ANALYTICAL ESTIMATION OF MANEUVERABILITY OF MOORED FPU WITH INTERNAL TURRET IN CLOSE ICE

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ABSTRACT
The prospective use of floating structures for offshore oil and gas field developments in areas with harsh ice regime demands finding ways for ice loads decreasing. Among possible approaches, Floating Production Units (FPU) of ship-shape type are considered, including variants with non-disconnectable internal turret. The concept suggests the ability of the FPU to perform rather quick turning on the spot in close ice in order to take a position against current or anticipated ice drift direction. The problem involves a number of parameters of high indefiniteness. In the paper, the process of FPU turning on spot is analyzed within relatively simple engineering models based on consideration of mass, momentum and energy balances. Within the model, an assessment is performed of the mechanical work that is required for ice breaking and further ice piling up near moving FPU hull. Certain conclusions about the total thruster power on the ship can be derived on this basis. Conservative estimates demonstrate that the active turn of an FPU on 90 degrees at reasonable angular velocity in medium first-year level ice would require the total power being well above values now available. Another model considers an FPU turning on the spot in broken compact ice. Comparison with the case of passive turning of FPU under the action of drifting ice demonstrates that the overall design of the FPU mooring system can be inadequate when the design ice loads are addressed only for the case of FPU heading against ice drift direction.

INTRODUCTION
In connection with the prospects of development of deposits of hydrocarbons in the Arctic and in other icy seas, located at depths that preclude the use of fixed platforms (at least at the level of modern technology), on the agenda are submitted technical feasibility of the application of floating structures of FPU type. One of the main issues here is a matter of stationkeeping (see, e.g., Bonnemaire et al. 2007, Riska and Coche, 2013 and references inside). Drift patterns of ice may influence the level of ice loads, and sudden changes in the ice drift may lead to high action situations and overloading of the mooring system.

One of the tasks that require thorough research, is the ability of the moored ship to perform active and/or passive turning on the spot (ice vaning property) in conditions of variability in ice drift direction. It is known that in the Arctic seas ice can change its direction very quickly and abruptly. Among relevant examples is frequent changes in the ice drift direction in the eastern Pechora Sea where the changes in ice drift direction were identified about 1.5 times per day at least of 135º in less than 15 min (Bonnemaire et al., 2007). Ice vaning, orienting the bow of the drillship or FPU into ice drift in ice conditions, is similar to weathervaning, however, ice is a much more damping medium.

The use of Dynamic Positioning (DP) is a promising stationkeeping technology for Arctic offshore operations. In recent years a number of time-consuming research in the ice tanks were implemented in order to estimate the possibilities and limitations of DP assistance in ice conditions, corresponding numerical studies were performed as well (see, e.g., Haase 2012, Metrikin et al. 2013, Neville, 2013). It is known that DP has a limited capacity in heavy ice. Riska and Coche (2013) noted that the limiting ice thickness beyond which a DP system cannot perform depends on the ship particulars, propulsion system and also on the control system. They set a limit for ice thickness when active turning is possible somewhere in range 50…100 cm, even in managed ice.

Therefore, the task of constructing a model for the analysis of passive turning in heavy ice remains relevant. It appears that particular danger situations are those accompanied by compression of ice, because in such conditions the ice clearance in the vicinity of the sides of the FPU is significantly hampered. The problem of the passive turning also studied in ice tankes and in the framework of numerical simulation (see, e.g. Aksnes and Bonnemaire, 2009, Kovalyov et al. 2013, Tsarau ye al. 2013).

Note that the critical parameter here is the maximum value of the mooring force, realized during the process of turning. In the problem there are several important parameters that influence, to varying degrees, on this value. Among key parameters are the ice load, velocity of ice drift, the stiffness of the mooring system, FPU hull characteristics, including the location of the internal turret. The complexity of the experiments in the ice tanks does not allow for the full parametric analysis of this problem. The same applies to modeling using the discrete element method for describing the ice cover (Aksnes, 2011, Lubbad and Løset, 2011).

The aim of this work is to construct an engineering model, in which the ice load is modeled in a relatively simple way that makes it possible to perform an efficient parametric analysis of the problem of passive turning of a large size FPU in the conditions of drifting ice.

STATEMENT OF THE PROBLEM

The problem on the turn of FPU under the action of drift ice and FPU thrusters is considered (Fig. 1). FPU is equipped by internal turret displaced to the FPU bow shown by point S in Fig. 1. It is assumed that surrounded ice drifts with speed V_0 in the direction of the axis Y of fixed frame reference OXY, where the origin O coincides with initial location of the center of gravity of the ship. Ice action influences displacement and rotation of the FPU hull due to the reaction of mooring lines mounted to the turret.

In the present paper we consider only FPU displacements in the horizontal plane (X,Y) and FPU rotation around the vertical axis Z. Other movements of FPU (heave, pitch and roll) are assumed small and therefore ignored. Actual position of FPU is determined by the coordinates X_G and Y_G of the FPU gravity center G and by the heading angle φ between the axis X and the FPU axis (Fig. 1). Positive angles φ are associated with counter clockwise rotation of the FPU hull.

ACTIVE TURNING OF FPU

Let us consider a problem on active turn of FPU around the gravity centre G in ice conditions under the action of two Azipode thrusters located at the distance L from the point G (Fig. 1b). Point G is located in the middle of FPU. Azipods are azimuthing (steering around their vertical axis) propulsors, which make possible to create the thrusts oriented at the arbitrary angle with respect to the ship centreline. The trust directions are shown by vectors Q_1 and Q_2 in Fig. 1b for the bow and stern of the FPU respectively. Further it is assumed that $Q_1 = Q_2 = Q_0$, and torsion moment $\mathrm{Mom}_0 = 2LQ_0$ creates the rotation of the FPU on the spot.

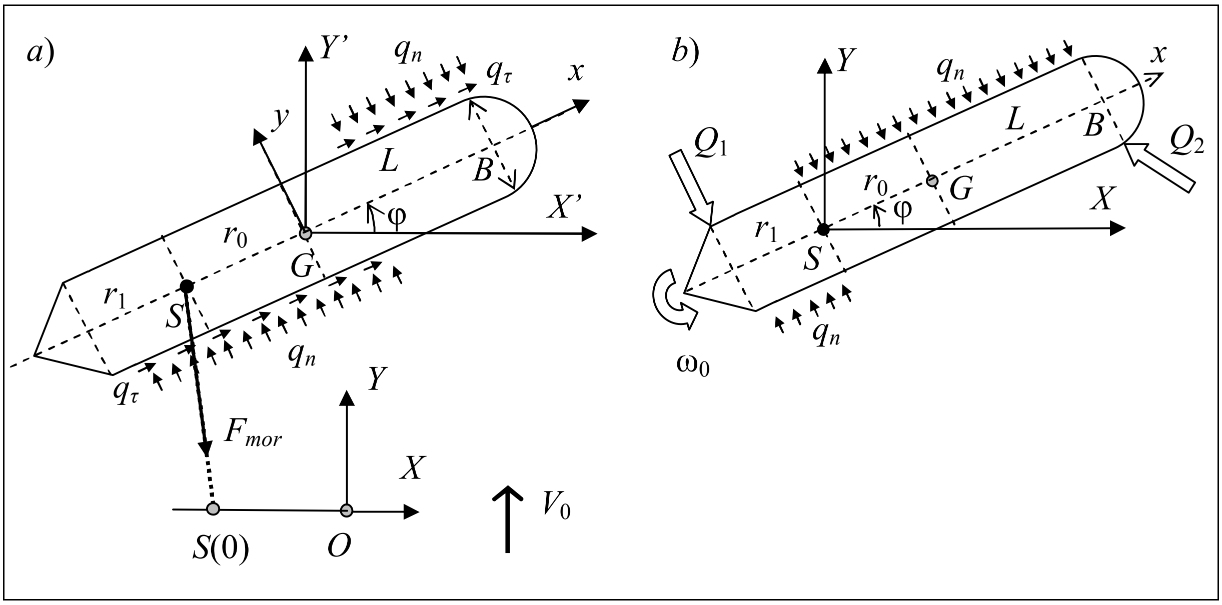

Figure 1 – Calculation scheme for moored FPU passive turning in drifting ice and active turning in static ice

The ice resistance to the displacements of the FPU hull is performed by a combination of normal ice stress q_n and shear ice stress $q_\tau = \mu q_n$, where μ is the coefficient of friction between ice and steel ($\mu = 0.1 \ldots 0.15$). Stresses q_n and q_t are integrated over the ice thickness and therefore their dimension is N/m. In case of uniform distribution of the ice thickness along an FPU side we may express the normal stress as a product $q_n = p_n h$ where p_n is the ice pressure and h is the ice thickness.

According to the information from ABB Marine (ABB, 2010) propeller power – bollard pull thrust diagram for the different Azipod frame sizes is almost linear and for study purposes can be approximated as

$$Q \approx 10P \qquad (1)$$

where Q is an ultimate thrust of one Azipod module expressed in MN, while P is its available power expressed in MW, with limiting power of one module up to 21 MW. Note, that there are various estimates for the matter, e.g., Riska and Coche (2013) mention somewhat more favorite relationship $Q \approx 5P$.

Further we estimate necessary power for the FPU turning on the spot for two scenarios of ice interaction with the FPU hull.

Case of constant ice action on the ship hull (Scheme 1)

Suppose ice breaks near FPU sides by crushing with cleaning of the hull by water flow. The FPU turns with constant angular velocity ω, and normal ice stress q_n (e.g. crushing strength) is a constant. Shear stress q_t is ignored.

The work of normal ice force acting on the length dx of the FPU hull is determined by the formula $(dx) = q_n dx \delta u$, where δu is normal displacement of the FPU hull during the time δt. Total work of the normal force applied to the FPU hull is calculated by the formula

$$\delta A = 2 \int_0^L q_n \delta u dx, \qquad (2)$$

where $\delta u = x\omega_0 \delta t$. Since $W = \delta A / \delta t$, we find $W = q_n \omega_0 L^2$. Angular velocity is estimated from the assumption that FPU turning on 90° is realized in 25 min, i.e. $\omega_0 = 10^{-3}$ rad/s. In this case it follows $W = 22.5 q_n$, when L=150 m.

Required power reaches 22.5 MW when $q_n \sim 1$ MN. Resulting angular moment applied to the FPU by ice is

$$\text{Mom}_{ice} = q_n L^2 = 2.25 \cdot 10^4 \, q_n \,, \qquad (3)$$

while torsion moment created by Asipods is equal to

$$\text{Mom}_{thrust} = 2LQ_0 = 300Q_0. \qquad (4)$$

Thus it is clear that in case of Asipod propulsion $Q_0 \sim 1$ MN, the ice resistance is much greater than propulsion action even for $q_n = 0.1$ MN/m. From above performed estimates it follows that more or less efficient FPU turning on the spot in active mode is available in very weak ice cover when normal ice stress applied to the FPU hull is about $q_n \sim 0.01$ MN/m .

Case of increasing ice action on the ship hull (Scheme 2)

The turning of FPU is accompanied by ice rubble pile building up near the hull (Fig. 2a). Ice piling up near the hull of offshore structures is observed for relatively small slope angles of the hull (angle α in Fig. 2a) as well. For example, Wright (1999) and Neville et al. (2013) noted in application to drilling barge Kulluk that situations with pressure in the ice cover provoked occasionally the creation of accumulation of ice upfront of Kulluk and that these events created the highest loads on Kulluk. This scenario was also observed in model experiments with floating structure by Aksnes and Bonnemaire (2009). It was noted by Palmer and Croasdale (2013) that ice accumulation can result even in managed ice in ice pressure scenarios.

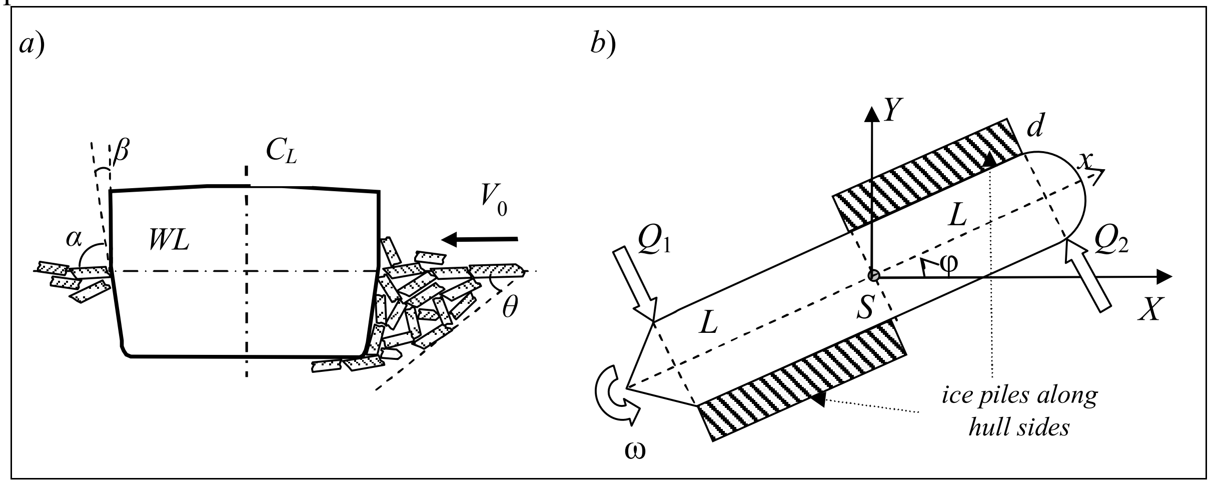

Figure 2. Accumulation of ice near the ship hull

During FPU turning, especially in close ice condition, ice rubble piles form from both sides of the hull (Fig. 2b). To estimate the values of ice loads due to the ice rubble accumulation, one can use the results of the study performed by Marchenko and Makshtas (2005) for ice ridge building up process between two compressed floes. It was shown that the load per unit length of the ridge can be accessed through the following formula:

$$q_n = 2\rho_{ice} g \mu_{ii} U (1 + \kappa^2)^{-1} \qquad (5)$$

where ρ_{ice} is the density of ice ($\rho_{ice} \approx 920$ kg/m^3), g is gravity acceleration, μ_{ii} is a coefficient of ice-to-ice friction ($\mu_{ii} \approx 0.3$), U is a specific volume (per unit length) of the ice rubble pile, and κ is a ratio of the pile draft to the pile height. The value $\kappa \approx 4$ is used for ice ridges based on the in-situ observations when ridge sails and keels are in hydrostatic equilibrium. Formula (5) has been derived for ice ridge build up scenario when the ice ridge growth is provided by the pushing of a train of ice blocks through the ridge at the water line level. In the present paper it is assumed that a train of ice blocks is pushed through the rubble pile up to the FPU side. Formula (5) specifies total friction between the ice blocks and the sail and the keel of the rubble pile. Breaking of the hydrostatic equilibrium between rubble sail and keel can influence the increase of the load (Marchenko, 2010).

In case of FPU turning, the situation is not uniform along the hull side. The amount of ice feeding the ice pile at point x (measured from the rotation axis), obviously depends on the length of the arc covered by point x during the turning, and $U(x) = hx\varphi$. Hence

$$q(x, \varphi(t)) = 2\rho_{ice} g\mu h x \varphi(t)(1+\kappa^2)^{-1}, \quad (6)$$

where $\varphi(t)$ is equal to the current turning angle of the ship taken from the initial position. When FPU turns around the gravity center G the torsion moment is calculated by the formula

$$\mathrm{Mom}(\varphi(t)) = 2\int_0^L q_n(x, \varphi(t)) x dx = \frac{4}{3}\rho_{ice} g\mu h L^3 (1+\kappa^2)^{-1}\varphi(t). \quad (7)$$

We find $\mathrm{Mom}(\pi/4) = 562h$ (MN · m) when $\varphi = \pi/4$ and L=150 m. Note, that for $h \sim 1$ m this value exceeds Mom_{thrust} determined by formula (4). It means that active turning of FPU would be almost stopped rather quickly. At the same time the linear load is quite moderate at this instant of time: $q(L; \pi/4) \sim 0.03\, h$ MN/m (h is taken here and further in meters), which corresponds to the not fully developed pile.

It is assumed that maximal compression force is reached when the draft of the ice rubble pile becomes equal to the FPU draft. The specific volume of the pile is estimated by the formula $U \approx h_k^2/(2\tan\theta)$, where θ is the slope angle of the rubble pile and h_k is the keel depth. Assuming $\theta \approx 30°$ and $h_k \approx 15$ m, we find $U \approx 195$ m^2. It follows from formula (5) that maximal compression per unit length of the FPU hull due to ice pile building up can be estimated as $q_{ridge} = 0.062$ MN/m.

For comparison, it can be mentioned that for the Kulluk drilling barge, which draft equals T = 11 m and diameter at the waterline equals D = 70 m, field data presented by Wright (2000) give estimates for peak measured ice loads in situations with poor clearance of managed ice of thickness up to 2 m as

$$F = 0.87h + 0.91 \ \text{(MN)}, \quad (8)$$

which results in values for linear loads as

$$q \sim 0.012h + 0.013 \ \text{(MN/m)}. \quad (9)$$

At the same time, results of the numerical study of possible ice loads on the Kulluk obtained by Lawrence (2009) with the help of discrete elements modeling give larger values 0.11 MN/m, 0.24 MN/m and 0.44 MN/m for ice thickness h = 1.0, 1.5, 2.0 m, respectively.

Those were scenarios of level ice impacting the structure that were studied, so, certain parts of the loads are due to level ice bending failure. The corresponding value at first approximation can be estimated through the following 2-D theory equation (Croasdale, 1980)

$$F_b = \frac{\sigma_f h^2 e^{\pi/4}}{6\ell}\tan(\alpha + \psi) \quad (10)$$

where σ_f is flexural strength of ice, α is the slope angle of a structure face to horizon at the sea surface (note that in the case of the ship hull value $(\pi/2 - \alpha)$ corresponds to frame (or flare) angle β; see Fig. 2a), $\psi = \mathrm{atan}(\mu)$, and ℓ is the characteristic length equal to

$$\ell = \left(\frac{E}{12(1-\nu^2)\rho_w g h}\right)^{1/4} h$$

where E and ν are elastic modulus and Poisson ratio of ice, and ρ_w is the water density. It can be noted that a refined approach with more accurate accounting for axial force in the ice beam, which was presented by Goldstein et al. (2005), gives somewhat higher values of F_b for thick ice, up to 30 % for h = 2 m.

Adopting the following parameters $\sigma_f = 0.5$ MPa, E = 3 GPa, ν = 0.3 and $\rho_w = 1000$ kg/m^3 one can get

$$F_b \approx 0.0142h^{5/4}\tan(\alpha + \psi), \quad (11)$$

where the force is measured in MN and h is taken in meters.

In Table 1 results from different sources are presented that relates to the ice loads on wide slope structures. Note the proximity of values in the fourth and fifth columns. This allows us to consider the values in the third column as the load due to ice rubble formation before the Kulluk face. These quantities are much lower than those in columns 6, 7 and 8, however larger that value $q_n \sim 0.01$ MN/m, which we adopted as a limit for which efficient active ship turning was possible. Based on the rough analyses performed, it seems the quantities of $q_n \sim 0.1 \ldots 0.3$ MN/m can be considered as quite possible for case of long ship in close ice including the scenario of compressed broken ice.

Table 1. Horizontal force (per meter) acting on a slope face (α ≈ 30º), MN/m

h, m	Kulluk, field measurements[1]			ice failure, bending	Kulluk, numerical modeling [2]	Sevan-type concept[3]	ridge building[4]
	unbroken ice	managed ice, poor clearance	the difference	μ = 0.1/0.15			
1	*2*	*3*	*4*	*5*	*6*	*7*	*8*
1.0	0.034	0.025	0.009	0.011/0.012	0.086/0.069	-	0.062
1.5	0.049	0.032	0.017	0.018/0.020	0.19/0.14	-	0.062
2.0	0.064	0.038	0.026	0.025/0.028	0.32/0.23[5]	0.20	0.062

[1] *Wright (2000); restoring force of mooring system is shown*
[2] *Lawrence (2009); loads on the hull / mooring force are shown for level ice case*
[3] *Løset and Aarsnes (2009), model-scale testing for level ice with h=1.9 m*
[4] *Marchenko and Makshtas (2005), ultimate values for ridge building are shown*
[5] *extrapolated values*

PASSIVE ICE VEINING

Basic equations

Equations of slender body theory (see, e.g., Kornev, 2013) are further used for the modelling of FPU manoeuvring in ice conditions. Equations are written down in the ship-fixed reference system *Gxyz* with axes x and y lying in a horizontal plane and vertical axis z with positive direction upward. The x axis is the longitudinal coordinate, positive astern, the axis y is the transverse coordinate, positive to the starboard side (note the difference with the traditional orientation of axes). The origin is in the plane of symmetry, referred usually as centreline. The vertical location of the origin lies at the level of the undisturbed free surface when the ship is at the rest.

Ship motion equations in the ship-fixed reference system are written as follows

$$\begin{cases} (m + m_{11})\frac{dV_x}{dt} - (m + m_{22})V_y\omega = F_x \\ (m + m_{22})\frac{dV_y}{dt} + (m + m_{11})V_x\omega = F_y \\ (I_{zz} + m_{66})\frac{d\omega}{dt} + V_xV_y(m_{22} - m_{11})/2 = M_z \end{cases} \quad (12)$$

$$
\begin{cases}
\frac{d\varphi}{dt} = \omega \\
\frac{dX_G}{dt} = V_x \cos\varphi - V_y \sin\varphi \\
\frac{dY_G}{dt} = V_x \sin\varphi + V_y \cos\varphi
\end{cases}
\qquad (13)
$$

where m is total FPU mass, m_{11} and m_{22} are the added masses in the longitudinal and transversal directions relatively the FPU centerline, I_{zz} is polar moment of inertia of FPU relatively the vertical axis, m_{66} is the added polar momentum of inertia of FPU relatively the vertical axis, V_x and V_y are the projections of the velocity of FPU gravity center (point G in Fig. 1) on the axes x and y, X_G and Y_G are the coordinates of FPU gravity center in the fixed frame of reference (X,Y), ω is the angular velocity of FPU relatively the gravity center, F_x and F_y are the projections of total external force applied to FPU hull on the axes x and y, M_z is the total torsion moment of external forces, and t is the time. Expressions for the forces F_x, F_y and moment M_z are specified in the Appendix.

Results of numerical simulations

Numerical simulations were performed for the modelling of passive turn of FPU under the action of drifting ice. Geometrical and mass characteristics of FPU are specified as follows

$$L = 150 \text{ m},\ B = 50 \text{ m},\ T = 15 \text{ m},\ r_0 = 90 \text{ m},$$

$$M = 200 \cdot 10^6 \text{ kg},\ I_{zz} = 1 \cdot 10^{12} \text{ kg·m}^2,\ m_{11} = 0{,}05M, m_{22} = M, m_{66} = 0{,}85 I_{zz}$$

(added masses are taken in accordence with general ship theory; see, e.g. Korotkin, 2009).
The ice action is specified by the following values

$$q_n = 0.3 \text{ MPa},\ q_{n0} = 0.03 \text{ MPa},\ \mu = 0.1,\ V_0 = 0.5 \text{ m/s}.$$

Note that in order to smooth partly the effect of the abrupt application of ice action, we use at the initial stage of the process a gradual growth of ice pressure:

$$\tilde{q}_n(t) = \begin{cases} (t/t_0) q_n, & t \le t_0 \\ q_n, & t > t_0 \end{cases},\quad \tilde{q}_{n0}(t) = \begin{cases} (t/t_0) q_{n0}, & t \le t_0 \\ q_{n0}, & t > t_0 \end{cases}$$

where t_0 is chosen about 2-10 minutes.
The reaction of mooring lines and riser system is specified by the coefficient $k_{\text{mor}} = 1.2 \text{ MN/m}$, limit mooring force 60 MN and limit offset 50 m (or, equivalently, 40 MN for limit offset 33 m). The initial values taken for the governing system (12) and (13) correspond to the motionless state of the FPU at the position with centreline being perpendicular to ice drift direction (Fig. 3), which is given by following values for $t = 0$:

$$X_G = 0,\ Y_G = 0,\ \varphi = 0,\ V_x = 0,\ V_y = 0,\ \omega = 0.$$

In Figure 3, solid red curve is the trace of successive positions of center of gravity G, while dot blue curve shows that for the turret point S.
Figure 4 displays snapshots of a ship location (shown as a blue bar) at the successive instants. Following are the characteristic features of the ship motion in the process of its passive turning. At the first phase (up till about $t = 2$ min) a scenario can be seen in which the ship is caught by the flow of ice and shifted almost without rotation (Fig. 3a, 3b), despite the emergence of positive (counter clockwise) twisting moment from the mooring forces. An explanation of this pattern is due to the fact that when the ship is attempting to start the turning, the active influence of ice on the stern from the starboard side and on the bow from the starboard is arising (something like that shown in Fig. 1a).

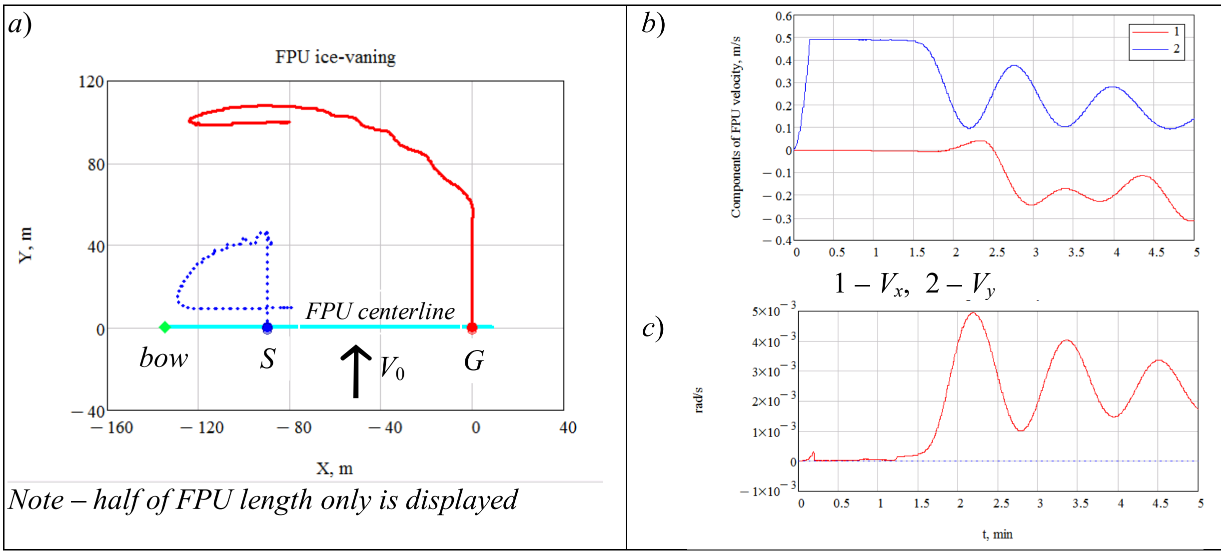

Figure 3. Initial position and traces (a), linear velocities (b) of point *G*, and angular velocity (c) of the FPU during passive turning

Emerging negative moment (directed clockwise in this case) balances at this phase oppositely directed torque forces from the mooring system and effectively prevents rotation of the ship. The total ice load on the hull of the ship is approximately equal during this phase to the mooring force and oppositely directed. Note that this effect is mainly a result of the assumption of tight contact of both sides of the ship with ice, which seems quite reasonable for the scenario with a close ice under compression.
Then there comes an instant when the balance of forces, sustained in the first phase, disrupted: begins the phase of rotation of the ship (t = 2.5 to 5 min). When this occurs the ice begins to move along the sides of the ship, developing the friction that generates an additional load component favourable for rotation. The ship start a combined motion with simultaneous turning and displacement of the centre of gravity (point *G*) in transverse to the ice drift direction. The rotation is fairly slow – the ship is rotated at 90° about a half-hour (t = 30 min). After that the ship retains the orientation against the ice drift performing, for a long time, a slow side drift to the equilibrium. The residual motion is maintained even at t = 90 min. A low rate of passive turning was observed earlier in model experiments. For example, Aksnes and Bonnemaire (2009) observed that a ship 120 m long made the turn through 90° about a half-hour (the stiffness of the mooring system was 2.2 MN/m).

Note that for modelling in tank a scheme of reversed motion was applied (when a ship model is forcibly pulled through the ice). In this approach the external (with respect to the incoming ice cover) side of the ship actually stays free from contact with ice. Therefore, in the scheme of reversed motion the ship is subjected to less turning resistance as compared to the case of direct motion. In the scenario of the action of close drifting ice, which is considered in the present work, the ice acts alternately on both sides of the ship, facilitating effective damping, and even blocking, its turning.

The maximum value of the restoring force is 57.0 MN, and it was reached approximately at the beginning of phase of active tutrning (t ≈ 2 min). For comparison, under the assumption on the absence of rotation of the hull in drifting ice, the proposed model predicts a static value for the restoring forces as 76.5 MN, while in the dynamic calculation it reaches 87.7 MN.

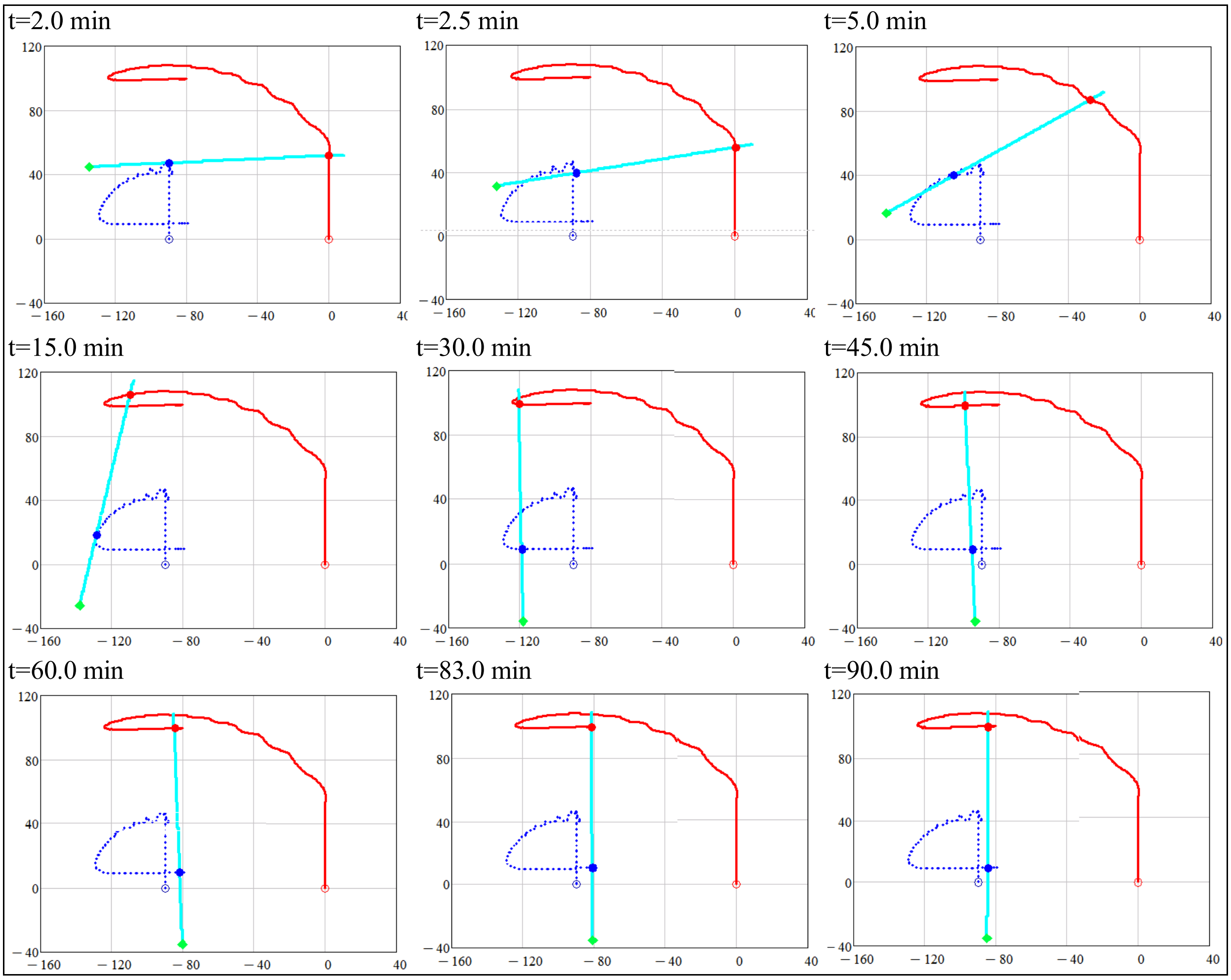

Figure 4. Successive positions of an FPU performing passive turning on the spot under drifting close ice action (t_0 = 2 min)

It should be noted that parametric calculations performed using the presented model, in which the value of q_n is changed in the range 0.1...0.5 MN/m, V_0 lies in the range 0.1...0.5 m/s, k_{mor} takes values 1.2, 2.0 and 4.0 MN/m, showed that in most cases the ratio of the maximum restoring force in the scenario of a passive ship turn to a static value for non-rotating ship, when exposed to ice perpendicular to the side, is not less than 0.7.

In some cases, that ratio was even higher than 1 (the most typical case corresponds to a relatively low value q_n = 0.1 MN/m and a high value of the velocity of ice drift V_0 = 0.5 m/s, when the ratio reached the value of 1.15). However, no clearly expressed tendencies was found at present.

It has been checked that a specific value for t_0 within the range 1…10 min has rather small influence on the calculation results, with the exception of very slow ice drift. The general trend is a moderate decreasing of the maximal value of the mooring force as t_0 is increasing. The proper choose of t_0 is somewhat unclear due to potentially rather quick ice load growth even under small velocities of ice drift.

CONCLUSIONS

Sudden appearence of ice drift from the rest state influences the trapping of FPU by the ice during 5-10 min. During this time FPU displaces together with ice in translational manner without rotation. FPU rotation is blocked by the ice pressure from the sides of the FPU. FPU begin to turn when the reaction of mooring lines and riser system reaches relatively high value. Presented model of passive ship turning on the spot under the influence of drifting ice utilizes an approach in which the ice acting on the ship is described by a continuous model, unlike models where the ice is treated with discrete elements modeling. While constructing the model, an attempt is made to consider the conditions of consolidated ice in compression conditions, when during the rotation of a ship the appearance of free water space around its sides is not obvious.

According to the results of a series of parametric calculations, in which the parameters of the ship were fixed, including the location of the turret, the following conclusions can be made. In conditions of compressed ice, the assistance of thrusters when making a turn on the spot can be efficient only in relatively weak ice that produces the pressure on the ship hull of the order of 0.01 MN/m.

When ice drift begins against the side of a ship, first, there is an event of "capture" of the ship by ice flow, and a rather long phase is observed (2-10 min depending on the rate of ice drift) of translation motion of the ship in the direction of ice drift with no pronounced rotation.

The apparent turning starts only after reaching some critical value for the restoring force, which greatly exceeds the ice resistance of the ship, when the ship bow is oriented against the ice drift, and ranges from 70 to 100% of the static value corresponding to ice action on the ship in non-rotating case.

ACKNOWLEDGMENT

The authors wish to acknowledge the support of the Research Council of Norway through the project 243930 (FIMA) and the Russian Foundation for Basic Research through the project 15-55-20003 Norv_a.

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APPENDIX

External force is represented by a sum of the ice load $\mathbf{F}_{ice}$ applied to the FPU hull, the reaction from mooring lines $\mathbf{F}_{mor}$ applied to the turret (point S in Fig. 1), and the hydrodynamic resistance $\mathbf{FD}$ applied to the FPU hull. Projections of $\mathbf{F}_{ice}$, $\mathbf{F}_{mor}$ and $\mathbf{FD}$ on the axes x and y are determined by the formulas

$$F_x = F_{ice,x} + F_{mor,x} + FD_x, \quad F_y = F_{ice,y} + F_{mor,y} + FD_y, \tag{A1}$$

$$F_{ice,x} = [q_\tau(2L - B) + q_{n0} B \sin\varphi] f_\varepsilon(u_x), \tag{A2}$$

$$F_{ice,y} = q_n \int_{-L+B/2}^{L-B/2} f_\varepsilon\big(u_y(x)\big)dx + q_{n0}\cos\varphi\left(\int_{-L}^{-L+B/2} + \int_{L-B/2}^{L}\right) f_\varepsilon\big(u_y(x)\big)dx, \tag{A3}$$

where $2L$ and B are the FPU length and width, while $u_x = V_0 \sin\varphi - V_x$, $u_y(x) = V_0\cos\varphi - V_y - x\omega$ are relative longitudinal and transverse components of ice drift with respect to the FPU hull at the point x.

Quantity q_n characterizes normal ice loads applied to the unit length of the FPU hull in transversal direction to the FPU centerline. The shear ice load is calculated with the formula $q_\tau = \mu q_n$, where μ is the coefficient of ice-steel friction. Quantity q_{n0} characterizes ice action on bow and stern parts of the FPU hull which should be much less than normal load q_n due to well streamed geometry of the FPU hull in the longitudinal direction.

It is assumed that ice acts on one or another FPU sides (at a given point x) only when the difference of velocity components normal to the hull of ice drift and the FPU hull is positive at x. Difference of the tangential components of the ice and FPU velocities specifies the side, starboard or portside, and the direction of the shear force from the ice action at the point x. Signs of these differences can change due to FPU motion. We introduce a threshold function

$$f_\varepsilon(U) = \begin{cases} \text{sign}(U), & |U| \geq \varepsilon \\ 0, & |U| < \varepsilon \end{cases}, \quad \varepsilon \sim 10^{-3} \ldots 10^{-2} \text{ m/s} \tag{A4}$$

to specify a minimal value for mutual velocity of ice and an FPU, below which ice load at a given point x is assumed to be zero in order to smooth abrupt change of the direction of ice action on the ship hull.

Projections of the reaction of mooring and riser system on the axes x and y are determined by the equations

$$F_{mor,x} = -k_{mor}\left[\Delta_x \cos\varphi + \Delta_y \sin\varphi\right], F_{mor,y} = -k_{mor}\left[-\Delta_x \sin\varphi + \Delta_y \cos\varphi\right], \tag{A5}$$

where $\left(\Delta_x, \Delta_y\right) = \overrightarrow{S(0)S(t)}$ is the displacement vector for the turret.

Hydrodynamic resistance of the hull is specified by the formulas

$$FD_x = \left[\frac{1}{2}\rho_w BT(u_x)^2 \text{sign}(u_x) + k_{D,x} KD_x^{(crit)} u_x\right], \tag{A6}$$

$$FD_y = \frac{1}{2}\rho_w T \int_{-L}^{L} \left(u_y(x)\right)^2 \text{sign}(u_y(x)) dx + k_{D,y} KD_y^{(crit)} \left(V_y - V_0 \cos\varphi\right), \tag{A7}$$

where ρ_w is the water density, T is the FPU draft, and $KD_x^{(crit)}$ and $KD_y^{(crit)}$ specify critical damping

$$KD_x^{(crit)} = 2\sqrt{k_{mor}(m + m_{11})}, \; KD_y^{(crit)} = 2\sqrt{k_{mor}(m + m_{22})}. \tag{A8}$$

The first summands in (A6) and (A7) characterize the hydrodynamic resistance of the FPU hull, the second summands specify the damping from the mooring and riser system of FPU due to their motions. According to the recommendations of DNV-OS-E301 (2010) и simulation results of Ormberg and Larsen (1998) numerical values of linear damping coefficients for the surge and sway displacements are given by the formulas $k_{D,x} = 0.1$ and $k_{D,y} = 0.2$.

Forces $\mathbf{F}_{ice}$, $\mathbf{F}_{mor}$ and $\mathbf{FD}$ creates torsion moments specified by the formulas

$$M_{ice} = \int_{-L}^{L} \left(q_n x + q_\tau \frac{B}{2} f_\varepsilon(u_x)\right) \cdot f_\varepsilon(u_y(x)) dx, \; M_{mor} = -F_{mor,y} \cdot r_0, \tag{A9}$$

$$MD = \frac{1}{2}\rho_w T \int_{-L}^{L} \left(u_y(x)\right)^2 x \text{sign}(u_y(x)) dx, \tag{A10}$$

where r_0 is the distance between the gravity center and the turret. Total torsion moment is equal to the sum $M_z = M_{ice} + M_{mor} + MD$.