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OA	ION Raluca, MUŞAT, Sorin Dumitru, CHIRICĂ Ionel, BOAZU Doina, and BEZNEA Elena Felicia. Torsional analysis of ship hull model. *The Annals of University "Dunărea de Jos" of Galați, Fascicle VIII, Tribology.* 15(2). 2009. p.168-173.

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This article has been retracted as it is found to contain a substantial amount of materials from the previously published papers: (1) Ionel Chirica, Sorin Dumitru Musat, Raluca Chirica, and Elena-Felicia Beznea, “Torsional behaviour of the ship hull composite model,” Comput. Mater. Sci., Vol. 50, pp. 1381-1386, 2011; (2) Ion Raluca, Musat Sorin Dumitru, Chirica Ionel, Boazu Doina, and Beznea Elena Felicia, “Torsion analysis of ship hull made of composite materials,” Materiale Plastice, Vol. 47, Issue 3, pp. 364-369, 2010; (3) Ion Raluca, Musat Sorin Dumitru, Chirica Ionel, Boazu Doina, and Beznea Elena-Felicia, “Torsional analysis of ship hull model,” The Annals of University “Dunarea De Jos” of Galati, Fascicle VIII, 2009 (XV), ISSN 1221-4590, Issue 2, Tribology.

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A Numerical Model for Torsion Analysis of Composite Ship Hulls

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TORSIONAL ANALYSIS OF SHIP HULL MODEL

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ABSTRACT

In this paper a new methodology (numerical and experimental) proposed to analyze the ship hull torsion is treated. The torsion analysis is performed on a scale model (1:50) of a container ship, made of composite material. The outline of the section is considered as a polygonal one. The material is orthotropic. For a straight line portion of cross section outline is corresponding a longitudinal strip plate. Due to the torsion of the thin walled beam, in the strip plate, the stretching-compression, bending and shearing occur. The strip plate is treated as an Euler-Bernoulli plate. The stiffness matrix of the macro-element is obtained by assembling the stiffness matrices of the strips. The results obtained from numerical and from experimental analysis are presented.

KEYWORDS: Torsion, ship hull, thin-walled beam, composites.

1. INTRODUCTION

Ship hull with large deck opening has a very poor torsional stiffness. Due to its particular construction, ship hull structure can be considered as thin-walled structures. Plates and shells have one physical dimension, their thickness, small in comparison with their other two dimensions. In thin walled beams all three dimensions are of different order of magnitude. For such structures, the wall thickness is small compared to any other characteristic dimension of the cross-section, whereas the linear dimensions of the cross-section are small compared with the longitudinal dimension. Due to their wide applications in civil, aeronautical/aerospace and naval engineering, and due to the increased use in their construction of advanced composite material systems, a comprehensive theory of thin walled beams has to be developed: this is one of the aims of this paper. The aim of the work is to analyze the influence of the very large open decks on the torsion behaviour of the ship hull made of composite materials. One should in principle be able to derive the theory for beams, shells and plates, and massive bodies by using the equations of 3-D continuum theory, and taking advantage of the factors which serve to distinguish each type of structure. In this acceptance, the theory of plates and shells constitutes a two-dimensional approxi-mation of the three-dimensional elasticity theory, while solid cross section and thin/thick walled beams are both one-, two- and three- dimensional approximations of three-dimensional continuum theory and constitute an enormous challenge for someone who is not familiar with the capabilities that the implementation of the tailoring technology can provide. In spite of this commonality, the theory of thin walled beams is basically different from that of solid cross-section beams. The new ship structures have to provide higher performances, unattainable by the classical structures built of traditional materials. The advent of advanced composite materials, of smart materials and functionally graded materials has constituted the strongest stimuli for such developments.

Moreover, their incorporation is likely to expand the use and capabilities of thin-walled beam structures. The new and stringent requirements imposed on ship structures will be best met by such new types of material structures.

However, incorporation of these new material structures in the various areas of advanced technology and the solution of many challenging problems involving their static/dynamic response, stability and control, requires a good understanding of the various aspects of their modeling and computational methodologies. While the directionality property of composite materials provides new degrees of freedom to the designer, enabling him to achieve greater structural efficiency, it constitutes an enormous challenge for someone who is not familiar with the capabilities that the implementation of the tailoring technology can provide.

2. THIN WALLED BEAM MODEL

In the paper, the new methodology proposed to analyse the ship hull torsion as thin walled beam using macro elements is treated. The outline of the section is considered as polygonal one. The material is an orthotropic one. For a straight line portion of cross section outline is corresponding a longitudinal strip plate. Due to the torsion of the thin walled beam, in the strip plate the stretching-compression, bending and shearing occur. The strip plate is treated as an Euler-Bernoulli plate. The stiffness matrix of the macro-element is obtained by assembling the stiffness matrices of the strips.

Two coordinate systems are used:

- global system O_oXYZ having axis OoX along the torsion centers line of the cross sections of the beam;

- local system attached to each plane k ($F_k^o x_k y_k z_k$) having the axis $F_k^o x_k$ parallel with OX.

2.1. Cross Section Type Open

The hypotheses of the Vlasov theory are used:

- the material is linear-elastic, homogeneous, orthotropic generally, having the coordinate system $F_k^o x_k y_k z_k$ as the main orthotropic axis;

- the shear stresses occurring in the beam cross section are parallel with the median line Γ.

During the beam deformation, the median line Γ does not remain plane. The projection of the median line on the cross section plane remains the same as its initial shape (non-deformed outline hypothesis). For small displacements, the displacement v of the current point F placed on the median line has the equation

$$v(x,s) = \tilde{r}(s)\varphi(s) \qquad (1)$$

The displacement u along the axis OoX of the point F is considered as constant on the wall thickness. The displacement u is considered to be in the form:

$$u(x,s) = -\omega(s)\varphi'(s) \qquad (2)$$

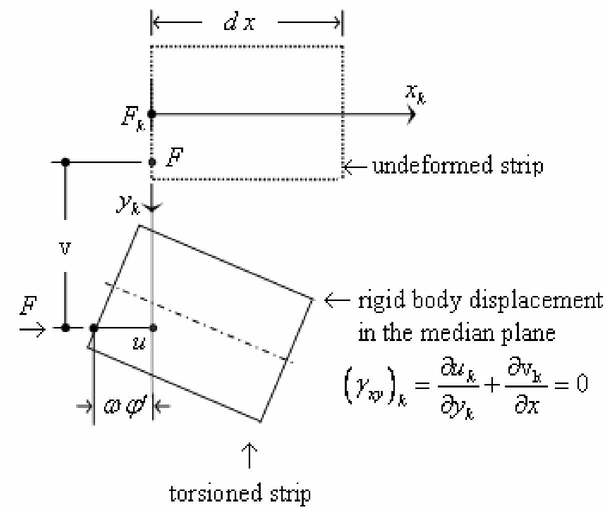

Fig. 1. Strip deformation.

The torsion of the thin-walled beam generates the torsion of the strips and the loading of the strips in their plane.

Using the equations (1) and (2) for a strip k, it may be written:

$$v(x,s) = \tilde{r}_k(s)\varphi(s) \qquad (3)$$

$$u_k(x, y_k) = -\omega(y_k)\varphi'(s) \qquad (4)$$

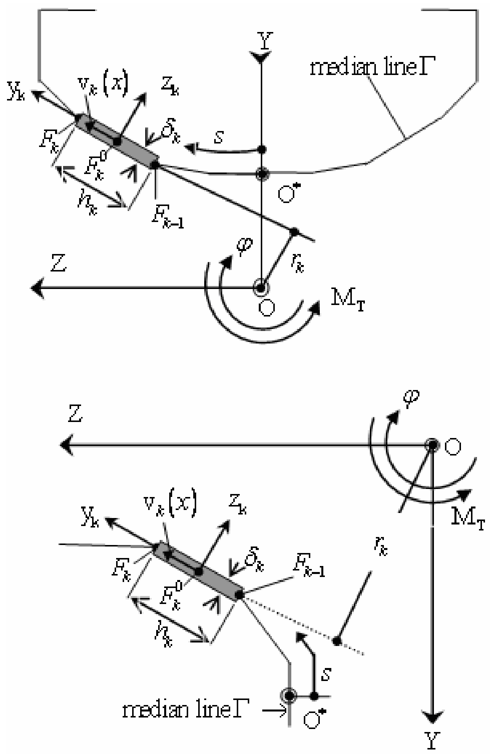

Fig. 2. Cross section of the macro-element.

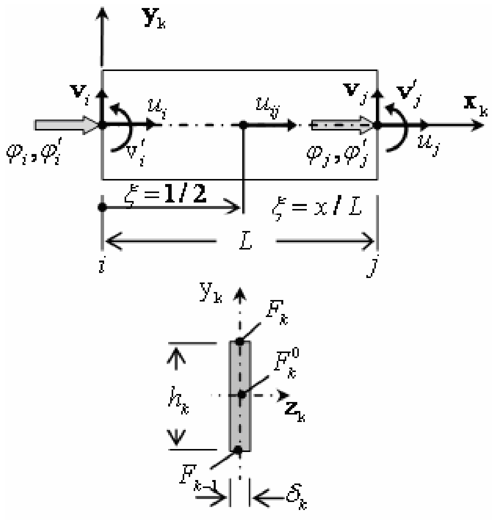

Fig. 3. Finite strip element.

For the displacement u, due to the tension-compression loading of the strip k the approach function is a parabolic one, having the form

$$u_k(\xi) = P_1(\xi)u_i^k + P_2(\xi)u_{ij}^k + P_3(\xi)u_j^k \qquad (5)$$

2.2. Cross Section Type Close

For the case of closed section, we assume that u is proportional to the generalized sectorial co-ordinate $\hat{\omega}$ evaluated to O and O^*. Different from classical theory or Benscoter theory, we assume that u is proportional to the rate of twist:

$$u(x,s) = -\hat{\omega}(s)\,\varphi'(x)$$

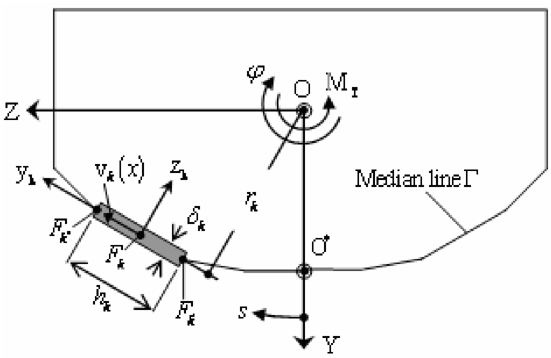

Fig. 4. Poligonal closed cross section.

The generalized sectorial co-ordinate is defined as

$$\hat{\omega} = \omega - \tilde{\omega},$$

where

$$\omega(s) = \int_0^s r(s)\,ds \quad , \quad \tilde{\omega} = \omega_0 \tilde{s}/\tilde{S} \quad ,$$

$\omega_0 = \int_\Gamma r(s)\,ds$ - the double of the area surrounded by Γ

$$\tilde{s} = \int_0^s \frac{ds}{\delta(s)} \quad , \quad \tilde{S} = \oint_\Gamma \frac{ds}{\delta(s)} = \sum_{k=1}^n \frac{h_k}{\delta_k}$$

where n is the number of strip-plates.

The torsional loading of the beam generates a planar loading of the strip-plate. For each strip-plate, one obtains

$$v_k(x) = r_k\,\varphi(x), \qquad (6)$$
$$u_k(x, y_k) = -\hat{\omega}(y_k)\,\varphi'(x)$$

The equations 6 define the displacement field for each stripe-plate. The continuity of the displacement u along the jointing edges between two stripe-plates is embedded in the above relation. The linear variation of $\hat{\omega}_k$, the generalized sectorial co-ordinate along the axis y_k (in the reference system $F_k x_k y_k z_k$ associated to stripe-plate k) may be expressed as

$$\hat{\omega} = \hat{\omega}_k + (\hat{\omega}_{k''} - \hat{\omega}_{k'})\eta$$

where

$$-1/2 \le \eta = y_k/h_k < 1/2,$$

The coordinates $\hat{\omega}_k$, $\hat{\omega}_{k'}$, $\hat{\omega}_{k''}$ characterize the points, F_k, F_k', F_k'' ($\hat{\omega}_k = (\hat{\omega}_{k'} + \hat{\omega}_{k''})/2$).

For the longitudinal displacement, one obtains

$$u_k(x, y_k) = -\left[\hat{\omega}_k + (\hat{\omega}_{k''} - \hat{\omega}_{k'})\eta\right]\varphi'(x)$$

Using the hypothesis, the strain generated in the stripe-plate k is

$$\varepsilon_k = \frac{\partial u_k}{\partial x} = -\left[\hat{\omega}_k + (\hat{\omega}_{k''} - \hat{\omega}_{k'})\eta\right]\varphi''(x),$$

$$\gamma_k = \frac{\partial u_k}{\partial y_k} + \frac{\partial v_k}{\partial x} = \Delta_k\,\varphi'(x)$$

where $\Delta_k = \omega_0/(\tilde{S}\,\delta_k)$.

Normal stresses σ_k appear in each stripe-plate k due to the warping, having the equation

$$\sigma_k(x, y_k) = -E\hat{\omega}(y_k)\,\varphi''(x) \qquad (7)$$

In each cross-section, these stresses form a system of distributed forces in self-equilibrium.

The tangential stresses τ_k associated with the deformations γ_k have the equation

$$\tau_k(x) = G\gamma_k = \frac{G\omega_0}{\tilde{S}}\frac{1}{\delta_k}\varphi'(x) \qquad (8)$$

The flux of these stresses, $\tau_k\delta_k$, is constant for each section of thin-walled beam.

The differential equation of the twist angle φ obtained by the Ritz method is

$$EI_{\hat{\omega}}\varphi''' - GI_T\varphi' = -M_T(x) \qquad (9)$$

where

$I_{\hat{\omega}} = \sum_{k=1}^n I_{\hat{\omega}k}$, $I_{\hat{\omega}k} = h_k\delta_k\left[\hat{\omega}_k^2 + (\hat{\omega}_{k''} - \hat{\omega}_{k'})^2/12\right]$ is sectorial moment of inertia,

$I_T = \omega_0^2/\tilde{S}$ (conventional polar moment of inertia), M_T is the transmitted torque.

The differential equation reveals two components of the transmitted torque:

$M_\gamma = GI_T\varphi'$ (Saint Venant torque),

$M_\varepsilon = -EI_{\hat{\omega}}\varphi'''$ (warping torque)

The component M_γ of the transmitted torque is the part associated with the strain γ_k and stress τ_k (Saint Venant torsion). The component M_ε is the part of the transmitted torque associated with the shear forces generated by strip-plates bending, can by obtained only from equilibrium condition.

For the displacements $\varphi(\xi)$ and $v_k(x)$ polynomial functions (third order) are chosen:

$$\varphi(\xi) = H_1(\xi)\varphi_i + LH_3(\xi)\varphi'_i + H_2(\xi)\varphi_j + LH_4(\xi)\varphi'_j \tag{10}$$

$$v_k(\xi) = H_1(\xi)v_i^k + LH_3(\xi)\theta_i^k + H_2(\xi)v_j^k + LH_4(\xi)\theta_j^k . \tag{11}$$

For bending and torsion, the well known matrices of the beam are used

$$\mathbf{k}_v^k = \frac{EI_k}{L^3}\begin{bmatrix} 12 & 6L & -12 & 6L \\ & 4L^2 & -6L & 2L^2 \\ & & 12 & -6L \\ \text{symm.} & & & 4L^2 \end{bmatrix};$$

$$\mathbf{k}_\varphi^k = \frac{GI_{T_k}}{L}\begin{bmatrix} 6/5 & L/10 & -6/5 & L/10 \\ & 2L^2/15 & -L/10 & -L^2/10 \\ & & 6/5 & -L/10 \\ \text{symm.} & & & 2L^2/15 \end{bmatrix}. \tag{12}$$

In the methodology, the classical thin-walled beam theory for isotropic materials was used. Taking into account the materials characteristics, the orthotropy of the material was considered.

The equivalent stiffness coefficients for the tension-compression, bending and shearing loading of the strip k are determined.

$$(EA)_k = 2\left(\sum_{i=1}^{ns} E_{i,k} g_{i,k}\right) h_k$$

$$(EI)_k = 2\sum_{i=1}^{ns} E_{i,k} I_{i,k} = \frac{1}{6}\left(\sum_{i=1}^{ns} E_{i,k} g_{i,k}\right) h_k^3$$

$$(GI_T)_k = 8\left(\sum_{i=1}^{ns}\left(G_{i,k} z_{i,k}^2 g_{i,k}\right)\right) h_k \tag{13}$$

$$E_k = \frac{2}{\delta_k}\sum_{i=1}^{ns} E_{i,k} g_{i,k}$$

$$G_k = \frac{24}{\delta_k^3}\left(\sum_{i=1}^{ns}\left(G_{i,k} z_{i,k}^2 g_{i,k}\right)\right)$$

The equations (13) are determined according to figure 5.

Finally, the results obtained with the proposed methodology for a prismatic hull beam are compared with the ones obtained with analytical solutions.

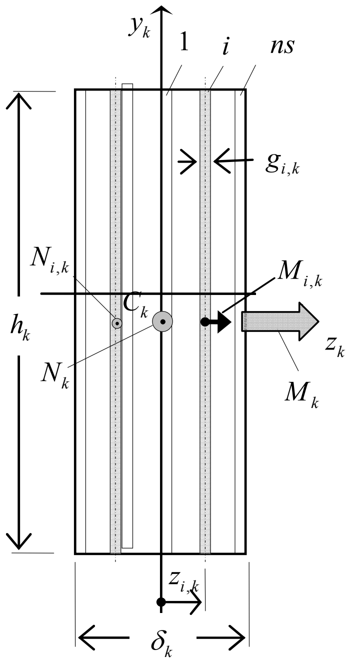

Fig. 5. Thickness of the plate (lay-out)

3. NUMERICAL ANALYSIS

For the present study, a soft based on the theory presented above was done. The results obtained with this code were compared with the ones obtained with COSMOS/M FE soft. A 3-D model with 4-node SHELL4L composite elements of COSMOS/M was used. The ship hull model was loaded by a torque M_x in the midship. Due to the fact, the real ship has much stiffened structure in the both end, the model is clamped at the ends. In figures 7 and 8, the stress state on the deformed ship hull numerical model, according to the numerical calculus with COSMOS/M is shown.

4. EXPERIMENTAL ANALYSIS

The experimental test on the composite model of a prismatic ship was done. The model has the main characteristics: length L=2.4 m, breadth B=0.4 m, depth D=0.2 m. The material is E-glass/polyester having the characteristics, determined by experimental tests:

E_x=46 GPa, E_y=13 GPa, E_z=13 GPa,
G_{xy}=5 GPa, G_{xz} = 5 GPa, G_{yz} = 4.6 GPa,
μ_{xy}=0.3, μ_{yz}=0.42, μ_{xz}=0.3.

The thicknesses' values of the hull shell are 2 mm for side shell and 3 mm for deck and bulkheads.

In figure 11, the experimental rig for torsion of the ship hull model is presented. The stress state in the ship shell was determined by the strain gauges measurements. The results both for numerical and experimental studies are presented in figure 10.

The torsion angle of the ship hull model cross section is determined by taking into account the displacements of the points placed on the outline (fig. 6), according to the rotation of the rigid body (thin-walled beam hypothesis).

The displacements were obtained with test rig (LVDT equipment, as it is seen in figure 11).

The stress state obtained with the FE analysis was compared with the values of the stresses determined by measurements done in the bulkheads sections.

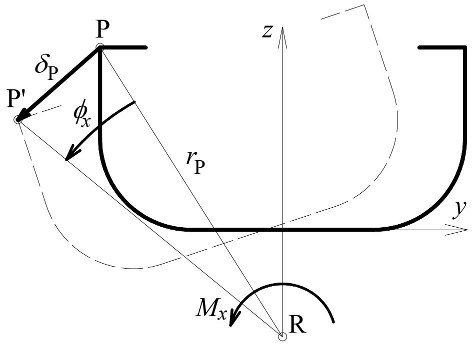

Fig. 6. Calculus of experimental torsion angle.

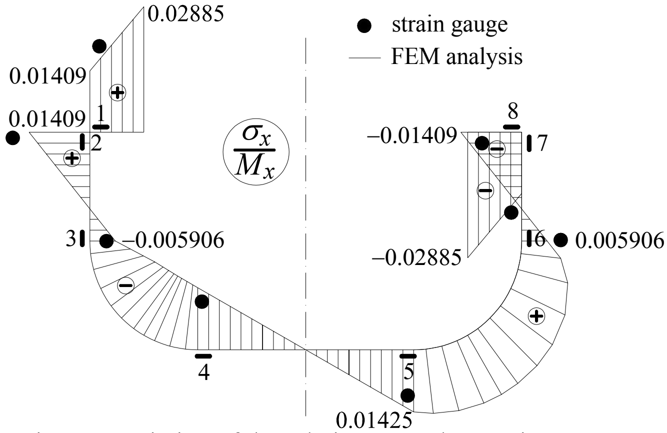

Fig. 10. Variation of the relative normal stress in the midship open section.

Fig. 11. Torsion rig for experiments.

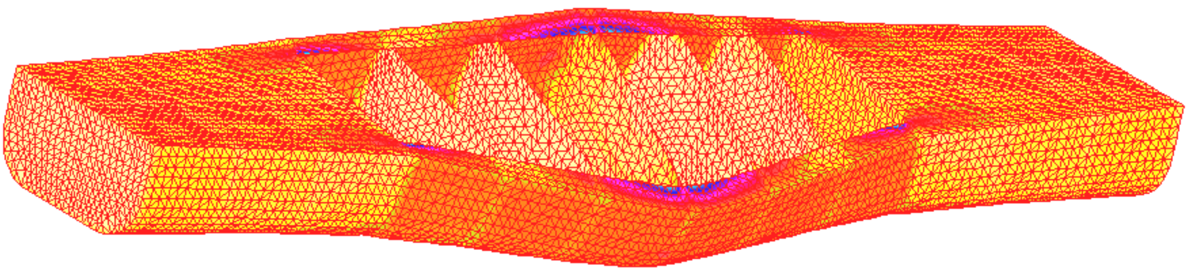

Fig. 7. Deformed FEM model.

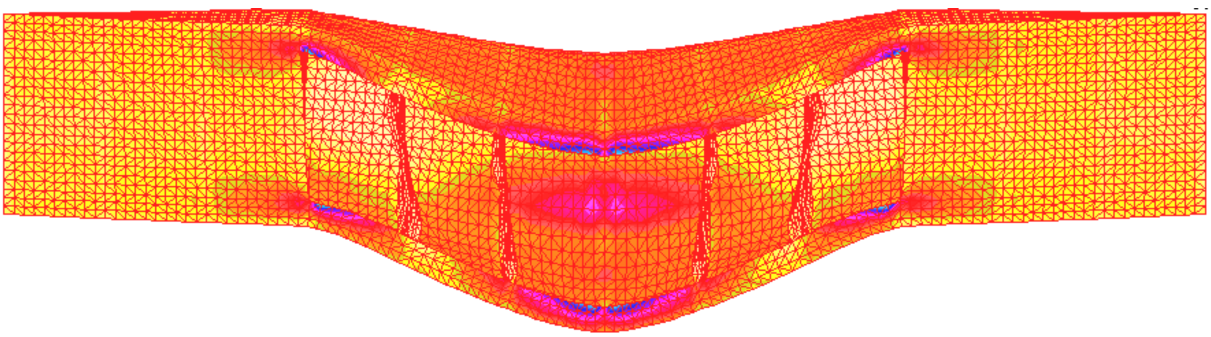

Fig. 8. Deformed FEM model: torsion coupled with horizontal bending.

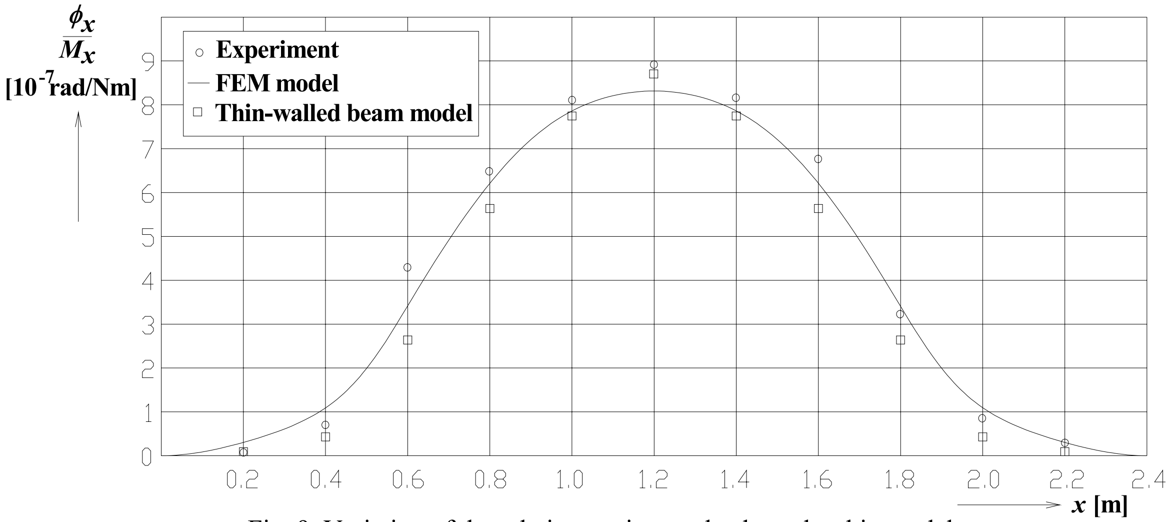

Fig. 9. Variation of the relative torsion angle along the ship model.

5. CONCLUDING REMARKS

In figures 7 and 8, the deformed ship hull numerical model, according to the numerical calculus with COSMOS/M is shown. Due to the variation of the cross section shape of the model, a coupled torsion with lateral bending occurred (fig. 8).

In figure 11, the experimental rig (strain gauge and LVDT equipment) used for torsion of the ship hull model is presented.

The values obtained with FE analysis and according to the strain gauges measurements are presented.

The variation of the relative normal stress in the midship open section is presented in figure 10. In the figure, only results obtained in FEM analysis and in experimental tests for the 10 measure points are presented. Due to the fact that the variation of the normal stress is linear, the variation of the ratio (σ_x/M_x) was plotted with continuum line. The values of the stresses obtained with strain gauges were plotted in figure 10.

The variation of the relative torsion angle (ϕ_x/M_x) along the ship model obtained so from FEM analysis, thin-walled beam model and from experimental measurements is presented in figure 9.

Due to the closed section type in the ends, the torsion stiffness of the model in these areas is much higher than in the middle part. Thus, as it is seen in figure 8, the maximum value of the relative torsion angle (ϕ_x/M_x) in the midship is almost 2 times more than the maximum torsion angle in the closed area.

The new thin walled beam methodology proposed for torsion analysis of the ship hull may be considered as a good tool for a very quick torsion calculus.

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