OPTIMIZATION OF PIPE LINE ELASTICITY IN OIL CARRIER

MURINEANU A.*, ANGHEL L., MODIGA M.***

* *"Dunarea de Jos " University of Galatzi*, ** *ICEPRONAV Galatzi*

Abstract: We use a complex modeling with finite elements that allow the consideration of temperature variations, interior pressures and interactions with the body of the ship.
For the cargo transfer installations of oil tankers we study the behavior of eight pipe system models, which have different dilatation lyre (U compensator) arms. This gives the pipe system an extra elasticity and is usually preferred for compensating thermal dilatations and displacements required by the deformation of ship's body. Based on these analyses we can obtain practical recommendations useful in designing.

Keywords: ship pipe line, expansion pipe

1. INTRODUCTION

The studies undertaken on complex models with finite elements of naval piping systems and adjacent structures, that take account of al tasks that arise in exploitation (pressure, temperature variation and imposed displacements caused by the general deformation in the hull of the ship), highlight the appearance of high values of stress main pipe and in particular in the elements of its passage through the watertight cross bulkheads. One of the solutions for reducing this stress is the increase of the pipe system elasticity.

For the purpose of accomplishing ship loading-unloading operations in a time as short as possible, the pipes used in the cargo transfer installations have relatively large diameters. This requires shorter pipe routes, chosen in such a way that the interaction between other elements on the deck or the ship bottom does not lead to hazardous situations. Usually, for this kind of piping, it is preferable to use straight routes, because, the dilatations/shortenings caused by the temperature variations and displacement imposed by the general deformation of the ship hull, are undertaken by the dilatation compressor. However, most of the time, the access to the piping of the cargo transfer installation is done with difficulty. Therefore, for this kind of installations we prefer the compensation solution with U shaped dilatation lyres, which do not require maintenance operations more often than those regarding the tubular material within its assembly.

Table 1

λ	j	λ	j
0,00	1,000	0,30	0,176
0,05	0,762	0,50	0,075
0,10	0,568	0,75	0,035
0,20	0,307	1,00	0,020

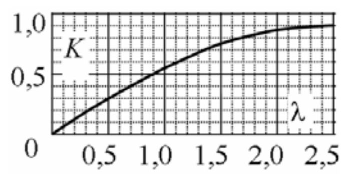
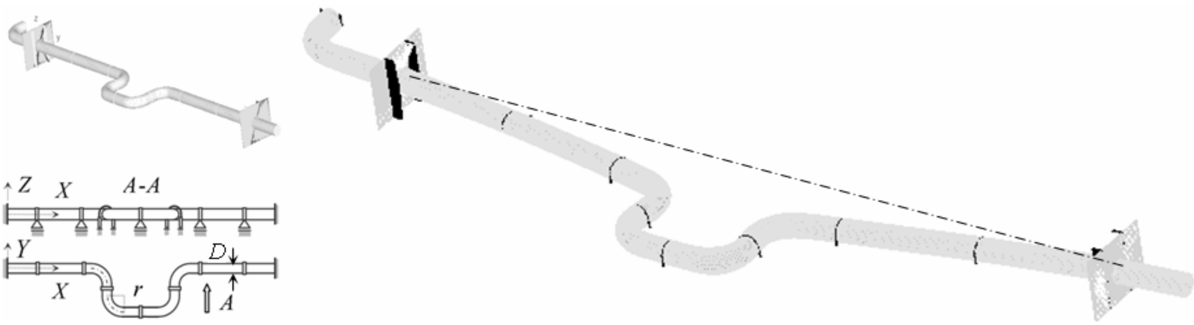


Figure 1

The presence of the dilatation lyre make an asymmetry of the pipe to the axis of the centers of passage sections through the bulkheads – introducing an extra elasticity – hereby producing an atypical deformation o the piping system. Even at the sole action of the interior pressure, the pipe section is deformed in the plane of the compensator, as in figure 2, b. This effect is amplified under the action of the temperature variations and/or displacements imposed by the hull girder deformation. All the elements situated between the bulkheads have negative displacements by Y axis, with the maximum value in the middle of the compensator, although the piping system is fixed on bottom by bridles. The curve sections of the pipes enhance the elasticity of the system because of the decrease in rigidity as a consequence of the elbow sections deformation ([9], [14]). In the analytical method of the elastic center ([**Error! Reference source not found.**]), the highlighting of this effect, at the bending of the piping systems, is made by multiplying the inertial moments of the curve sections with an elasticity coefficient, K, depending on the geometrical characteristic of the elbow, $\lambda = 4sr/D^2$ (s – thickness of pipe). Due to the calculations, based on the cylindrical curve plates theory and minding some corrections experimentally determined, in literature is recommended to use a range of relations for K. Two of the most used are Beski and Markl relation, $K = \lambda/1{,}65$ (used for $\lambda \leq 1$) and Karman relation $K = (1 + 12\lambda^2 - j)/(10 + 12\lambda^2 - j)$, where j depends on λ in accordance with the following table. In the presence of interior pressure p_i, we use a rectified relation $K_p = K\,[(1 + 6(p_i/E)(D/2s)^{7/3}(2r/R)^{1/3}]$.

The dependence with λ of K is displayed in figure. 1. K and λ are used for the determination of the intensification coefficients of normal stress within the curve sections of the pipe. Therefore we define 4 such coefficients: β_i , β_e – for normal axial stress and γ_i , γ_e – for normal circumferential stress; the index numbers i and e refer to the pipe plane(M_i) and respectively extra-plane (M_e) bending indicial. For plane bending, we frequently use the following expressions $\beta_i = 0{,}67K^{-1}\,[(5 + 6\lambda^2)18]^{1/2}$ – for $\lambda < 1{,}472$, $\beta_i = (12\lambda^2 - 2)/(12\lambda^2 + 1)$ – for $\lambda \geq 1{,}472$ and $\gamma_i = 18\lambda(12\lambda^2 + 1)$.

In the naval area, where economical problems, regarding tonnage and space, often appear, it is recommended to use lyre compensators with as small sizes as possible. The decrease in size of the compensator has unfavorable effects upon their compensation capacity. As a consequence of the elbow deformation, the increase in length of the arms of the lyre (U shaped sections that are normal for the pipe) leads to the decrease of the axial force within the straight pipe sections and implicitly the force that acts upon the passages trough the bulkheads. It is not without importance the fact that, along with the length increase of the lyre arms, there is a pipe displacement decrease by the Y axis (see fig. 2, *b*). This fact leads to the decrease of the angle between the deformed pipe and X axis (perpendicular on the bulkhead plane). These angular displacements induce major bending stress within the bulkhead, leading to the forming of important stress hotspots on the circumference of all components of the watertight passage. By reducing the mentioned angle, the axial force in the pipe (caused by thermal effects and/or displacements caused by the general deformation of the hull girder) is uniformly transmitted to the bulkhead, therefore reducing the asymmetrical character of the loading.



a) *b*)

Figure 2

2. STUDIED MODELS

We model a pipe system on the cargo transfer installation of an oil tanker (L = 228 m), pipe situated at 1,5 m above the base-plane in the compartment on the amidships with L = 23910 mm length. The pipe has D = 600 mm diameter, s = 16 mm thickness and is fitted with compensator tip expansion **U** pipe with $r = 1{,}5D$ = 900mm, joints at the crossing through the transversal bulkhead and fixing supports on bottom (see figure 2, *a*). For the study of the influence of the length of the lyre arms, we analyze different dimensional versions, presented in table 2. In all of these versions the section of the lyre that is parallel with the X axis is a = 1200 mm. PLATE type elements were used for the pipes (16 *mm* thick), joints (20 *mm*), neighboring wall zones (16 *mm*), stiffeners (20 *mm*) and welding (20 *mm*). BAR type elements were used for supports (32 *mm* in diameter). Model 1 has the length of the section parallel with Y axis equal with l = 800 *mm*. It contains 31845 nodes and 31917 elements. Models 2…8 (figure 4) have been made throughout successive interlaying of a cylindrical module to Model 1. With a 50 mm length of the FEM on the generating line, the interlayer modules have lengths equal with (4 × 50) j [*mm*], j = 1…7 (152j elements). Besides the pressure p and a temperature variation of ΔT = 50°, the loads given by the ship deforming under the hogging effect were considered. The general hull girder deformation and the displacements transmitted by it to the piping system have been estimated using a DNV Rules based methodology. The models have been realized using Code FEMAP 8.3.

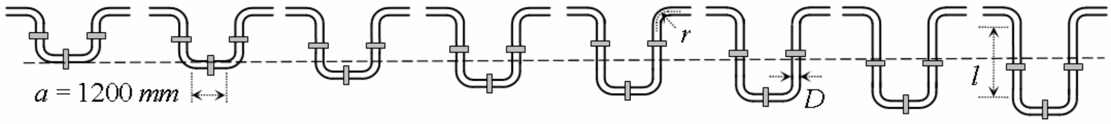
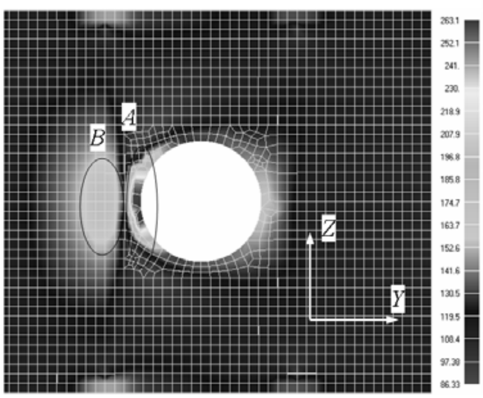


Figure 3



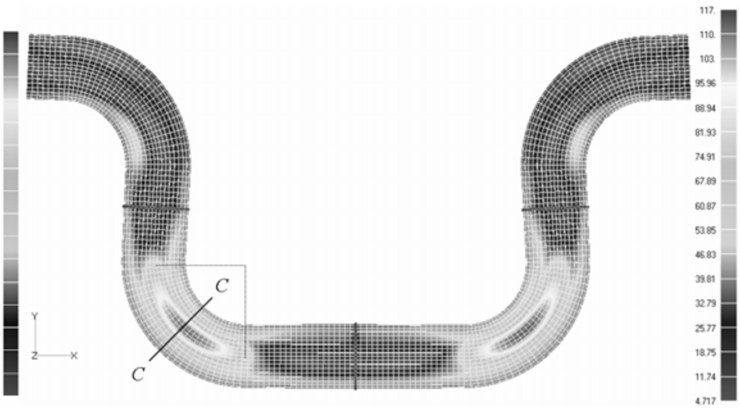


Figure 4

Figure 5

Table 2

Type U-compensator	**1**	2	3	4	5	6	7	**8**
Length l [*mm*]	**800**	1000	1200	1400	1600	1800	2000	**2200**

3. REZULTS AND THEIR PROCESSING

By successively dimensional modifications of the lyre presented in figure 4, after the calculations it was found that the maximum stresses for each component of the piping system has the values presented in Table 3.

Table 3

Type U / Element	**1**	2	3	4	5	6	7	**8**
	Von Mises Stresses [*MPa*]							
Bulkhead	**263**	240,3	222,1	207	195,4	185	177	**171,1**
Joining tube	**249,5**	240	232,3	225,8	220,2	215,5	211,5	**208,1**
Expansion pipe	**117**	106,1	97,59	89,6	83,08	79,9	77,27	**74,74**

Welding joints	**120,1**	118	116,3	114,9	114,4	114,2	114	**113,8**

From analyzing of the obtained results we can see that at the same time with the extension of the lyre arms, the maximum stress in the bulkhead and in the other elements of the watertight passages decreases, but in a different degree. The most important reduction is obtained for the stress in the bulkheads where the difference between Models 1 (with the shortest arm) and 8(with the longest arm) is approximately 90 *MPa*. For these two extreme cases the maximum stress is reduced with approximately 60 *MPa* in the joining tubes, 40 *MPa* in the lyre (see figure 3 too) and only 17 *MPa* in the welding joints. Using the values in table 3 we get the plots in figure 6. The asymptotic behavior of the curves in the plots leads to the conclusion that an exaggerated increase of the arms of the lyre is absurd because it does not produce significant effects. The resulting values and their trend of the variation allow us to choose the optimum solution in terms of stress value and other design criteria.

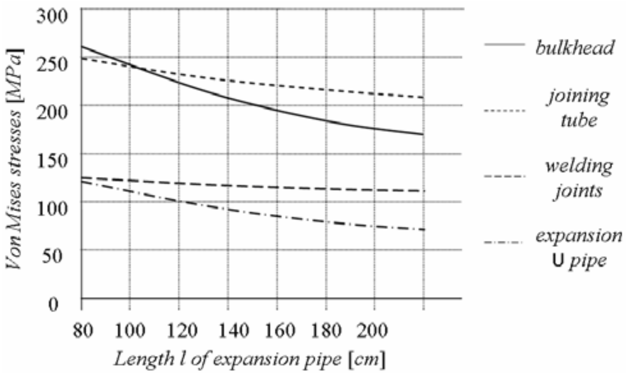


Figure 6

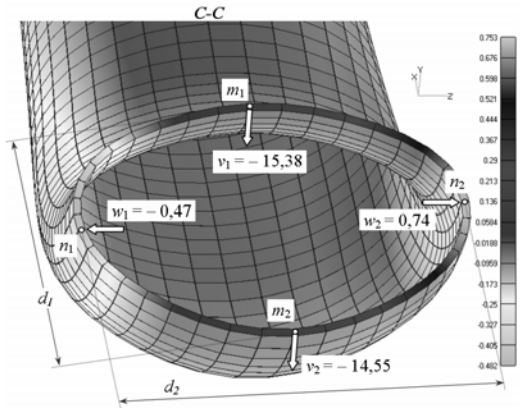


Figure 7

The FEM allows for the obtaining of the stress chart on all circumference of the piping system as well as the deformation of all cross sections of the pipes and elbows (figure 7). Hereby, for section C-C in figure 5, disposed at a 45^o angle from the XZ and YZ planes, for the points m_1, m_2 and n_1, n_2 we get the variations $\Delta v = v_2 - v_1 = -0{,}83$ *mm* respectively $\Delta w = w_2 - w_1 = 1{,}21$, and from generalized Theorem of Pythagoras it results that: $d_1 = (d_o^2 + \Delta v^2 - 2d_o\Delta v\cos45^o)^{1/2} = 576{,}4$ *mm*, $d_2 = (d_o^2 + \Delta w^2 - 2d_o\Delta w\cos45^o)^{1/2} = 577{,}9$ *mm*, where $d_o = 577$ *mm* is the interior diameter of the no deformed pipe. The deformation of the elbow section is $d_1 - d_2 = 1{,}5$ *mm* and the relative deformation is $f_o\,(\%) = (d_2 - d_1)/d_o = 0{,}26$ %.

4. CONCLUSIONS

In ship piping system with a relatively high rigidity there are important loadings due to the general deformation of the hull girder, witch is cumulated with loadings due to the interior pressure and temperature variations. In the same time these kinds of pipes produce dangerous stress states in the cross bulkheads witch they pass. An optimum choice of the lyre compensators can lead to the decrease of the stress in the bulkhead and in the other components of the watertight passage.

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