Application of the Hull Turnover Afloat as a Shipbuilding Tool

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ABSTRACT

The upside down construction increases the assembly speed. However, the conventional turnover involves considerable difficulty. This paper describes a procedure to accomplish a turnover with the hull floating on the water. According to this procedure a relatively small force, applied by a crane, take the hull up to a given heel angle where the unbalanced weight and force of buoyancy rights the structure. A numeric method is introduced aimed at providing the designer with a tool to figure out the proper solutions for the turnover planning. Typical output obtained from ordinary stability software is used as data entry.

Keywords: *Hull turnover, Upside down hull construction, Ship stability.*

1. INTRODUCTION

Figure 1 Turnover afloat of a patrol vessel's hull at INACE Shipyard

The upside down position presents many advantages for building hulls that have too much shape, frequently without or with a short parallel middle body. This type of hull form is typical for military vessels, fast PSVs, crew boats, tugs, fishing boats, mega-yachts and the like. The usage of the deck as a supporting base to assembly the framing or blocks is simpler than the erection in the normal position upon cradles. The upside down construction methodology increases the assembly speed and improves the shell distortion control. A better final product is obtained, with significant cost reduction.

Assuming that the upside down methodology gives several benefits to the hull construction, in the other hand the conventional turnover involves considerable difficulty. Normally the shipyards dedicated to the construction of small and medium sized vessels doesn't have cranes or grant cranes with sufficient capacity to turnover the entire hull in an only operation. One usual solution is to divide the hull, after the assembly, in two or more large blocks, noting that the joints between these units were not previously welded, but only aligned. However, an increasingly quantity of blocks implies in more re-alignment problems, with the risk of loose the gains achieved before in the upside down phase. In the same way the adoption of big wheels, commonly used to turnover small boats, rolling them on the ground is not technically or economically feasible for larger vessels.

Additionally, in a turnover on land the interaction between the concentrated lifting forces and the distributed loading, associated to the hull own weight, may induce permanent deformations to the structure.

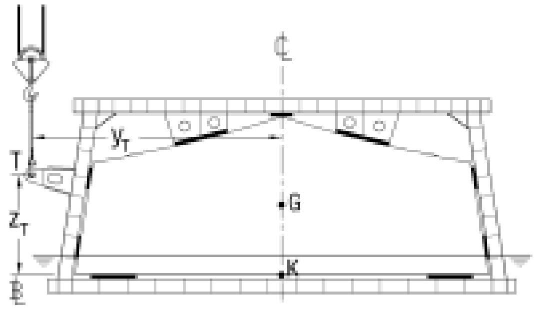


Figure 2 Typical turnover ring assembly

INACE Shipyard, located at Fortaleza, Brazil, introduced this procedure on 1996 for a Brazilian Navy patrol vessel. Until now three patrol vessels were successful turned according this technique. Figure 1 shows the last operation on October of 2005, for a 46 m patrol vessel under construction for the Namibian Navy

2. DEFINITIONS

2.1 Turnover Ring Assembly

Figure 2 is a hull transverse section showing a typical assembly for the turnover ring. It is a structural frame that surrounds the shell and the deck, in order to transfer the lifting force applied by the crane at the point T. In the figure the hull is floating in the upside down position.

Along this paper a hypothetical aluminium hull for a fast PSV / crew boat was considered as a useful case study. In the condition shown in Figure 2 the hull structure is completely finished. All apertures on deck are closed or are not cut yet, providing a full watertight body.

Although in a heavy hull more than one ring can be necessary, in this study we considered only one attached to the hull. Indeed, the evaluations on Section 4 shows that the maximum force applied by the crane is relatively low, if compared with the hull weight.

The parameters indicated on Figure 2 are defined as the following:

- KG = Height of the center of gravity from the inverted base line (located over the flat main deck) = 1.53 m
- yT = transversal position from center line (half beam) of the point T, where the crane lifting force is applied
- zT = vertical position from base line (height) of the point T.

There are no previously defined values for yT and zT as the main goal of this paper is just to provide a criteria to determine a correct and safe position for the point T.

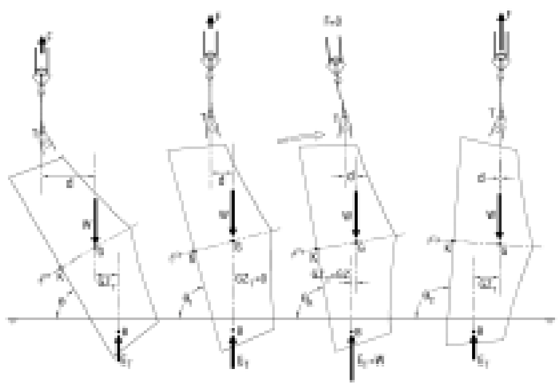


Figure 3 Turnover sequence, showing the interaction between the forces acting on the hull.

Figure 3 shows the turnover sequence. The ring structure was omitted for better visualization.

2.2 Parameters

Figure 3a represents the hull at a given heel angle θ. The parameters shown are defined as the following:

- W = Weight of the hull plus the ring, applied at the center of gravity G, equal to 42.06 metric tons for the present case study.
- F = Lifting force applied by the crane at the point T. The secondary goal of this paper is to determine the maximum lifting force demanded by the turnover, so the designer can figure out the safety for the cranes intended to use in the operation.
- E_T = Buoyancy force acting on the buoyancy center B. Note that differently of what occurs on an ordinary flotation condition, where the buoyancy force is always equal to W, in the turnover afloat process. E_T changes for each heel angle, following the variation of F.
- GZ_T = Distance between the vertical lines of W and E_T. Although similar, this distance is not equal to the righting arm GZ of an ordinary flotation condition.
- d = Distance between the vertical lines of W and F.

3. STATIC AND DYNAMIC CONSIDERATIONS

The correct determination of the turnover point (T) position on the ring is the main goal for the present paper. A wrong evaluation could introduce high risks to the operation or, on the other hand, the turnover may not occur.

3.1 Turnover Angle and Turnover Range

In Figure 3b the hull reached the beginning of the turnover. Then, from that point there will be only “capsizing” moments around point G, and the hull can even turnover itself without the help of F. We define θ_1 as the heel angle where the vertical line of B matches the vertical line of W, or the Turnover Angle.

Up to the Turnover Angle θ_1 there is equilibrium between the moments around G due to the forces E_T and F. This means that the crane has full control over the hull, and if something goes wrong the operator can lowering it back to the initial position (Fig.2). Near the Turnover Angle, F is very low and E_T is almost equal to W.

After the Turnover Angle θ_1 the position that corresponds to θ_R on Figure 3c could be anywhere in the range between θ_1 and θ_2 (Fig. 3d). In such condition the hull is turning free. Within this range the vertical line associated to F is yet placed “before” or very close to the vertical line of W. So, to restore the equilibrium F should act inverted, pointing to the water. This is impossible for the crane, and the hull remains accelerating itself “after” θ_1.

The movement is relatively fast, and the crane has no mechanical conditions to follow it. The cable becomes loose for a while.

When the hull reach the position that corresponds to heel angle θ_2 the vertical line of F goes slightly “after” the vertical line of W, as shown in Figure 3d. Now, at least theoretically, F can balance ET again

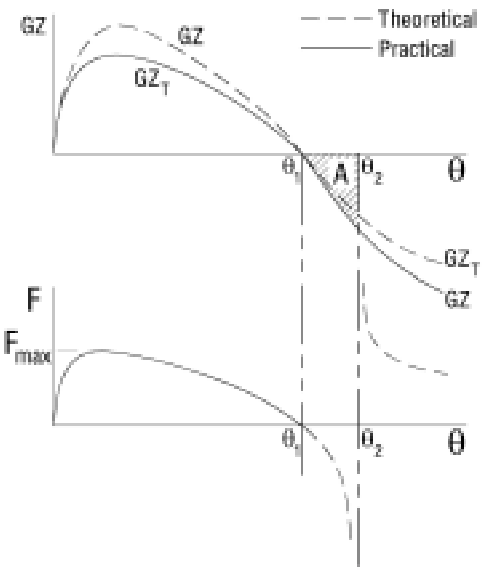
However, the distance d is yet very short at θ_2. On the other side, at the same time, GZ_T is already larger than d. So F must be very intense to compensate the lack of lever arm.

Additionally, the hull acquired considerable kinetics energy, accelerated by the moment generated by W and E_T, turning free between θ_1 and θ_2. Therefore, the crane may be hit by a severe strike at θ_2.

Figure 4 suggests that the way to avoid accidents in the turnover is to keep the heel angle range between θ_1 and θ_2 as short as possible.

As the force F “not exists” between θ_1 and θ_2, only weight and displacement acts on the hull, in a conventional way. Thus the “capsizing” arm GZ_T works like the conventional GZ in an ordinary static stability curve.

Figure 4 Synchronism of GZ_T, GZ and F behaviours



It is known that the area below the GZ curve is directly proportional to the work generated by the “capsizing” moment. This work will be transformed in kinetics energy. Therefore, a good criteria to figure out if the heel angle range is too large is to compare the energy generated with the capacity of the crane system to absorb it.

We should note in Fig. 4 that the curve for F after θ_1 was considered theoretical due to the fact that after this point up to θ_2 it is impossible to the crane to control the movement, and after θ_2 it will be unnecessary try to keep the balance between forces F and E_T. The crane operator should release the cable, allowing the hull to “fall” towards the up right position, maybe only applying the breaks slightly to reduce the speed.

3.2 No Turnover

A short range between θ_1 and θ_2 means a more safety operation. But a very short range, or even no range, can avoid the turnover. Secondary effects like the wind or water resistance and friction are difficult to figure out correctly in such situation, so the designer should take into account a margin to compensate them.

A wrong position for the point T could avoid the turnover too. Figure 5 shows the crane acting on point T’ instead of point T. At that point the arm d becomes too short to compensate GZ_T. Although a well planned turnover occurs with very low values for force F when the hull is getting close to θ_1, this force could reach values nearing W. This value is even greater than Fmax, which should occur at low heel angles. This means that the hull is being lifted instead of turned.

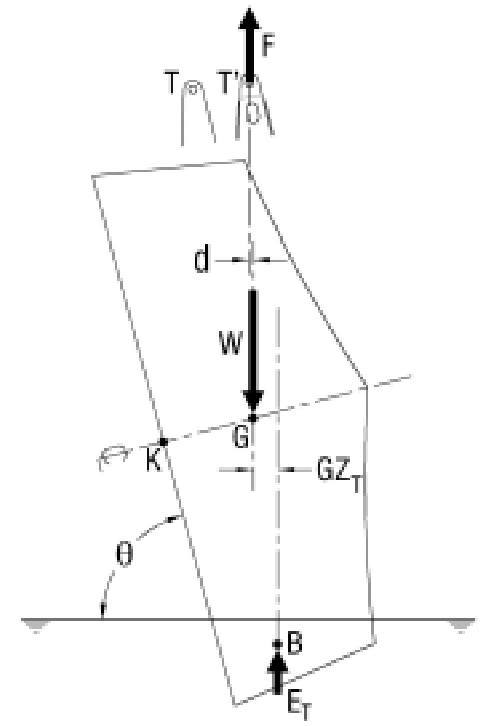


Figure 5 Applying F on point T’ instead of T only lifts the hull

4. NUMERICAL METHOD

The methodology described in the present Section has the following objectives:

- Find the ideal position for the point T
- Figure out the maximum force that acts on the point T, and what is the related heel angle.

4.1 Equilibrium Equations

Considering that the process can be undertaken in an almost static way, we can state that the equilibrium equations are:

$$\sum M_G = 0 \tag{1}$$

$$\sum F_x = 0 \tag{2}$$

$$\sum F_y = 0 \tag{3}$$

$$\sum F_z = 0 \tag{4}$$

The forces acting along x and y directions (length and beam respectively) are not significant, and equations (2) and (3) will not be used.

In order to simplify the evaluations, the moments are considered around the point G. Taking into account all the moments based on the forces from the Figure 3a we can develop Eq.(1) as the following:

$$\sum M_G = F \cdot d - E_T \cdot GZ_T$$

$$0 = F \cdot d - E_T \cdot GZ_T$$

$$F = \frac{E_T \cdot GZ_T}{d} \tag{5}$$

Considering all the vertical forces shown in Figure 3a we can develop (4) as the following:

$$\sum F_Z = E_T + F - W$$

$$0 = E_T + F - W$$

$$F = W - E_T \tag{6}$$

Thus to determine F there are two equations. According Eq. (5) F could be known if the moment balance becomes known too. To Eq. (6) the relation is about the vertical forces. So, only to best comprehension of the proposed method we can rewrite (5) and (6) as:

$$F_M = \frac{E_T \cdot GZ_T}{d} \tag{7}$$

$$F_V = W - E_T \tag{8}$$

If the results obtained by our method are consistent thus:

$$F = F_M = F_V \tag{9}$$

This means that the difference between the two forces should be:

$$\Delta F = F_M - F_V = 0 \tag{10}$$

4.2 Interpolation of the Cross Curves

An important characteristic of the present method is the capability of using conventional cross curves calculated by ordinary computer software. One recommendation is to choose a software that can run the curves for any desired angle. This property will allow the possibility to study the heel angle at steps as close as 1 degree, giving better precision to accurately find the heel angles θ_1 and θ_2 mentioned on Section 3.

Other recommendation is to run the curves using an inverted offsets table. For a sample, as in the present case study, a heel angle of 60 degrees in an upside down basis corresponds to 120 degrees from the up right position.

Table 1 is a typical output generated by an ordinary naval architecture computer software for the stability cross curves. It presents the curves in a numeric format, where for each listed heel angle:

- D = Displacement in salt water expressed in metric tons.
- GZ_0 = Righting arm considering the center of gravity hypothetically placed over the point K

Table 2 is a sample of how to determine F, E_T and GZ_T to a arbitrary chosen heel angle of 60 degrees,.

The corrected value of the conventional GZ, for the real z coordinate of G, was obtained using the well known equation:

$$GZ = GZ_0 - KG \cdot \sin\theta \tag{11}$$

The lever arm for the force F was obtained using the following equation:

$$d = (KG - z_T) \cdot \sin\theta + y_T \cdot \cos\theta \tag{12}$$

The value for KG was given in Section 2.1, and for merely checking how the method works we arbitrate y_T = 5.5 m and z_T = 2.2 m as the position of point T. Thus, substituting these values in (12) we found:

$$d = 2.17 \text{ m}$$

Note that the above value is constant for all displacements related to the current heel angle of 60 degrees.

Looking to the ΔF column in Table 2 we can note that with the increase of the displacement that difference is increasing too, positively. In the highlighted lines, between the displacements 26.01 t and 32.27 t, the signal of ΔF changes from negative to positive, meaning that the curve describing that parameter passed across a zero value in the range between -3.04 t and 6.50 t.

Therefore if ΔF is zero F_M and F_V matches each other and we can interpolate the values for E_T and GZ_T. In this study we use linear interpolation, observing that it not introduces significant imprecision to the results.

Table 1 Cross Curves of Stability for the hull related to the case study mentioned on 2.1.

10°		20°		30°		40°	
D (t)	GZ_0 (m)	D (t)	GZ_0 (m)	D (t)	GZ_0 (m)	D (t)	GZ_0 (m)
1.09	4.28	0.54	4.19	0.38	3.89	0.31	3.46
4.50	4.08	2.21	4.09	1.54	3.84	1.27	3.44
10.39	3.88	5.07	4.00	3.53	3.79	2.91	3.41
18.83	3.68	9.16	3.91	6.34	3.74	5.23	3.39
29.91	3.48	14.50	3.82	10.02	3.69	8.23	3.36
43.70	3.27	21.11	3.72	14.57	3.64	11.95	3.34
60.23	3.07	29.02	3.63	20.01	3.59	16.40	3.31
79.52	2.86	38.21	3.54	26.33	3.54	21.57	3.29
101.57	2.66	48.73	3.44	33.57	3.48	27.47	3.26
126.4	2.46	60.63	3.35	41.70	3.43	34.11	3.23
153.9	2.26	73.89	3.25	50.73	3.38	41.49	3.21
184.1	2.06	88.52	3.16	60.68	3.33	49.62	3.18
216.9	1.86	104.5	3.06	71.57	3.28	58.49	3.15
252.1	1.67	121.9	2.97	83.42	3.22	68.11	3.13
289.7	1.49	140.7	2.87	96.23	3.17	78.47	3.10

50°		60°		70°		80°	
D (t)	GZ_0 (m)	D (t)	GZ_0 (m)	D (t)	GZ_0 (m)	D (t)	GZ_0 (m)
0.29	2.92	0.30	2.29	0.34	1.60	0.46	0.87
1.18	2.92	1.22	2.31	1.41	1.63	1.89	0.94
2.70	2.91	2.78	2.32	3.20	1.67	4.29	1.01
4.85	2.91	4.99	2.34	5.75	1.71	7.68	1.09
7.63	2.90	7.86	2.35	9.04	1.75	12.06	1.16
11.06	2.90	11.38	2.37	13.08	1.79	17.45	1.23
15.16	2.89	15.57	2.38	17.88	1.83	23.83	1.31
19.92	2.89	20.45	2.40	23.46	1.86	31.24	1.38
25.36	2.88	26.01	2.41	29.83	1.90	39.64	1.45
31.48	2.88	32.27	2.42	36.99	1.94	48.70	1.51
38.26	2.87	39.22	2.44	44.93	1.98	58.21	1.55
45.74	2.87	46.85	2.45	53.66	2.01	68.08	1.59
53.91	2.86	55.19	2.47	63.14	2.05	78.22	1.61
62.76	2.85	64.23	2.48	73.16	2.08	88.61	1.63
72.31	2.85	73.98	2.49	83.57	2.10	99.22	1.65

The values interpolated on Table 2 for E_T and GZ_T were:

$$E_T = 28.00 \text{ t} \quad ; \quad GZ_T = 1.08 \text{ m}$$

In the same way the values interpolated on Table 2 for F_M and F_V were:

$$F_M = 14.06 \text{ t} \quad ; \quad F_V = 14.06 \text{ t}$$

It is important to note that they matched precisely, thus according (9):

$$F = F_M = F_V = 14.06 \text{ t}$$

Only for an additional check of the results coherence we can rearrange (6) to obtain exactly the same hull weight W defined in Section 2.2:

$$W = F + E_T$$

$$W = 14.06 + 28.00 = 42.06 \text{ t}$$

Table 2 Sample of interpolation from the Cross Curves, for a heel angle of 60° (y_T = 5.5 m; z_T = 2.2 m)

D From CC (t)	GZ_0 From CC (m)	GZ Eq.(11) (m)	F_M Eq.(7) (t)	F_V Eq.(8) (t)	ΔF Eq.(10) (t)
0.30	2.29	0.96	0.13	41.76	-41.63
1.22	2.31	0.98	0.55	40.84	-40.29
2.78	2.32	0.99	1.27	39.28	-38.01
4.99	2.34	1.01	2.33	37.07	-34.74
7.86	2.35	1.02	3.71	34.20	-30.49
11.38	2.37	1.04	5.48	30.68	-25.20
15.57	2.38	1.05	7.57	26.49	-18.92
20.45	2.40	1.07	10.13	21.61	-11.48
26.01	**2.41**	**1.08**	**13.01**	**16.05**	**-3.04**
32.27	**2.42**	**1.09**	**16.29**	**9.79**	**6.50**
39.22	2.44	1.11	20.15	2.84	17.31
46.85	2.45	1.12	24.29	-4.79	29.08
55.19	2.47	1.14	29.12	-13.13	42.25
64.23	2.48	1.15	34.19	-22.17	56.36
73.98	2.49	1.16	39.72	-31.92	71.64

4.3 Analysis of the Turnover Force F

Repeating the process described in 4.2 for each desired heel angle generates the Table 3. Now we are looking in a first instance for the range where F becomes negative, i.e. the heel angles θ_1 and θ_2 defined in 3.1.

In Table 3 the normal step for the heel angle is 10 degrees. But near the Turnover Angle we refined the results lowering the interval to 1 degree. In the highlighted lines we see that F becomes negative from 80 to 83 degrees. These are the values for θ_1 and θ_2 respectively

Note as F falls to a low value at 79 degrees, close to the Turnover Angle, and how it jumps from a negative value to a high positive one after reaching θ_2, as shown in Figure 4.

Other important result is the maximum value F_{max} for the force F, and the heel angle θ_{max} where it occurs. In the two last columns of Table 3 we obtained a maximum force equal to 16.34 t at a heel angle of 20 degrees.

Table 3 Turnover parameters as function of the heel angle (y_T = 5.5 m ; z_T = 2.2 m)

θ (deg.)	E_T (t)	GZ_T (m)	d (m)	F (t)
0	0	0	5.50	0
10	26.03	3.28	5.30	16.03
20	25.72	3.14	4.94	16.34
30	25.85	2.78	4.43	16.21
40	26.23	2.28	3.78	15.83
50	26.87	1.71	3.02	15.19
60	28.00	1.08	2.17	14.06
70	30.62	0.47	1.25	11.44
75	33.97	0.18	0.78	8.09
79	41.03	0.01	0.39	1.03
80	**44.92**	**-0.02**	**0.30**	**-2.86**
81	**44.51**	**-0.08**	**1.38**	**-2.45**
82	**46.47**	**-0.12**	**1.29**	**-4.41**
83	**48.70**	**-0.17**	**1.20**	**-6.64**
84	4.50	-0.77	-0.09	37.56
85	8.72	-0.73	-0.19	33.34
90	18.42	-0.74	-0.57	23.64

The results given in Tables 2 nd 3 were evaluated for a specific position of the turnover point, represented by the coordinates y_T and z_T. A slight change in the coordinate values can produce great modifications in the final results.

Table 4 shows an analysis where only the coordinate z_T changes, keeping the same value for y_T. We can see where the range Δθ is relatively large, and where the turnover shouldn't be accomplished properly.

Table 4 θ range and F_{max} comparison for different z_T values

Z_T (m)	θ_1 (deg.)	θ_2 (deg.)	$\Delta\theta$ (deg.)	F_{max} (t)	θ_{max} (deg.)
1.0	80	95	15	15.70	15
1.2	80	93	13	15.79	15
1.4	80	91	11	15.89	15
1.6	80	89	9	15.99	15
1.8	80	87	7	16.09	15
2.0	80	85	5	16.20	20
2.2	**80**	**83**	**3**	**16.34**	**20**
2.4	80	81	1	16.50	25
2.6	79	79	0	16.69	25
2.8	79	79	0	16.94	30
3.0	No Turnover			28.96	75
3.2	No Turnover			35.69	73

4.4 The Energy Criteria

Results similar as shown in Table 4 brings very important information about the turnover physics, and we can analyse what happens for each Turnover Point coordinates y_T and z_T associated with a given combination of hull weight W and KG.

Despite we can figure out the value of $\Delta\theta$ for any situation, how to choose the most suitable one?

Normally a crane manufacturer states what is the maximum speed which his equipment can lower a given weight. For a sample a certain crane can lower 40 metric tons at 25 m / min. Expressing the weight in terms of mass, and converting the speed to m/s we obtain:

$$m = 40000 \text{ kg}$$

$$v = 0.42 \text{ m/s}$$

The classic equation for kinetics energy is:

$$K = \frac{1}{2} \cdot m \cdot v^2 \tag{13}$$

Substituting the values for m and v in (13) we found the kinetics energy absorption limit for the crane:

$$K_L = 3.5 \text{ kJ} \tag{14}$$

In the other hand, as shown in Figure 4, only the conventional GZ is effective between the heel angles θ_1 and θ_2, describing a portion of a regular Static Stability Curve. In an ordinary flotation condition only the weight and displacement acts on the hull. As they are constant in such condition the Static Stability Curve can be plotted in terms of moment simple by multiplying the righting arm GZ by the constant weight value.

Expressing the heel angle in radians and the moment in N.m any area below the curve corresponds to the work necessary to change the hull from a given heel angle to another one. If the hull is coming back after heeling this is the work generated, transformed in kinetics energy achieved by the turning hull.

Therefore knowing the area A shown in Fig. 4 makes possible to determine the kinetics energy that arises between the heel angles θ_1 and θ_2. Table 5 shows the integration (by trapeziums) of this area for the condition analyzed in Table 3:

Table 5 Integration of the kinetics energy below the static stability curve

θ (deg.)	GZ_0 m	GZ m	M kN.m	ÁA (kJ)
80	1.47	-0.04	-15	-
81	1.42	-0.09	-38	-0.46
82	1.37	-0.15	-60	-0.85
83	1.31	-0.21	-86	-1.27
			Σ	**2.6**

Then the kinetics energy developed by the hull turning free up to θ_2 is equal to 2.6 kJ. This value corresponds to 74% of the crane limit, equal to 3.5 kJ, given by (14).

5. CONCLUSION AND DISCUSSION

Section 4.4 shows that for the case study analysed a $\Delta\theta = \theta_2 - \theta_1 = 3$ degrees is a suitable value to the turnover operation, with the hull kinetics energy corresponding to 74% of the crane limit,

Obviously is preferable to not exceed the crane limit, to avoid a strike against it. But we suggest, depending upon further investigation, that the $\Delta\theta$ should be adjusted to avoid kinetics energy values below 50% of the crane limit, as a margin can be necessary to compensate secondary effects due to wind, water resistance and water friction.

It is important to note that if we change, for a sample, only the value of z_T from 2.2 m to 1.6 m, keeping the same values for all the others parameters, we verified on Table 4 that $\Delta\theta$ is equal to 9 degrees.

Appling to this condition the same procedure used to create Table 5 reveals that when reach θ_2 the hull would have accumulated kinetics energy equal to 18 kJ, approximately 5 times the crane limit. This fact shows how a relatively small variation of the coordinate z_T (0.6m, corresponding to 17% of hull depth) can produce great risks for the operation.

Other interesting point, not covered in this paper, is the investigation about the influence of the trim variation along the heeling process.

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