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A Design Method for the Prediction of the Behavior of Moorings

Filippo Cucinotta[a1] and Felice Sfravara[a]
[a] *Department of Engineering, University of Messina, Contrada Di Dio (S. Agata), 98166 Messina, Italy.*

[1]ORCID: 0000-0002-0304-4004, Scopus Author ID: 57192216810
[2]Scopus Author ID: 57192214583, ORCID: 0000-0003-3922-8494

Abstract
The ships are constrained mainly by mooring lines and fenders to rigid structures such as jetties and piers. The evaluation of the forces that affected the mooring lines and fenders is essential for proper design of all components of the plan such as the chocks, the cables and the bollards. A calculation software that can estimate the loads acting and how they are distributed is essential to be able to perform these calculations with accuracy. The article describes the design phases of the program and the theoretical aspects discussed and is a guide for designers who want to predict the behavior of the mooring plan.

Keywords: Moorings optimization, berthing, ship design, safety, environmental forces

INTRODUCTION

The mooring of a berthed ship is a complex theme and it involves different aspects. A moored ship is constrained with different methodologies and it is free to move itself in response to many external factors such as: loading and unloading cargo, waves, wind, tides and currents. These movements are bounded inside limits dependent on the response of the mooring equipment. Therefore they are depending on the method used for constraining the ship [1]. Cables and fenders are the principal elements of a mooring equipment. The first responses exclusively to tensile forces, the second reacts only to compression forces. One of the first analytical method for solving the mooring problem of a berth ship was proposed by Chernjawski [2]. This method involves the reduction of the non-linearity equations that govern the physic of the problem in an algebraic one and the solution is founded with a matrix approach for a particular condition of environmental effects. Natarajan et al. [3] proposed an alternative analytical method with a more one step: each external condition is evaluated with empirical formulations in order to account the actual condition of the ship under investigation. This methodology helps to provide a good mooring design for each environmental condition with fast and accurate results. The empirical formulations are very good for preliminary design but it is clear that more accurate results can be founded with the new technologies such as : Computational Fluid Dynamics for the interaction of the ship with the external forces [4–7] and structural analysis for the evaluation of the final resistance of the cables and fenders [8–13]. Naturally, these methods are too much complex and time consuming and usually the choice for solving the mooring problem is oriented to analytical methods with empirical formulations. A fast and accurate assessment of the forces involved during the mooring in each environmental condition allows also to evaluate the possible failure of the cables, so reducing problems such as collisions with the port quay [14] or unexpected movements of the ship during the loading and unloading operations. These events could lead to dangerous conditions in term of safe for people and for environmental pollution [15, 16] and so it's very important to conduct this calculation rapidly in order to follow the fast changes of the external environmental conditions. The mooring problem of a berthed ship can be studied with the aid of matrix methods taking in account the unilaterality of the constraints. In this study the basis of the analytical approach follow the studies proposed in technical literature but new improvements are proposed with particular focus on the accuracy, rapidly and simplicity of the methodology.

NOMENCLATURE

Definition	Symbol	Formula	Unit
Cross-Sectional Area of i-th mooring line	A_i	-	m^2
Wind angle direction	η		
Current angle direction	ε		
Direction of the wave	μ		
Waterplane area of the Ship	A_W	-	m^2

[1] Corresponding author

Wind exposed projected area in i-th direction	A_{pi}		m^2
Current exposed projected area in i-th direction	A_{ci}		m^2
Breadth of the Ship	B	-	m
Coordinates of the ith bollard	B_i (B_1,B_2,B_3)	-	m
Coordinates of the ith chock	C_i (C_1,C_2,C_3)	-	m
Matrix of i-th chock coordinates	$[\mathbf{C}]_i$		m
Wind drag coefficient along the ζ direction	$C_w(\zeta)$		
Current drag coefficient along the ε direction	$C_C(\varepsilon)$		
Added mass coefficient of the ship	C_m		
Depth of the water	D		m
j-th fender reaction force	D_j		N
Deflection of the j-th fender	d_j		m
Draught of the ship	d	-	m
Young's modulus of the mooring lines	E	-	MPa
Stiffness of j-th fender	f_j		N/m
Centre of Gravity of the Ship	G_i	-	m
Gravity acceleration	g		m/s^2
Longitudinal metacentric	GM_L		m
Transversal Metacentric	GM_T		m
Incident wave height	H		m
Length of the mooring line	l		m
Unit vector of the generic mooring line	$\mathbf{l}_i$		
Reaction moments of i-th mooring line	$\mathbf{m}_i$		N m
Vector of external moments	$\mathbf{M}$		
Number of mooring lines	n		
Number of fenders	N		
Vector of the external forces	$\mathbf{Q}$		
Axis centered to Gravity of the Ship	$X_{1,2,3}$	-	-
Axis centered on the generic chock	$X_{1i,2i,3i}$		
Stiffness of i-th mooring line along its own axis	k_i		N/m
Stiffness of jth fender	k_j		N/m
Displacement vector of the ship	$\mathbf{U}(U_1,U_2,U_3)$	-	m
Displacement vector of the chock	$\mathbf{U_{CI}}$		m
Velocity of the Wind	$\mathbf{V_{Wind}}$		m/s
Velocity of the current	$\mathbf{V_{Current}}$		m/s
Displacement of the Ship	Δ	-	t
Angular vector of the ship	$\boldsymbol{\theta}(\theta_1,\theta_2,\theta_3)$	-	rad
Wave number	k		
Matrix of projected stiffness	$[\mathbf{K}]_i$		
Mass density of the air	ρ_a		
Mass density of the water	ρ_W		
Angle between Segment mooring Line and plane $X_{1i}X_{2i}$	α_i		
Angle between Segment mooring Line and plane $X_{1i}X_{3i}$	β_i		
Angle between the mooring lines and its local axis	$\boldsymbol{\varphi}(\varphi_{1i},\varphi_{2i},\varphi_{3i})$		

In the equations, the vectors are identified by bold fonts and the matrixes by square brackets.

MATERIALS AND METHODS

The mooring problem - Reaction forces:

The mooring lines that constrain a ship have a generic spatial orientation and assume, in function of the force with which they are put in tension, a form of catenary.

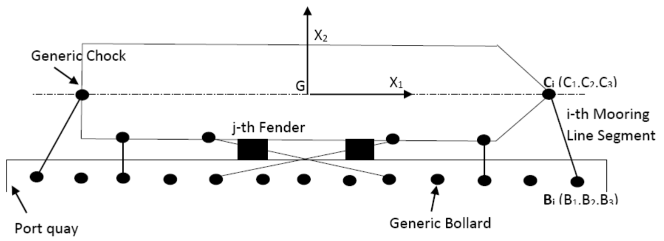

Figure 1. Generic mooring for a berthed ship

There are different lines during the mooring of a berthed ship and the name of each one of them is influenced by the constrain that they apply on the ship (bow lines, spring lines, breast lines and stern lines). In general, each line can be described, with an acceptable error, like a straight-line segment that starts on the ship in a chock and ends on the ground in a bollard. In this way, the catenary is approximated as a straight line but the error is very small when, as it is normally, the cable is sufficiently tense. Similarly, the contact between ship and fender can be taken as punctual, rather than superficial as in reality. Even this error can be considered acceptable in most cases because the size of the fenders is very small compared to the size of the ship. In the case in which the approximation is not acceptable it is possible to use a discretization of the fender in more points of contact neighbors. The system studied in fact has no limits in the number of mooring lines and fenders. The general reference used is a fixed reference system centered on the center of gravity of the ship and with the longitudinal axis X_1, positive towards the bow, X_2 positive to starboard and X_3 positive in vertical direction, as shown in Figure 1. In the picture are shown also the three orientation angles of ship respect the three axis. The position of the ship is identified by the displacement vector **U (u l θ)**, with the generic displacement u_1,u_2,u_3 and the angles θ_1, θ_2 e θ_3.

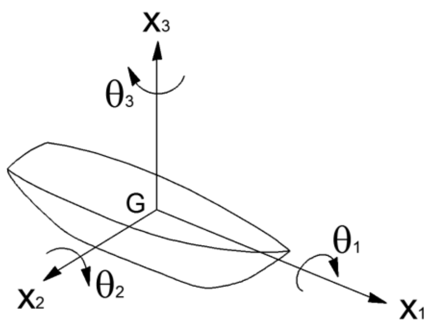

Figure 1. Reference system and relative angles for motion of ship

The mooring lines:

A local reference system, parallel to the general one, for each mooring line and centered on the i-th chock, with the axes X_{1i}, X_{2i} and X_{3i}, is defined in order to treat the generic mooring line such as segment. The origin is in the chock position defined by the vector $\mathbf{C_i}$(C_{1i},C_{2i}.C_{3i}) respect to the general reference frame, while the end of this segment is in the relative i-th bollard with position $\mathbf{B_i}$(B_{1i},B_{2i},B_{3i}). The line is geometrically defined by these points and by the angle α_i (angle between the segment line and the Plane $X_{1i}X_{2i}$) and the angle β_i (angle between the Segment Line and the Plane $X_{1i}X_{3i}$) as shown in Figure 2.

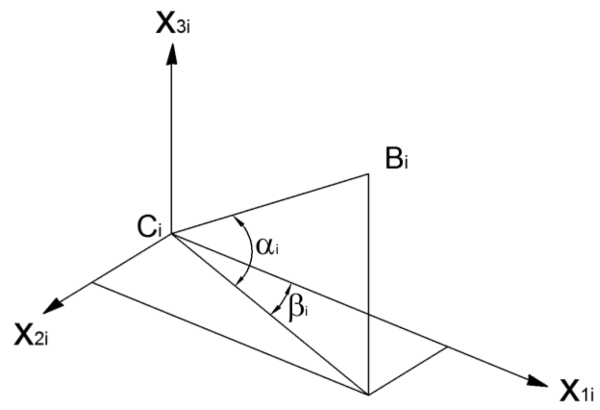

Figure 2. Segment mooring line and angles with Local Reference system

The unit vector $\mathbf{l_i}$ is parallel to the i-th segment mooring line, it is defined with the **φ** vector angle with the components (φ_{xi},φ_{yi},φ_{zi}) solved with the Equations (1-3) and shown in Figure 3.

$$(cos\varphi_1)_i = cos\alpha_i cos\beta_i \quad (1)$$

$$(cos\varphi_2)_i = cos\alpha_i sin\beta_i \quad (2)$$

$$(cos\varphi_3)_i = sin\alpha_i \quad (3)$$

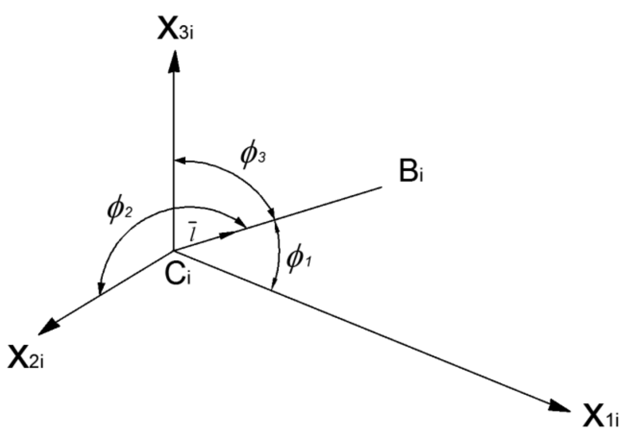

Figure 3. The angles of the Segment mooring line with the local reference system

Each mooring lines is considerable as a spring which has its elastic stiffness along its axis of work. The generic ropes reacts with a force only if it is stressed along the direction of tension. The i-th mooring line has a stiffness defined by the Equation (4) in case of ropes stressed with tension force, and Equation (5) in case of compression condition. The relation depends by the Young Modulus (E), the cross-sectional area (A) and by the length of the ropes between the chock position and the bollard one (l).This stiffness is the one along the axis of the i-th rope.

$$k_i = \left(\frac{E(T)A}{l}\right)_i \quad (4)$$

$$k_i = 0 \quad (5)$$

The movement of the ship produced by the external forces is defined by the displacement vector **U**, with the its translational components (u_1,u_2,u_3) and with its angular components (θ_1, θ_2 e θ_3). This displacement of the center of gravity produces a stretching of the ropes along its own axis (6) and this deformation produces a tension (7) in each mooring line.

$$\boldsymbol{\delta l_i} = \boldsymbol{u} \cdot \boldsymbol{l_i} \quad (6)$$

$$\boldsymbol{\delta t_i} = k_i \cdot \boldsymbol{\delta l_i} \quad (7)$$

The i-th mooring line, in response to the displacement of the chock position (9-10), reacts with a force (11) in function of the projected stiffness defined by the Eq. (8).

$$[\boldsymbol{K}]_{\boldsymbol{i}} = k_i \begin{bmatrix} cos^2\phi_1 & cos\phi_1 cos\phi_2 & cos\phi_1 cos\phi_3 \\ cos\phi_1 cos\phi_2 & cos^2\phi_2 & cos\phi_2 cos\phi_3 \\ cos\phi_1 cos\phi_3 & cos\phi_2 cos\phi_3 & cos^2\phi_3 \end{bmatrix} \quad (8)$$

$$\boldsymbol{U}_{ci} = \boldsymbol{u} - [\boldsymbol{C}]_{\boldsymbol{i}}\boldsymbol{\theta} \quad (9)$$

$$[\boldsymbol{C}]_i = \begin{bmatrix} 0 & -C_3 & C_2 \\ C_3 & 0 & -C_1 \\ -C_2 & C_1 & 0 \end{bmatrix} \quad (10)$$

$$\boldsymbol{f_i} = [\boldsymbol{K}]_{\boldsymbol{i}} \cdot \boldsymbol{U}_{ci} = [\boldsymbol{K}]_{\boldsymbol{i}} \cdot \boldsymbol{u} - [\boldsymbol{K}]_{\boldsymbol{i}} \cdot [\boldsymbol{C}]_{\boldsymbol{i}}\boldsymbol{\theta} \quad (11)$$

Each reaction force produced by the mooring line is not applied in the center of gravity of the ship, so there are also moments (12) that depend by the position of the chock respect to the gravity of ship.

$$\boldsymbol{m_i} = [\boldsymbol{C}]_{\boldsymbol{i}} \cdot \boldsymbol{f_i} = [\boldsymbol{C}]_{\boldsymbol{i}} \cdot [\boldsymbol{K}]_{\boldsymbol{i}} \cdot \boldsymbol{u} - [\boldsymbol{C}]_{\boldsymbol{i}} \cdot [\boldsymbol{K}]_{\boldsymbol{i}} \cdot [\boldsymbol{C}]_{\boldsymbol{i}}\boldsymbol{\theta} \quad (12)$$

$$[\boldsymbol{A}]_{\boldsymbol{i}} = -[\boldsymbol{K}]_{\boldsymbol{i}} \cdot [\boldsymbol{C}]_{\boldsymbol{i}} \quad (13)$$

$$[\boldsymbol{B}]_{\boldsymbol{i}} = [\boldsymbol{C}]_{\boldsymbol{i}} \cdot [\boldsymbol{A}]_{\boldsymbol{i}} \quad (14)$$

Thanks to the simplification (13-14) is possible to resume the total effect of each mooring line (15).

$$\boldsymbol{F}_i = \begin{bmatrix} [K]_i & [A]_i \\ [K]_i^t & [B]_i \end{bmatrix} \cdot \boldsymbol{U} \quad (15)$$

The complete effects of all mooring lines can be summarized with a unique 6x6 matrix that is the sum of the single effects of each mooring line. This matrix assumes that the mooring lines are spring with a bilateral behavior so it is symmetric and it does not consider the unilateral behavior of the ropes.

$$[K_{Moor}] = \begin{bmatrix} \sum_{i=1}^{n}[K]_i & \sum_{i=1}^{n}[A]_i \\ \sum_{i=1}^{n}[K]_i^t & \sum_{i=1}^{n}[B]_i \end{bmatrix} \quad (16)$$

The fenders:

In addition to the mooring lines, the ship is constrained with fenders in order to avoid impact against the quay and so protecting the ship during the rest in port. The fender works when it is compressed and the reaction force (D_j) depend by the correspondent deflection (d_j) by means of the stiffness of the fender (k_j)

$$D_j = k_j d_j \quad (17)$$

$$[K_{fender}]_j = \begin{bmatrix} 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & -k_j & 0 & k_j r_{3j} & 0 & -k_j r_{1j} \\ 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & k_j r_{3j} & 0 & -k_j r_{3j}^2 & 0 & k_j r_{1j} r_{3j} \\ 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & -k_j r_{1j} & 0 & k_j r_{1j} r_{3j} & 0 & -k_j r_{1j}^2 \end{bmatrix} \quad (21)$$

The fender is comparable to a spring with a stiffness that depends by the characteristics of the material. This quantity is equal to zero if the fender is working in tension condition so $k_j = 0$ if the fender is in tension; $k_j = f_j$ if the fender is in compression (material characteristic). The columns 1, 3 and 5 are null because, for longitudinal and vertical translation and trim rotation, the fenders can not react.

The ship restoring forces:

The ship has the capacity to response to the displacement caused by external forces with restoring forces and moments. These quantities are dependent by hydrostatic characteristics of the ship and they work with a force reaction along axis X_3 and with two moments respect to the axis X_1 and X_2.

$$F_{REST3} = \rho_w g A_w u_3 \quad (18)$$

$$L_{REST1} = \Delta GM_T \theta_1 \quad (19)$$

$$L_{REST2} = \Delta GM_L \theta_2 \quad (20)$$

The effects of these restoring forces can be resumed with a ship stiffness matrix (21) and all reactions of the ship are bilateral.

$$[K_{Rest}] = \begin{bmatrix} 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & \rho_w g A_w & 0 & 0 & 0 \\ 0 & 0 & 0 & \Delta GM_T & 0 & 0 \\ 0 & 0 & 0 & 0 & \Delta GM_L & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 \end{bmatrix} \quad (21)$$

The mooring problem - External forces

There are several kinds of external actions that can cause displacement of the ship during the rest in port. There are forces caused by the weather, such as wind or currents or waves. There are forces caused by internal operations like moving cargo or loading and unloading of weights. Usually all these kinds of forces acting in a dynamic manner, but it is acceptable to consider the hypothesis of quasi-static condition and apply this forces in static way.

Wind and current force:

The force produced by the wind is dependent by the exposed surface of the ship and the velocity of the wind. The most accurate way to know this quantity is the fluid-dynamics simulation, but it takes a long time so there are empirical formulations that help to know this quantity with sufficient precision and in very little time.

$$F_{wind}(\eta) = \frac{1}{2}\rho_{air}C_w(\eta)V_w^2\,(A_{p1}cos^2(\eta) + (A_{p2}sin^2(\eta)) \quad (21)$$

The Eq. 21 helps to know the force of the wind in each direction of the wind. The exposed area is projected along the normal of the wind direction. The projection of this force along the longitudinal and transversal axis allows to know the external forces produced by the wind. These forces are not applied in the center of the gravity of the ship so they produce also moments. Usually the values of the C_w are between 0.4 and 1.2 for the case with wind along the longitudinal axis and between 0.6 and 1.4 for wind with transversal direction (the higher values are for tugs and trawlers, the lower values are for ferries).

In the same way is possible to calculate the force caused by the current (22). In this case the exposed surface is the immersed one. Also in this case there are moments produced by the force.

$$F_{current}(\varepsilon) = \frac{1}{2}\rho_{Water}C_C(\varepsilon)V_C^2\,(A_{c1}cos^2(\varepsilon) + (A_{C2}sin^2(\varepsilon)) \quad (22)$$

The drag coefficient C_C can vary between 0.1 and 0.4 for the longitudinal direction and between 0.7 and 1.0 for transversal direction in deep water. This coefficient is highly influenced by the depth of the water, in particular in shallow water, where the d/D ratio is less than 6, this coefficient can increase up to 6 times respect cases where the d/D is equal to 1.1.

Wave force

The wave load can be statically calculated thanks to the Eq. (23).

$$F_{wave}(\mu) = \rho_W C_m\,\mu\,g\,H\,D_0^2\,\frac{\pi^2}{8}\left(\frac{\sinh kD - \sinh k(D-d)}{\cosh kd}\right) \quad (23)$$

$$D_0 = (L_S - B)\sin\mu + \mathrm{B} \quad (24)$$

Generally C_m varies between 0.13 and 0.25 in longitudinal direction and between 1.45 and 1.75 in transversal direction.

A vector of external forces can be defined with a vector matrix (25) where there are the external forces components and the external moments components **Q = [F$_{ext}$l M$_{ext}$]**.

$$\boldsymbol{Q} = \begin{bmatrix} F_{wind1} + F_{current1} + F_{wave1} \\ F_{wind2} + F_{current2} + F_{wave2} \\ F_{tide3} + F_{load3} + F_{wave3} \\ M_{wind1} + M_{current1} + M_{wave1} + M_{load/unload1} \\ M_{wind2} + M_{current2} + M_{wave2} + M_{load/unload2} \\ M_{wind3} + M_{current3} + M_{wave3} \end{bmatrix} 25)$$

Unilaterality constraints problem

The problem of the unilaterality of the constraints must be taken in account in order to evaluate in correct way the reaction forces of the mooring lines and fenders. These two elements work only, respectively, in tension condition and compression condition. To solve this problem it is necessary, in the stiffness matrices of the mooring lines and fenders, to annul those terms related to a compression work of the mooring lines and those related to a pull work of the fender. The new stiffness matrices change from time to time in function of the external loads and they have an asymmetric shape.

Unilaterality of the mooring lines:

In order to understand if the generic mooring line is in compression condition we need to evaluate the sign of the external forces and moments. There is compression condition if the sign of the external forces are in agreement with the sign of the cosine director of the mooring line. In order to evaluate it, the fastest way is the sign of the product, element by element, between the **F**$_{ext}$ and the unit vector of the i-th mooring line **l**$_i$. If the sign is positive, it indicates that the rope is in compression condition and so it is not able to react in that direction.

In the case of the moment, it useful to add three bidimensional reference local frame for each rope. Each defined local reference frame is centered in the vessel center of gravity. The first step is the projection of the chock point and bollard point in each plane of the fixed reference system X_pX_q ($p = 1,2,3;\ q = 1,2,3;\ p \neq q$). With this projection is possible to define a local reference system $X'_p\,X'_q$. The axis of this local reference system are defined thanks to the angles δ_i (26) for the projection of the chock point and γ_i (27) for the bollard point (Figure 4).

$$cos\delta = \frac{C_q}{\sqrt{C_p^2 + C_q^2}} \quad sen\delta = \frac{C_p}{\sqrt{C_p^2 + C_q^2}} \quad (26)$$

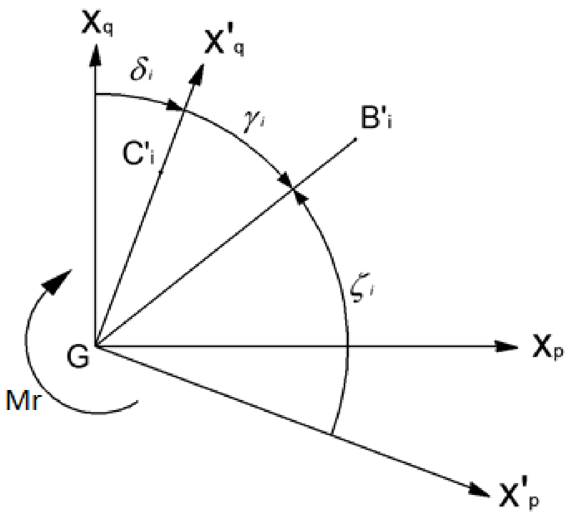

Figure 4. The moving and fixed bidimensional reference system

In the same figure it is also visible an external moment M_r ($r = 1,2,3;\ r \neq p \neq q$) that acts around the perpendicular axis to the plane and the angle ζ_i that is complementary to γ_i. Thanks to the Carnot theorem is possible to evaluate the angle between the projection of the chock point and the bollard point and so knowing the value of γ_i (27).

$$\gamma_i = \arccos \frac{\overline{GB_i^2} + \overline{GC_i^2} - \overline{C_i B_i^2}}{2\overline{GB_i GC_i}} \qquad (27)$$

In consequence of a rotation of the ship as a result of an external moment M_r, the angle γ_i will change, in the case of small displacements of a quantity $\delta\gamma_i$ (29) and it will become equal to γ_i' (28)

$$\gamma' = \gamma - \delta\gamma \qquad (28)$$

And the

$$\delta\gamma_i = \Psi \frac{M_r}{|M_r|} \cdot \frac{\cos\zeta_i}{|\cos\zeta_i|} \qquad (29)$$

The Ψ is an arbitrarily small positive quantity. The only important thing of this small angle displacement is the sign. This sign is influenced by the plane of projection with the angle ζ_i and by the sign of the moment. If the $\delta\gamma > 0$, the mooring line is not in tension and it does not work. If instead $\delta\gamma < 0$ then the mooring line works and reacts with a force. In order to take account of this process, in the matrix of the chock points the one that does not work is deleted and the matrix of stiffness is recalculated.

Unilaterality of the fenders

A linearization of the problem it is necessary in order to take account about the unilaterality of the fenders. The evaluation of the unilaterality can be made with the use of a virtual displacement of the ship as if it was constrained solely by the ropes. In this case is possible to find a virtual displacement vector **U'** (30)

$$\boldsymbol{U'} = [\boldsymbol{K_{moor}}]^{-1}\boldsymbol{Q} \qquad (30)$$

Because the matrix $[K_{fender}]_j$, related to the j-th fender, has only the columns 2, 4 and 6 non-zero, this columns are that ones have to be verified, in particular, exploiting the sign of the elements:

- the column 2 must be canceled if $u_2' R_{2j} < 0$;
- the column 4 must be canceled if $\theta_1' R_{2j} R_{3j} > 0$;
- the column 6 must be canceled if $\theta_3' R_{1j} R_{2j} < 0$;

Consequently, it is possible to build the matrix

$$[K_{fender}] = \sum_j [K_{fender}]_j$$

SOLUTION OF THE PROBLEM

The problem involves an algebraic system of equations. A global stiffness (31) matrix is obtained, with relative corrections in order to take in account the unilaterality characteristics of mooring lines and fenders.

$$[\boldsymbol{K}] = [K_{moor}] + [K_{feneder}] + [K_{Rest}] \qquad (31)$$

An external vector force is defined thanks to empirical formulations. The solutions of the problem is obtained with the relation in Eq. (32) in terms of displacements.

$$\boldsymbol{U} = [\boldsymbol{K}]^{-1}\boldsymbol{Q} \qquad (32)$$

With the Eq. (11) is possible to know the force on the i-th mooring line. In order to verify the is with the comparison of the i-th force with the maximum breaking load of the rope MBL_i multiplied for a coefficient (χ) that takes in account the limit of the breaks of the winches (generally calibrated to 70% of the MBL_i). This condition on the winches allows working always under the hypothesis of small displacement.

$$|f_i| \leq MBL_i\,\chi \qquad (33)$$

With the Eq. (17) it is possible to know the force on the j-th fender. Also in this case, the safe condition is evaluable with the condition defined above. When the force exceed the value of breakage, the hypothesis of small displacement will be no more valid.

The method here presented can be resumed with a flow-chart shown in Figure 5.

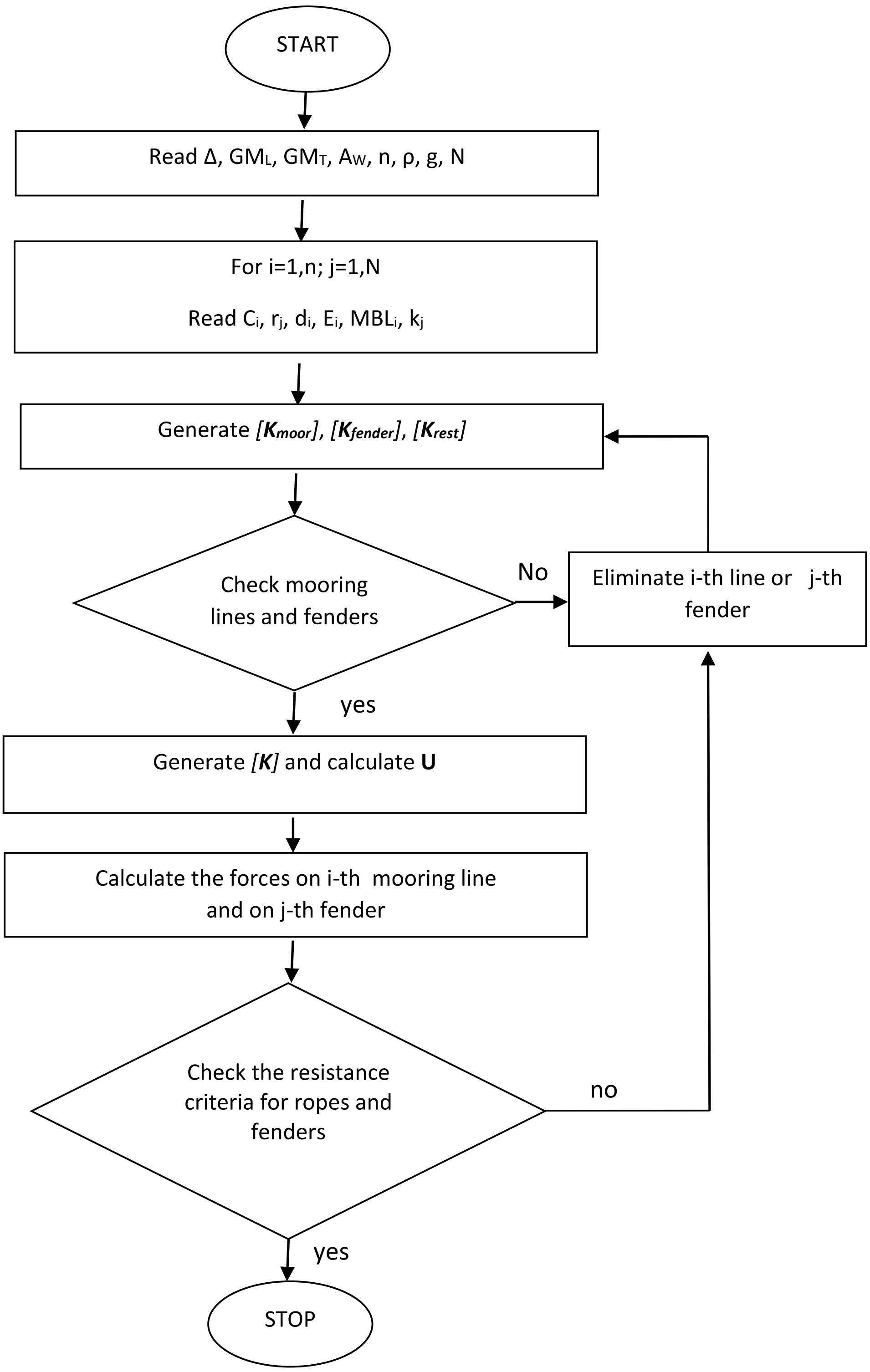

Figure 5. Flow chart of the entire process

CONCLUSIONS

The presented method is simple to implement and it can be used for solving the problem of mooring of a berthed ship. The works is based on literature but new improvement has been added in order to take in account the unilaterality of the constraints. The method involves different aspects such as evaluation of geometrical characteristics of the mooring elements (ropes and fenders), the evaluation with empirical formula of the external forces and the evaluation of the tensional state of the ropes and fenders. Each force is applied during the process with a quasi-static approach and the entire solution process is based on the hypothesis of small displacements. An important evolution of the method respect to precedent papers is the evaluation of the unilaterality behavior of the constrains, both for the mooring lines and fenders. For each condition, a global stiffness matrix is constructed ad-hoc and in most of the cases this matrix is not symmetric. Thanks to this approach it is possible to simulate many external conditions in order to find the worst condition for the integrity of the ropes and of the fenders. It is also possible to construct the polar curves that indicate, in function of the direction and intensity of the wind, the moment in which the ropes reach the breaking force. The simulation run very fast and in few seconds is possible to have a complete panoramic of the all mooring conditions.

REFERENCE

[1] Schelfn, T.E., Östergaard, C.: The vessel in port: Mooring problems. Mar. Struct. 8, 451–479 (1995).

[2] Chernjawski, P.E.: Mooring of surface vessels to piers. In: Society of Naval Architects and Marine Engineers (ed.) Hampton Roads SNAME Meeting. p. 27. , Jersey City (1978).

[3] Natarajan, R., Ganapathy, C.: Analysis of moorings of a berthed ship. Mar. Struct. 8, 481–499 (1995).

[4] Cella, U., Cucinotta, F., Sfravara, F.: Sail Plan Parametric CAD Model for an A-Class Catamaran Numerical Optimization Procedure Using Open Source Tools. Presented at the (2017).

[5] Cucinotta, F., Nigrelli, V., Sfravara, F.: Numerical prediction of ventilated planing flat plates for the design of Air Cavity Ships. Int. J. Interact. Des. Manuf. (2017).

[6] Brusca, S., Cucinotta, F., Galvagno, A., Lanzafame, R., Mauro, S.: Oscillating water column wave energy converter by means of straight-bladed Darrieus turbine. Energy Procedia. 1–10 (2015).

[7] Cucinotta, F., Nigrelli, V., Sfravara, F.: A preliminary method for the numerical prediction of the behavior of air bubbles in the design of Air Cavity Ships. Presented at the (2017).

[8] Cucinotta, F., Guglielmino, E., Risitano, G., Sfravara, F.: Assessment of Damage Evolution in Sandwich Composite Material Subjected to Repeated Impacts by Means Optical Measurements. Procedia Struct. Integr. 2, 3660–3667 (2016).

[9] Chen, X.B., Molin, B., Petitjean, F.: Numerical evaluation of the springing loads on tension leg platforms. Mar. Struct. 8, 501–524 (1995).

[10] Wu, S.: Adaptive dynamic relaxation technique for static analysis of catenary mooring. Mar. Struct. 8, 585–599 (1995).

[11] Krishnankutty, P., Vendhan, C.P.: Three-dimensional finite element analysis of the diffraction-radiation problem of hydrodynamically compact structures. Mar. Struct. 8, 525–542 (1995).

[12] Patel, M.H., Park, H.I.: Tensioned buoyant platform tether response to short duration tension loss. Mar. Struct. 8, 543–553 (1995).

[13] Hann, M.: Statics and dynamics of multi-cable systems for submersibles. Mar. Struct. 8, 555–583 (1995).

[14] Cucinotta, F., Guglielmino, E., Sfravara, F.: Frequency of Ship Collisions in the Strait of Messina through Regulatory and Environmental Constraints Assessment. J. Navig. 1–21 (2017).

[15] Cucinotta, F., Guglielmino, E., Sfravara, F.: Life cycle assessment in yacht industry: A case study of comparison between hand lay-up and vacuum infusion. J. Clean. Prod. 142, 3822–3833 (2017).

[16] Barone, S., Cucinotta, F., Sfravara, F.: A comparative Life Cycle Assessment of utility poles manufactured with different materials and dimensions. In: Advances on Mechanics, Design Engineering and Manufacturing. pp. 91–99. Springer International Publishing (2017).