Jurnal Teknologi

Full paper

Analysis of an Asymmetrical Bridle Towline Model to Stabilise Towing Performance of a Towed Ship

A. Fitriadhy[a*], H. Yasukawa[b], T. Yoneda[c], K. K. Koh[d], A. Maimun[d]

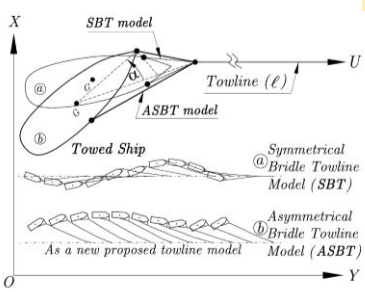
[a]Department of Maritime Technology, Universiti Malaysia Terengganu, Terengganu, Malaysia
[b]Department of Transportation and Environmental Systems, Hiroshima University, Japan
[c]Mazda Motor Co. Ltd, Hiroshima, Japan
[d]Department of Marine Technology, Universiti Teknologi Malaysia, Skudai, Johor, Malaysia

*Corresponding author: naoe.afit@gmail.com

Article history

Received :29 October 2013
Received in revised form :
1 November 2013
Accepted :5 December 2013

Graphical abstract



Abstract

This paper addresses an asymmetrical bridle towline model as a feasible solution model to stabilize towing performance of a towed ship. The basic thinking behind the approach adopted, is to deal with a better towing stability than employing the typical towline model[1]. Several towing parameters which may affect a towed ship motion behaviour i.e., tow angle and tow point position, are investigated theoretically. The nonlinear numerical time-domain simulation showed that the increase of towing angle up to 30 degrees and shifting tow point from 0.5 to 0.8 resulted in remarkable reduction in slewing motion of the unstable towed ship and the towline tension, which enhances effectively her towing stability.

Keywords: Asymmetrical bridle towline; tow angle; tow point; slewing motion; towed ship

Â1.0 INTRODUCTION

In the ship towing system, employing improper towline configuration may cause instability of towed ship indicated by an excessive heading angle or rigorous slewing motion. This will intrude on sea traffic ways, especially in the restricted waters; even, it will introduce serious towing accidents such as the towed ship collides with other ships or shore installations. Concern for safety, study on towline configuration, which affects towing behavior of a towed ship is therefore required.

Several researches have been performed on modeling a towline that typically employs in the ship towing system. The symmetrical bridle towline was modelled on towing of the unstable towed ship; the result revealed that by replacing of tow point to be more forward on the towed ship improved the towing stability indicated by sufficient reduction in her slewing motion.[1] inherently, this dealt with decreasing the towline.[2]

In this paper, the authors develop numerical simulation addressing effect of an asymmetrical bridle towline model to stabilize towing performance of a towed ship. Two main components of towing parameters associated with this towline model i.e., effect of tow angle and tow point forward on the towed ship, are discussed. A 2D-lumped mass method is adopted for modelling the towline motion. Due to highly non-linear interactions between tow and towed ships, the towed ship is decoupled from the tow ship, where her motion is assumed to be given with constant speed

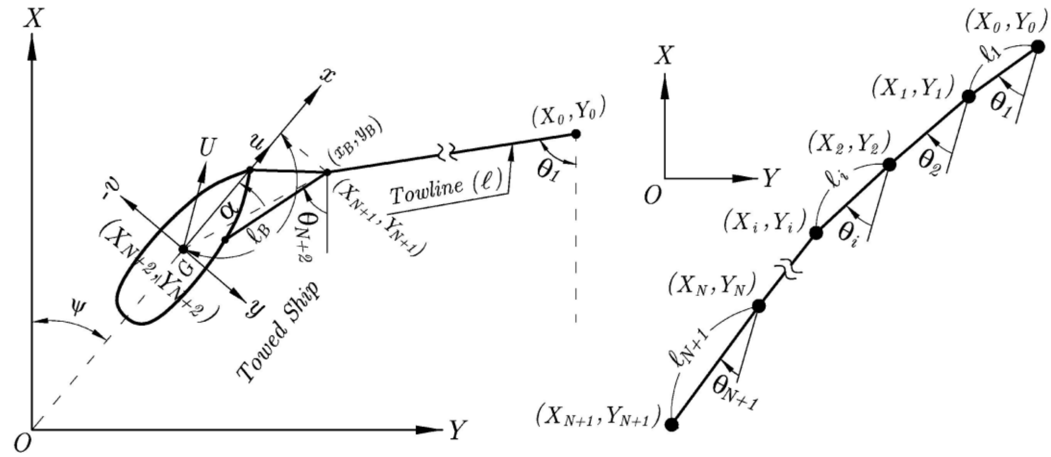


Figure 1 Coordinate system of asymmetrical bridle towline model (left) and lumped mass model of towline (right)

2.0 MATHEMATICAL MODEL

The nonlinear manoeuvring equation describes the motion of the towed ships along with the towline expressed in three degrees of freedom (surge, sway, yaw motions). The component of hydrodynamic force and restoring moment of the towed ship due to hull is modelled. The components of hydrodynamic force and restoring moment of the towed ship are taken into account

2.1 Coordinate System

Two coordinate systems are used Figure 1 (left). One set of axes is fixed to the earth coordinate system, which denoted as; $O-XY$ and $G-xy$ is fixed relative to ship moving coordinate system aligned with its origin at center of gravity. In the moving reference, x-axis points to forward and y-axis points to starboard. The heading angle ψ refers to the direction of the ship's longitudinal axis x with respect to the fixed X-axis. The instantaneous speed of ship can be decomposed into an advance velocity u and a transverse velocity. The angle between U and the x axis is the drift angle $\beta = -\tan^{-1}(v/u)$.

The towline is represented as the sum of lumped mass of finite number (N). Each mass is connected by the segmented element over the entire truss element. The lumped mass particulars describe the towline characteristics, such as the mass, the density and the drag. The coordinates of its lumped mass is labeled by (X_i, Y_i), where $(i = 1,2,3,...,N+1)$. The angle between X-axis and length of the i^{th} segmented towline ℓ_i is denoted as θ_i. Their connection points through a towline in the earth fixed coordinate systems are denoted as (X_0, Y_0) and (X_{N+1}, Y_{N+1}). With respect to each center of gravity of the local ship coordinate system, the distance of this tow point is defined as (x_B, y_B), where $(x_B = \ell_B)$. The coordinate system (X_i, Y_i), which defines θ_i and ℓ_i can be expressed as follow:

$$X_i = X_0 - \sum_{j=1}^{i} \ell_j \cos\theta_j, \qquad Y_i = Y_0 - \sum_{j=1}^{i} \ell_j \sin\theta_j \tag{1}$$

where $\theta_{N+2} = \psi$ and $\ell_{N+1} = \ell_B$

2.2 Motion Equations of a Towed Ship

The motion equation of the towed ship is rewritten in Equations (2), (3) and (4), $M_X = (m + m_x)$ and, $M_Y = (m + m_y)$ represent the virtual mass components in x and y, respectively. $I_Z = (I + J)$ is the virtual moment of inertia. F_x, F_y and M_z are the surge force, the sway force, and the yaw moment acting on the towed ship, respectively.

$$(m + m_x)\dot{u} - (m + m_y)v\dot{\psi} = F_x + F_{Tx} \tag{2}$$

$$(m + m_y)\dot{v} - (m + m_x)u\dot{\psi} = F_y + F_{Ty} \tag{3}$$

$$(I + J)\ddot{\psi} = M_z + M_{Tz} \tag{4}$$

The right side components i.e. $F_{Tx} = T_V\cos\gamma$, $F_{Ty} = T_V\sin\gamma$, and M_{Tz} are denoted as the surge and the sway forces, and the yaw moment due to the towline tension, respectively. Here, $M_{Tz} = T_V r_T$, where $r_T = x_B\sin\gamma - y_B\cos\gamma$ and $\gamma = \theta_1 - \psi$. The notations of u and v are the forward and lateral velocities of the towed ship in the local fixed coordinate system, respectively, and expressed as follows:

$$u = \dot{X}_{N+2}\cos\psi + \dot{Y}_{N+2}\sin\psi \tag{5}$$

$$v = -\dot{X}_{N+2}\sin\psi + \dot{Y}_{N+2}\cos\psi \tag{6}$$

Refer to Equations (2) and (3), the equation can be recast in the form;

$$M_X\dot{u} = \breve{F}_x + T_V\cos\gamma \tag{7}$$

$$M_Y\dot{v} = \breve{F}_y + T_V\sin\gamma \tag{8}$$

By eliminating of T_V, it yields

$$M_X\dot{u}\sin\gamma - M_Y\cos\gamma = \breve{F}_x\sin\gamma - \breve{F}_y\cos\gamma \tag{9}$$

Then, the equation of T_V should be of the form

$$T_V = M_X\dot{u}\cos\gamma + M_Y\dot{v}\sin\gamma - \breve{F}_x\cos\gamma - \breve{F}_y\sin\gamma \tag{10}$$

By substituting Eqs.(5) and (6) into Eq.(10) yields

$$\ddot{X}_{N+2} M_X + \ddot{Y}_{N+2} M_Y = F_1 - F_2 \qquad (11)$$

From substitution of Equation (1) into Equation (11) will obtain

$$\sum_{j=1}^{N+2} \ell_j (M_x \sin\theta_j - M_y \cos\theta_j)\ddot{\theta}_j = \sum_{j=1}^{N+2} \ell_B (M_x \cos\theta_j + M_y \sin\theta_j)\dot{\theta}_j^2 + M_x \ddot{X}_0 + M_y \ddot{Y}_0 - F_1 - F_2 \qquad (12)$$

Where

$$F_1 = -\tilde{F}_x \sin\gamma + \tilde{F}_y \cos\gamma$$
$$F_2 = M_X \dot{\psi} v \sin\gamma + M_Y \dot{\psi} u \cos\gamma$$
$$\tilde{F}_x = F_x + M_Y v \dot{\psi}$$
$$\tilde{F}_y = F_y + M_X u \dot{\psi}$$
$$M_x = M_X \sin\gamma \cos\psi + M_y \cos\psi \sin\gamma$$
$$M_y = M_X \sin\gamma \cos\psi + M_Y \cos\psi \sin\gamma$$

Furthermore, the substitution of Eqsuations (5), (6) and (10) into Equation (4) becomes

$$I_Z \ddot{\psi} - \ddot{X}_{N+2} I_x - \ddot{Y}_{N+2} I_y = M_1 + M_2 \qquad (13)$$

Similarly, the substitution of Eq.(1) into Eq.(13) results

$$I_Z \ddot{\theta}_{N+2} + \sum_{j=1}^{N+2} \ell_j \ell_B (I_y \cos\theta_j - I_x \sin\theta_j)\ddot{\theta}_j$$
$$= \sum_{j=1}^{N+2} \ell_j \ell_B (I_y \sin\theta_j - I_x \cos\theta_j)\dot{\theta}_j^2 + M_1 + M_2 + \ddot{X}_0 I_x + I_y \ddot{Y}_0 \qquad (14)$$

Where

$$M_1 = -r_T (F_x \cos\gamma + F_y \sin\gamma) + M_{Z2}$$
$$M_2 = r_T (M_X \dot{\psi} v \cos\gamma - M_Y \dot{\psi} u \sin\gamma)$$
$$I_x = r_T (M_X \cos\gamma \cos\psi - M_Y \sin\psi \sin\gamma)$$
$$I_y = r_T (M_X \cos\gamma \sin\psi - M_Y \sin\psi \cos\gamma)$$

2.3 Motion Equation of Towline

With refer to Figure 1 (right), the Lagrange's motion equations are used to describe the dynamic motion of the towline as derived in Equation (15).

$$\sum_{i=k}^{N} \left[\sum_{j=1}^{i} (m_{si} \sin\theta_k \sin\theta_j + m_{ci} \cos\theta_k \cos\theta_j)\ell_k \ell_j \ddot{\theta}_j \right]$$
$$+ \ell_k \sin(\theta_k - \theta_{N+1}) \sum_{j=1}^{N+2} \ell_j (M_{x2} \sin\theta_j - M_{y2} \cos\theta_j)\ddot{\theta}_j$$
$$= Q_{0k} - Q_{1k} - \ell_k T_{V3} \sin(\theta_k - \theta_{N+1}) \quad (k=1,2,3,...,N) \qquad (15)$$

where

$$m_{ci} = m_i (1 + k_{Fi} \sin^2\theta_i)$$
$$m_{ci} = m_i (1 + k_{Fi} \cos^2\theta_i)$$
$$T_{V3} = \sum_{j=1}^{N+2} \ell_j (M_{x2} \cos\theta_j + M_{y2} \sin\theta_j)\dot{\theta}_j^2 + M_{x2}\ddot{X}_0 + M_{y2}\ddot{Y}_0 - T_{V1} - T_{V2}$$
$$Q_{0k} = -\ell_k \sin\theta_k \sum_{i=k}^{N} (R_{Ci} \sin\theta_i + F_{Ci} \cos\theta_i) - \ell_k \cos\theta_k \sum_{i=k}^{N} (R_{Ci} \cos\theta_i + F_{Ci} \sin\theta_i)$$
$$Q_{ik} = \ell_k \sin\theta_k \sum_{i=k}^{N} \left(\ddot{X}_0 + \sum_{j=1}^{i} \dot{\theta}_j^2 \ell_j \cos\theta_j\right) m_{si} - \ell_k \cos\theta_k \sum_{i=k}^{N} \left(\ddot{Y}_0 + \sum_{j=1}^{i} \dot{\theta}_j^2 \ell_j \sin\theta_j\right) m_{ci}$$
$$- \ell_k \cos\theta_k \sum_{i=k}^{N} \left(\ddot{Y}_0 + \sum_{j=1}^{i} \dot{\theta}_j^2 \ell_j \sin\theta_j\right) m_{ci}$$
$$+ 2\ell_k \sum_{i=k}^{N} (\dot{X}_i \sin\theta_k + \dot{Y}_i \cos\theta_k) m_i k_{Fi} \dot{\theta}_i \sin\theta_i \cos\theta_i$$
$$- (\dot{X}_k^2 - \dot{Y}_k^2) m_k k_{Fk} \cos\theta_k \sin\theta_k$$

The notations of m_i and k_{Fi} are the mass and the added mass coefficients of the i^{th} lumped masses, respectively.

Two different external forces experienced on the segmented towline are resolved into normal and axial forces:

$$R_{Ci} = -\frac{1}{2} \rho S_i C_{Di} |V_{Ci}| V_{Ci} \qquad (16)$$

$$F_{Ci} = -\frac{1}{2} \rho S_i C_{Fi} |U_{Ci}| U_{Ci} \qquad (17)$$

where $V_{Ci} = -\dot{X}_i \sin\theta_i + \dot{Y}_i \cos\theta_i$ and $U_{Ci} = \dot{X}_i \cos\theta_i + \dot{Y}_i \sin\theta_i$. ρ is the water density, S_i the profile area of the segmented towline, C_{Di} the coefficient of the normal force, and C_{Fi} the coefficient of the axial force.

2.3 Equation of Dynamic Towline Tension

Referring to Equations (2) and (3), the components of towline tension that acting at the connection point (X_0, Y_0) with respect to the earth coordinate system (X, Y) are expressed T_X and T_Y, respectively. According to the work of Yasukawa et al., the resultant towline tension at the tow point of the tow ship can be expressed as $T_C = \sqrt{T_X^2 + T_Y^2}$.[3]

2.4 Hydrodynamic Forces and Moments acting on the Towed ship

The hydrodynamic forces and moments on the ships at constant speed and course can be expressed as functions of the velocities. The forces and moments F_x, F_y and M_z are presented as the hull forces and moment (X_H, Y_H, N_H):

$$\left.\begin{aligned} F_x &= X_H \\ F_y &= Y_H \\ M_z &= N_H - x_G Y_H \end{aligned}\right\} \qquad (18)$$

Where

$$X_H = \frac{1}{2}\rho L d U^2 (X'_{uu} u'^2 + X'_{vv} v'^2 + X'_{vr} v' r' + X'_{rr} r'^2)$$

$$Y_H = \frac{1}{2}\rho L d U^2 (Y'_v v' + Y'_r r' + Y'_{vvv} v'^3 + Y'_{vvr} v'^2 r' + Y'_{vrr} v' r'^2 + Y'_{rrr} r'^3)$$

$$N_H = \frac{1}{2}\rho L^2 d U^2 (N'_v v' + N'_r r' + N'_{vvv} v'^3 + N'_{vvr} v'^2 r' + N'_{vrr} v' r'^2 + N'_{rrr} r'^3)$$

The notations of L and d denote the length and the draft of ship, respectively. $U = \sqrt{u^2 + v^2}$ is the ship's speed. $u' = u/U$ and $v' = v/U$ are the non-dimensional surge and sway velocities, respectively. $r' = \dot{\psi} L/U$ is the non-dimensional yaw rate. Y'_v, Y'_r, N'_v, N'_r ... etc. are the hydrodynamic manouevring derivatives of ship. $-X_{uu}$ is the resistance coefficient.

3.0 SIMULATION CONDITION

3.1 Ship

The principal dimensions of towed ship called as barge 2B is presented in Table 1. The notation of k_{zz} is the radius of gyration for yaw. The towing speed of 7.0 knots is employed in the simulation. In addition, the hydrodynamic derivatives and resistance coefficients of the barge (X'_{uu}) were obtained from captive model test in the towing tank (see Figure 2) and completely summarized in Table 2.

Table 1 Principal dimensions of barge 2B

Description	Full scale	Model
LBP (m)	60.96	1.22
Breadth (m)	21.34	0.43
Draft (m)	2.74	0.06
Volume (m^3)	3292.40	0.03
LCB (m)	-1.04	-0.02
Block coefficient C_b	0.92	0.92
k_{zz}/L	0.25	0.25
L/B	2.86	2.86

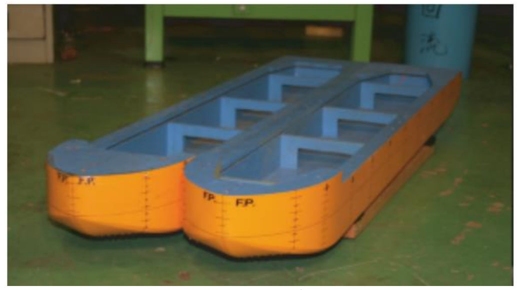


Figure 2 Model of barge 2B

Table 2 Resistance coefficient, hydrodynamic derivatives on manoeuvring and added mass coefficients

Non-dimensional Coefficient	2B
X'_{uu}	-0.0635
X'_{vv}	-0.0188
X'_{vr}	-0.0085
X'_{rr}	-0.0272
Y'_v	-0.4027
Y'_{vvv}	-0.2159
Y'_{vvr}	0.4840
Y'_{vrr}	0.495
Y'_{rrr}	-0.8469
N'_v	-0.1160
N'_{vvv}	0.0458
N'_{vvr}	0.0578
N'_{vrr}	0.2099
N'_{rrr}	-0.0982
m'_x	0.0391
m'_y	0.2180
J'_z	0.0124
C	-0.251

4.0 RESULTS AND DISCUSSION

Figure 3 (left) shows the effect of tow angle (α) on barge 2B motion performances associated with the dynamic tension in towline (T_C). In the range of α = 0° to 20°, it imposed the barge 2B to veer-off from the initial course while attenuating subsequently her amplitude of heading angle (ψ). Interestingly, the slewing motion of barge 2B was notably reduced and then settled steadily at $\psi = 18.4°$ as further increase of α up to 30°.

The reason for this can be explained that the increase of α was proportional to an increase of the sway forces alongside the hull, which caused barge 2B was vulnerable to drift to port-side and keeps her path once it reaches the prescribed heading angle. This means that the sway motion of barge 2B is more dominant than her yaw motion. As a result, this would stiffen r and implied barge 2B has a shorter period of the slewing motion. In addition, the lateral oscillation of barge 2B motion (Y) was significantly reduced and settled at the magnitude of 67 m, see Figure 6 (left). Meanwhile, the oscillation of impulsive T_C of 0.44 kgf also decreased proportionally and then resumed at steady T_C of 0.14 kgf. In general, the numerical simulation results seem to indicate the advantage of the asymmetrical bridle towline model in the ship towing system associated with increasing the tow angle allowed to improve adequate towing stability of the unstable barge 2B.

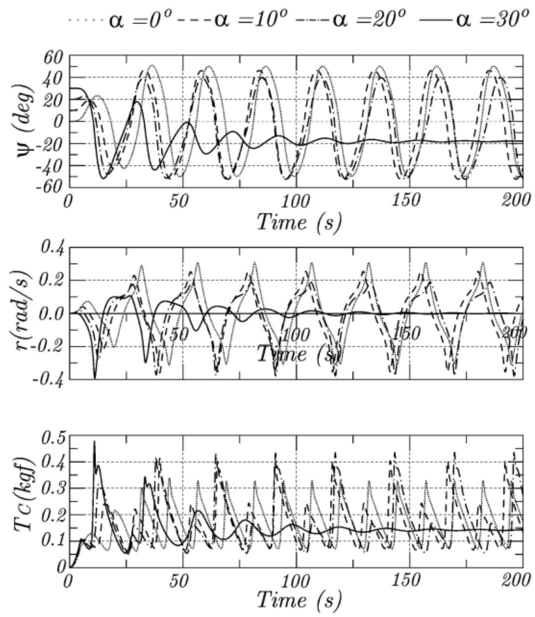


Figure 3 Effect of tow angle (α) on barge 2B motion and T_C ($\ell' = 2.0$ and $\ell'_B = 0.5$)

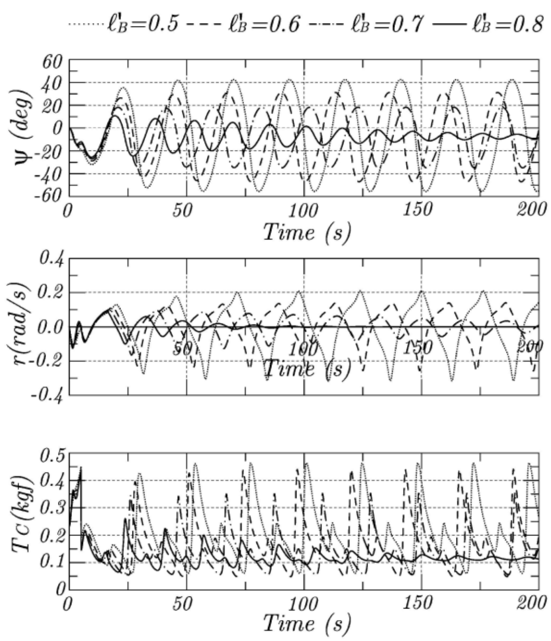


Figure 4 Effect of tow point (ℓ_B) on barge 2B motion and T_C ($\ell' = 2.0$ and $\alpha = 20^\circ$)

Furthermore, the effect of increasing ℓ'_B on the towing characteristics of barge 2B is illustrated in Figure 4. The simulation results revealed that towing stability of barge 2B was prone to be more stable indicated by the sufficient reduction in the amplitude of ψ and r as ℓ'_B increased from 0.5 to 0.8. From the Figure, the oscillation of r was prone to be zero that caused barge 2B to become steadier at port-side. The snapshot of the towing trajectories in Figure 6 (right) pointed-out that the amplitude of barge 2B lateral motion was diminished while she drifted and settled to a small oscillation of ψ. However, compared to the analysis of the symmetrical bridle towline model, the increase of ℓ'_B possessed a more adequate influence for reducing either the amplitude or the magnitude of T_C.[1] Refer to this analysis, the impulsive tension decreased from 0.48 kgf to 0.15 kgf at t = 200 seconds. This possibly occurred due to less acceleration and deceleration of barge 2B surge motion associated with slow-towing speed.

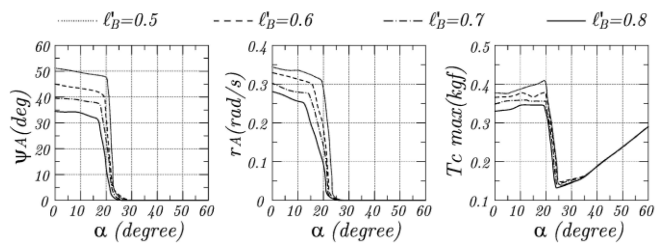


Figure 5 Characteristics of barge 2B motion and towline tension in different tow points ℓ'_B versus tow angle

The effect of tow angle on the asymmetrical bridle towline model associated with the shifting of tow point forward on barge 2B is illustrated in Figure 5. The increase of α either from 0° to 30° and 30° to 60° allowed generating a yaw damping moment, which implied the towing system was rendered relatively stiff. Although the amplitude of ψ and r reduced remarkably within $30^\circ \le \alpha \le 60^\circ$, the entire towing performance was still undesirable as indicated by the increase of T_C. It occurred since barge 2B was vulnerable to a rapid increase in the sway velocity at the beginning of the towing. As a result, this led to impose more vigorous motions of barge 2B indicated by the large magnitude of heading angle, which caused the resistance of barge 2B also proportionally increased. Trying to reduce the tension in the towline; however, a comparison for increasing values of ℓ'_B did not seem necessary, the influence appears to be trivial and r as α increased within the aforementioned tow angle above. It can be inferred that the increase of the tow angle in the asymmetrical bridle towline model has a more dominant effect to the overall dynamic towing system than shifting the tow point forward on the towed ship.

4.0 CONCLUSION

The effect of asymmetrical bridle towline configuration associated with the tow angle and the tow point position on the course stability of the towed ship were analyzed using the non-linear numerical simulation. Several conclusions are drawn as follows;

- Employing the asymmetrical bridle towline configuration associated with increasing tow angle ($0^\circ \le \alpha \le 30^\circ$) and the shifting the tow point forward on barge 2B ($0.5 \le \ell'_B \le 0.8$) improve effectively the towing stability of barge 2B, which results in significant reduction of the tension in the towline.
- However, the increase of the tow angle ($30^\circ \le \alpha \le 60^\circ$) causes impulses in the towline tension, which may threaten the safety of towing. This means that the tow angle has stronger effect than the tow point to the overall dynamic of the towing system.

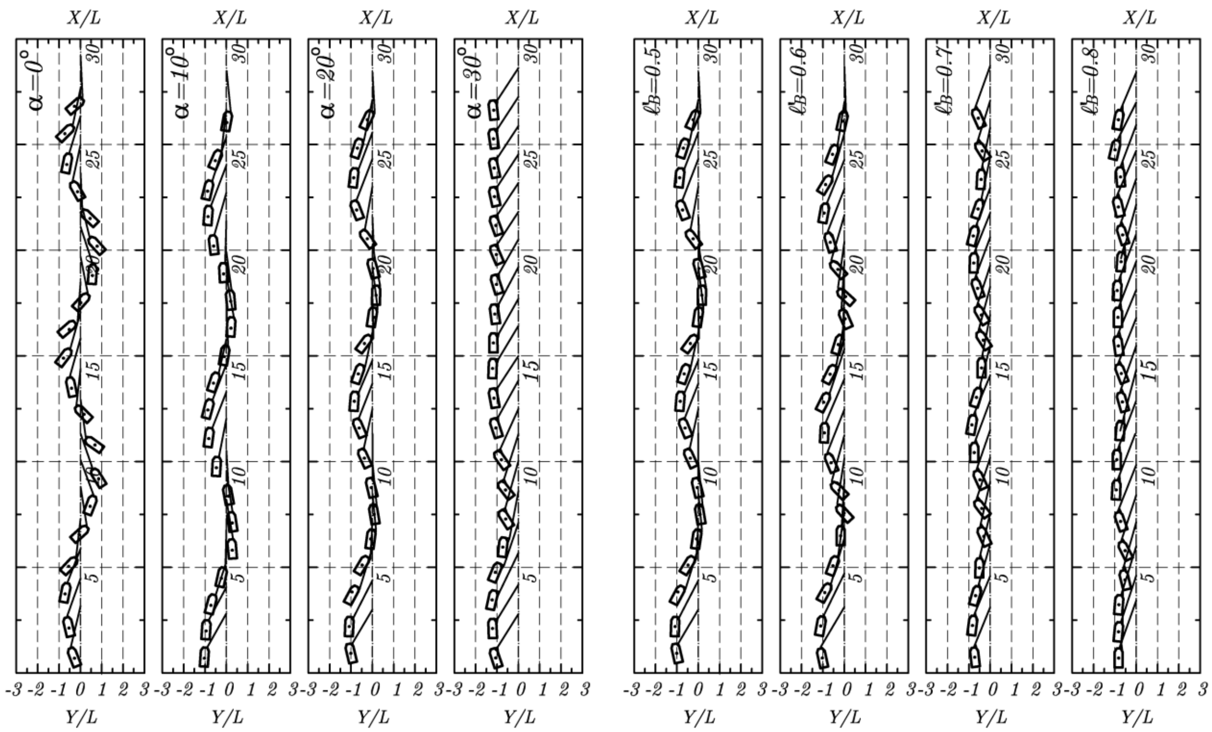


Figure 6 Snapshot of the towing trajectories for barge 2B in the various tow angles α (left) and tow points (ℓ_B) (right)

Acknowledgement

We thank to financial research grant TPM (Tabung Penyelidik Muda), Universiti Malaysia Terengganu (UMT).

References

[1] Fitriadhy, A., Yasukawa, H. 2011. Course Stability of a Ship Towing System. *Ship Technology Research Schiffstechnik*. 58: 4–24.
[2] Sisong, Y, and Genyu, H. 1996. Dynamic Performance of Towing System-Simulation and Model Experiment. *Trans. IEEE*. 1: 216–230.
[3] Yasukawa, H., Hirata, N., Nakamura, N., and Matsumoto, Y. 2006. Simulations of Slewing Motion of a Towed Ship. *Journal of the Japan Society of Naval Architects and Ocean Engineers*. 4: 137–14.