 20 °C
Carbon steel St52.4	1.0581 1630	490	350	21	355

Source: Vademecum of hydraulics, Vol. 3, Mannesmann Rexroth [7]

For the assessment of correctness of hydraulic installation design and execution on board are used following criteria:

(a) Line strength criterion – whether the tightness of the installation is kept at the test pressure,

(b) Correctness of pipe holders criterion – whether lines under external mechanical loads (from hull vibrations, knocks of the wind, *etc*.) and internal mechanical loads (from turbulence of transfluent medium – hydraulic oil) pipes do not fall in excessive mechanical vibrations, especially the resonance type, and
(c) Criterion of the resistance of lines on longitudinal extensions – whether sufficient quantity of expansion loops are installed and whether they are correctly situated so that in the case of line extensions appearing in the installation:
  – forces generated in anchor points of the installation will not exceed limited values (given by the maker), and
  – stresses in the pipe installation and expansion loops will not exceed limited values.

Pipe installation strength and correctness of pipe holders are checked by inspectors of international classificatory societies in the shipyard according to Societies Rules.

The greatest difficulty is design of line expansion loops system in order to fulfill the last criterion. Generally speaking, there are not enough procedures and regulations in the design process, especially the ones related to shipping-hydraulic installations.

Numerical analysis, results, and discussion

The piping installation mounted on board of ship's deck must be fastened to the deck construction (see requirements [17]). As a rule, places, called anchor points of the piping installation, are situated next to the expansion compensators. Namely, in case of axial forces in the installation, anchor points locking the axial pipes move and activate the taking over of deformations and stresses in installation by the expansion loop. At longer installation, anchor points prevent concentrating all postaxial forces on one compensator only, spreading them in proportion to all remaining compensators mounted along all installations. The structure of the anchor points are comparatively straight. It consists of the plate brackets welded directly to the pipe line and the elastic elements (rubber) fastened to the foundation construction – see fig. 3.

With taking over period of the axial reaction forces by an anchor point, the degree of the deformation of the expansion loop grows up as well as the pressure on the elastic element mounted in the anchor point. An aim of the analysis is determination of the expansion loop elasticity characteristics *i. e*. dependencies between the value of reaction forces generated on one

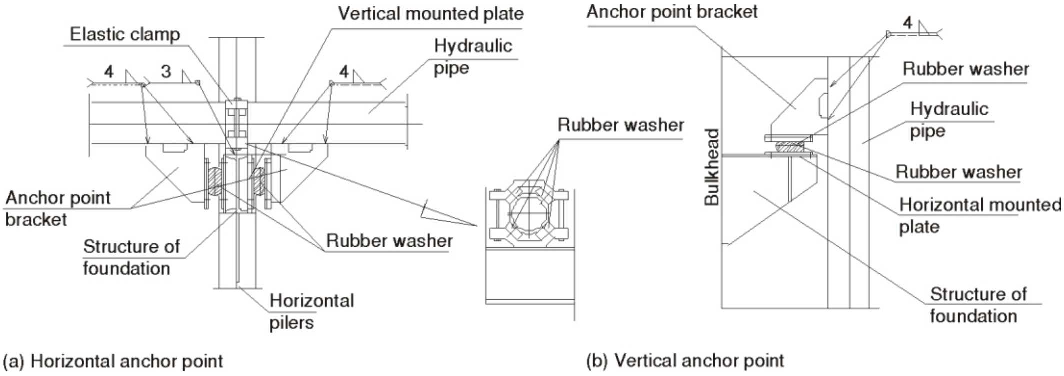

Figure 3. Constructions of anchor points of hydraulic lines [8]
(a) construction of the horizontal type anchor points; (b) construction of the vertical type anchor points

end of the compensator (in the anchor point) and the value of the free axial deformation on the other end. A row of the magnitudes has the influence on the value of the above mentioned function:

- mechanical properties of material whence the expansion loop is executed,
- size of the pipe diameter and its thickness,
- expansion loop dimensions, and
- values of hydraulic oil pressure inside the line.

In FEM analysis model, the right end of loop was fastened by pin type element [18]. Simultaneously left end of loop was free (see model in fig. 5). Assigned loop deformation dx [mm] are given on the left end. The analysis of deformations and generated forces in the pin fastening of the compensator was executed in numeric way by means of FEM system. Calculations were executed with the utilization of the computer system ABAQUS Ver. 6.7 [18] by means of FEM 3-D element S4R type. This is the element of the space type, with 9 nodes and 45 degrees of the freedom (see fig. 4). This element is well suitable to modeling of shell structures, pipes, and installations included. Calculation results of the anchor forces generated in the right end of the loop are given in tab. 4. For determination of the oil pressure influence on the value of the anchor point forces, calculations were executed at taking into account and without taking into account inside pressures of hydraulic oil. Analyzed expansion loop remains as the resilient element with the linear characteristic (see fig. 6). This is a big advantage of this type of expansion compensators. It may explain why analyzed

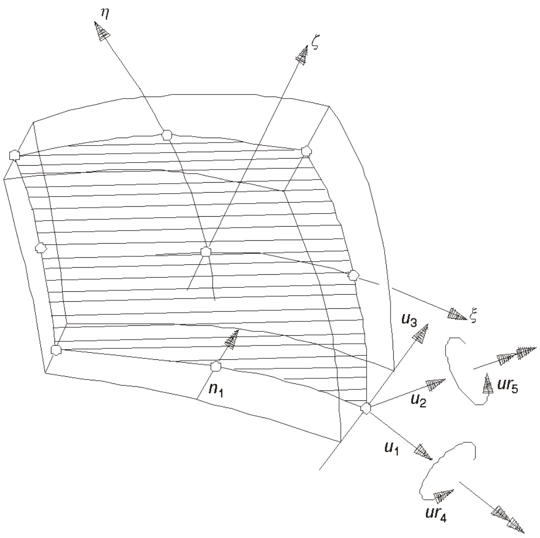

Figure 4. Model of FEM element 3-D ShellS4 type used in the FEM analysis
Source: User's Manual ABAQUS Ver. 6.7 [18, 19]

Table 4. Results of FEM calculations of anchor forces F_A in expansion loop at compression dx

Dimensions of expansion loop "U" type	dx	3-D FEM element S4R type Pressure inside 0 MPa	3-D FEM element S4R type Pressure inside 32 MPa
	[mm]	[N]	[N]
Pipe 130 13 mm L = 2550 mm R_g = 325 mm	5	5096	6794
	10	10180	11900
	15	15252	17000
	20	20312	22100
	25	25360	27200
	30	30395	32300

compensators are so often used in the shipbuilding area and in other industry branches. Numerical calculations are verified by experimental measurements executed in the workshop Framo/Norway [15]. If oil pressure is taken into account or if it is skipped, there are not large errors, not crossing the level of 10%.

In addition to calculations of the anchor forces calculations of stresses in the expansion loops were also carried out. In tab. 5 results of calculations of maximum Huber-Mises stresses σ_{HM} are presented. Calculations were performed in the loop under constructive deformations, with and without taking into account the hydraulic oil pressure. Received dependencies are presented in fig. 8. Analyzed dependence of maximum value Huber-Mises stresses in the loop from size of deformation dx has also a linear character and is strongly related to values of inside oil pressure. Therefore if oil pressure is not taken into account in Huber-Mises analysis, there are large errors unlike the anchor force calculations. Maximum values of Huber-Mises stresses in analyzed loops were situated in places of bends (see fig. 7).

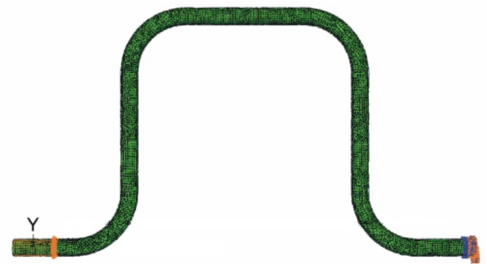

Figure 5. Digitalization of analyzed expansion loop by means of FEM-3-D element ShellS4 type

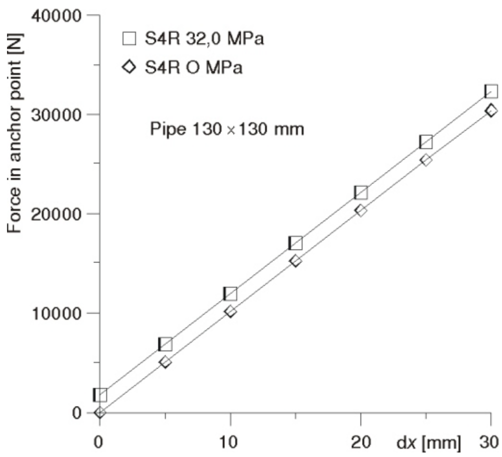

Figure 6. Dependence of the anchor forces values F_A from the dx deformation in expansion loop 130 13 mm

Results and discussion

In this paper we used "U" type expansion loops because this type connections (fixed with-

Table 5. Results of maximum value of Huber-Mises stress σ_{HM} calculations of the expansion loop 130 13 mm size at dx squeezing deformation

Dimensions of expansion loop "U" type	dx	3-D FEM element S4R type Pressure inside 0 [MPa]	3-D FEM element S4R type Pressure inside 32 [MPa]
	[mm]	[MPa]	[MPa]
Pipe 130 13 mm L = 2550 mm R_g = 325 mm	5	39.7	179.0
	10	79.5	213.7
	15	119.3	246.5
	20	159.1	284.7
	25	198.8	321.4
	30	238,6	358.5

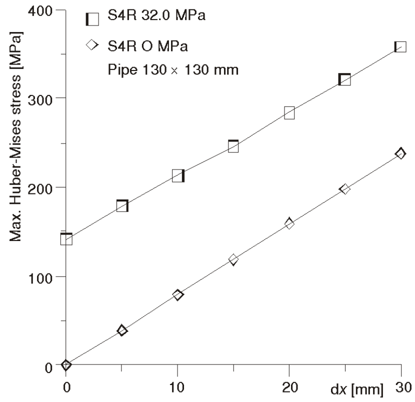

Figure 7. Dependence of the maximum Huber-Mises stresses values σ_{HM} from the d*x* squeezing deformation in expansion loop 130 13 mm

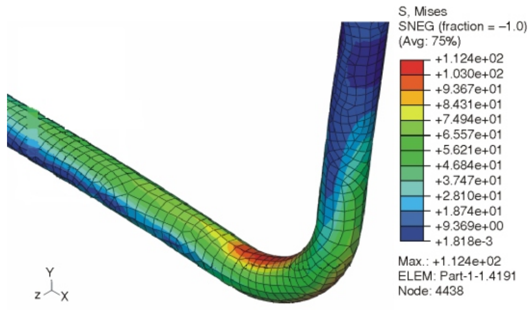

Figure 8. Concentrations of Huber-Mises stresses σ_{HM} in the expansion loop under squeezing deformations (color image see on our web site)

out flexible foses or euivalent) are the most safe, other compensations very often broken in exploitation time on board of tankers or are not sufficient on oil high pressure inside (300 bar). The "V" type expansion loops are NOT used in our area because is big problem with bending.

Please remember that maximum angle in case of bending pipes is 90 degree. It is important in big pipes as my main lines with size diameter 130 mm + thickness 13 mm. In preparation of cold bending (without heating – this is trickly required for hydraulics) of pipes, in case of bending with angle > 90 degree is problem with breaking of material pipes and reduction of material thickness in pipe in bending area in out side of pipe. Therefore "V" type expansion loops are not used in practice on ships.

Scientific importance of this work is high because wrong preparation of hydraulic installations on board of tankers can lead to breaking of hydraulic pipes and a big pollution of sea and water in harbours, also problems with transshipment operations on tankers.

Strictly thermal stress are not to important because stresses from mechanisms of mechanical work of compensations and stresses from high oil pressure inside are a lot bigger. Temperature rise causes on pipes material elongations and problems connect with this phenomena.

Conclusions

Long main pipelines of the hydraulic central loading systems mounted on modern product and chemical tankers are sensitive to thermal stresses and thermal elongations. It can often cause breaking of pipes and damage of very expensive hydraulic system. In order to avoid excessive stresses in these installations it is necessary to mount suitable expansion compensators. Expansion loops of "U" type are very often used in hydraulic systems. The quantity of compensators necessary to install on board the ship is related to the row of factors, as among other things, maximum changes of the pipe material temperature as result of ambient and transfluent hydraulic oil temperature (the size of deformations the hull on the wave included). So far practical approximation methods for calculations and procedures have often made large errors and inflating of the quantity of necessary compensators in the arrangement. FEM numerical calculations with the utilization of shell finite elements provide more exact determination of force values generated in anchor points of the installation and maximum stresses in the moment of deformations. It allows safer and more economical design of the hydraulic installation mounted on decks of modern product and chemical tankers. In Huber-Mises stress analysis in hydraulic pipes, should be taken into account in calculation model, the oil pressure inside pipes. Lack of this, can leads to large errors during maximum stress calculations.

Nomenclature

A_5 – elongation at breaking (min.), [%]
dx – horizontal squeezing deformation of expansion loop [mm]
d_z – outside pipe diameter, [mm]
E – Young's modulus of pipe material, [MPa]
e, f – characteristic dimensions of expansion loop, [mm]
F_A – force in anchor point, [N]
g – pipe wall thickness, [mm]
K – endurance coefficient at temp. 20 °C acc, instr. AD W4, [MPa]
L – total length of expansion loop, [mm]
l – length of the line after the temperature rise of pipe material, [m]
l_0 – start length of the line, [m]
n_1 – normal axis of FEM element
R_g – bending radius, [mm]
$R_{p0.2}$ – yield strength (min.), [MPa]
R_m – stretch endurance (min.), [MPa]
ΔT – temperature rise of pipe material, [K]
u_1, u_2, u_3 – possible axial movements in node of Shell type FEM element
ur_4, ur_5 – possible rotations in node of shell type – FEM element

Greek letters

α_L – coefficient of linear expansion of pipe material, [K^{-1}]
η, ξ, ζ – local axes in FEM element of shell type
σ – temsile stress in pipeline, [MPa]
σ_{HM} – maximum Huber-Mises stress, [MPa]

References

[1] Banaszek, A., Central Hydraulic Loading Systems Used on Board Product and Chemical Tankers (in Polish), *Hydraulika i Pneumatyka,* (2003), 4, pp. 13-17
[2] Banaszek, A., Big Power Central Hydraulic Loading Systems on Board Product and Chemical Tankers, *Proceedings*, 19th International Conference Hydraulics and Pneumatics, Prague, Czech Republic, 2006, pp. 52-59
[3] Banaszek, A., Petrovic, R., Calculations of the Unloading Operation in Liquid Cargo Service with High Density on Modern Product and Chemical Tankers Equipped with Hydraulic Submerged Cargo Pumps, *Strojniski Vestnik*, *56* (2010), 3, pp. 186-194
[4] Banaszek, A., The Concept of Central Hydraulic Loading System on Board Product and Chemical Tankers, *Proceedings*, 1st International Conference Ship Machines and Equipment Operation, Swinoujscie--Ystad – Copenhagen 2003, Scientific Books of Maritime Academy Nr. 68, Szczecin, Poland, 2003, pp.19-28
[5] Banaszek, A., The Concept of Capacity Control of Submerged Cargo Pumps in Liquid Cargoes Transport at Sea, Polish Academy of Sciences, Branch in Gdansk, *Marine Technology Transactions, 16* (2005), pp. 25-32
[6] Deng, Z.-W., Analysis on the Operational Conditions and Material Requirements of the Pressure Pipe, *Pipeline Technique and Equipment, 4* (2009), 6, pp.173-184
[7] ***, Mannesmann Rexroth GmbH, Vademecum of Hydraulics, 3, Design and Structure of Hydraulic Systems, Mannesmann Rexroth Verlag (Polish edition), RPL 00281/10.88, Lohra, Main, Germany 1992 (in Polish)
[8] ***, Stocznia Szczecinska SA, Technological Documentation B578-I /542-1 (Factory Inside Materials) (in Polish), Szczecin, Poland, 1997
[9] Koorey, K., Determination of Optimal Pipe Support Span for Geothermal Pipelines, *Proceedings*, World Geothermal Congress, Kyushu-Tohoku, Japan, 2000, pp. 1361-1364
[10] Fedorov, M., Thermal Modes Simulation of Heat Exchangers Networks Having Turbo-Machines, *Proceedings*, 4th International Conference on Marine Technology, Wessex Institute of Technology, UK, WIT press, Southampton-Boston, 2001, pp. 309-314

[11] Jo, J. C., Choi, J. H., Choi, S. K., Numerical Analysis of Unstady Conjugate Heat Transfer and Thermal Stress for a Curved Piping System Subjected to Thermal Stratification, *Journal of Pressure Vessel Technology, 125* (2003) 4, pp. 467-475
[12] Mahmood, L., Akhanda, Md. A. R., Experimental Study on the Performance Limitation of Micro Heat Pipes of Non Circular Cross-Sections, *Thermal Science, 12* (2008), 3, pp. 91-102
[13] Rajeev, T., *et al*., A Numerical Study for Inward Solidification of a Liquid Contained in Cylindrical and Spherical Vessel, *Thermal Science, 14* (2010), 2, pp. 365-372
[14] Madenci, E., Guven, I., The Finite Element Method and Applications in Engineering Using ANSYS, Springer Science+Business Media LLC, New York, USA, 2006
[15] Vakharia, D. V., Mohd Farooq, A., Determination of Maximum Span between Pipe Supports Using Maximum Bending Stress Theory, *International Journal of Recent Trends in Engineering, 1* (2009), 1, pp. 46-49
[16] ***, Framo Services AS Operation of FRAMO Cargo Pumps, Service Bulletin No. 10, Bergen, Norway, 1993
[17] ***, Stauff – Clamps, Catalogue , Walter Stauffenberg Gmbh, Werdhol, Germany, 2000
[18] ***, Hobbit, Karlsson, Soersen, Inc. ABAQUS ver.6.7 User's Manual, Pawtucket, USA, 2006
[19] Zylisnki, B., Finite Element Local Analysis of Wave Slamming on Offshore Structure, *Polish Maritime Research,18* (2009), 1, pp. 8-12

Paper submitted: June 1, 2010
Paper revised: September 16, 2010
Paper accepted: September 29, 2010