Intuitive operation and pilot training when using marine azimuthing control devices

AZIPILOT

Report Title:

# Deliverable 2.2:

# Review of Existing Ship Simulator Capabilities

# Contents

**PUBLISHABLE EXECUTIVE SUMMARY**


The aim of this task is to survey existing simulator capabilities with respect to the ability to simulate the most common influencing factors that affect ships when operating in close quarters including environmental effects such as effect of proximity of the shoreline, bank effect, effect of proximity of other ships and other effects experienced by ships equipped with azimuting control devices when manoeuvring in their most typical and critical situations.

The report consists of two parts. In Part 1 of the report the hydrodynamic interactions experienced by the manoeuvring ship in shallow water or in the canal, either surface channel or dredged fairway in shallow water area, and their effect on manoeuvring characteristics of ships, either with conventional propulsion or pod driven are discussed. Hydrodynamic reactions between two ships meeting or overtaking each other at close quarters are considered.

The effect of soft or muddy bottom on the required under keel clearance is considered and important results of model tests of ships sailing close to the water-mud interface are quoted drawing attention to the effect of muddy bottom on squat.

Steering of the ship when towing or under tow was considered and because of lack of data regarding pod propelled ship model tests with manned models were arranged where pod driven ship was towing or under tow, the results showing the effect of length of the hawser on directional stability of towed ship, either with conventional propulsion or pod-driven

The report considered also escorting operations performed with pod driven tug where the tug assists braking the escorted ship with rudder blocked or experiencing black-out by forcing sharp turn or keeping the escorted ship on straight course within the limits of the fairway. Some issues related to tug working in the proximity of stern of the ship towed are given in the Appendix.

In the Part 2 capabilities of existing simulators either Full Mission Bridge Simulators or Manned Models Simulators to simulate the effects discussed in the Part 1 of the report are discussed concluding that practically all well advanced simulator centres have the capability to simulate the majority of effects considered. However, the magnitude of these effects may be different in different Full Mission Bridge Simulators. In general, data regarding hydrodynamic characteristics of pod driven ships are rather scarce and further model and full scale tests are required.


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# INTRODUCTION

The aim of this task is to survey existing simulator capabilities with respect to the ability to simulate the typical environment experienced by ships equipped with azimuting control devices when manoeuvring in their most typical and critical situations. This will include consideration of the indirect hydrodynamic interactions experienced due to solid surfaces and other ships and consideration of the direct interactions when under tow or otherwise assisted by vessels equipped with azimuthing control devices. The objectives are to review the most common influencing factors that affect ships when operating in close quarters including typical interaction between target ships (azimuthing or otherwise), within the context of the impact of azimuthing control devices including a:

- **Survey of shallow water effect**
- **Survey of bank effects**
- **Survey of surface and submerged channel effects**
- **Explore ship-to ship interactions**
- **Discuss soft-bottom and mud**
- **Survey of steering with azimuthing control devices when towing**
- **Survey of steering with azimuthing control devices when under tow**
- **Explore issues associated with assisted braking including the indirect mode**
- **Discuss issues related to tugs operating near the stern of pod driven ship**

The task will culminate in a task report that will delineate the above aims and objectives and will constitute one deliverable.

This report consists of two parts;

**Part 1; Survey of influencing factors that affect ships operating at close quarters including typical interaction between target ships, (azimuthing or otherwise) and**

**Part 2; Survey of capabilities of existing simulators, either Full Mission Bridge Simulators (FMBS) or Manned Models Simulators (MMS) to simulate the above effects**

# PART 1

## SURVEY OF INFLUENCING FACTORS THAT AFFECT SHIPS OPERATING AT CLOSE QUARTERS INCLUDING TYPICAL INTERACTION BETWEEN TARGET SHIPS (AZIMUTING OR OTHERWISE)

Review of available sources indicate that there is general lack of information regarding influencing factors that affect ships fitted with azipod propulsion, although there is plenty of information regarding those factors influencing ships with conventional propulsion devices (single or twin screw propulsion). However, as those factors influence mainly hull forces their effect must not be substantially different with regard to ships fitted with azipod propulsion except for those cases where the form of the ship hull fitted with pod propulsion considerably differs from the form of the hull of conventional ships. This may be the case when at the stern of pod driven ships large stabilizing fins are fitted affecting substantially course-keeping ability.

Bearing this in mind the survey presented below concentrated on discussion of the above mentioned effects in relation to all ships, azimuting or otherwise, with special emphasis on pod driven ships in cases where the relevant information is available or where it is anticipated that those effects may be acting differently for ships with conventional and pod propulsion.

## 1.1. Survey of shallow water, bank and channel effects

### 1.1.1. Shallow water and canal effect on resistance and speed

When the ship in moving in shallow water or in a channel, surface or underwater, there are strong interaction effects between the ship and the bottom of the fairway or its banks. Those are complicated hydrodynamic phenomena which affect the flow around the moving ship and influence its propulsive as well as manoeuvring characteristics.

Generally, those effects depend mainly on characteristics of the ship hull, therefore they are equally applicable to ships with different types of propulsors, whether conventional fixed propellers or podded propulsion devices, although shallow water or channel might also influence characteristics of the propulsion devices to some extent. Those influences, however, are of secondary importance and possibly should be taken into consideration only in the detailed analysis of propulsive characteristics of ships operating in restricted waterways and there is no need to consider them in the simulation process of ship handling.

The main effects of shallow water or channel are as follows:

- Resistance of the ship is increasing, causing reduction of ship’s speed
- Ship changes its trim
- Draft of the ship is increasing (squat),
- Manoeuvring characteristics change

Effect of shallow water or canal navigation on ship resistance and trim is as follows: A ship sailing in shallow water or in channel experiences an increased resistance and with the same engine setting a drop of speed. The drop of speed is more pronounced if the clearance between the keel and the bottom reduces. As effect of shallow water on ship resistance in straight line motion is caused mainly by changes in flow around the ship hull there would be no significant changes of this effect if the ship is powered by podded propellers or any other types of propellers.

Usually when the depth of the water exceeds four to five ship drafts, then the increase of the resistance and the drop of speed as well as other phenomena relevant to restricted waters are not observed. PIANC [4] takes

$$h <\approx 3T \qquad \text{where}: h \text{ - depth of water, } T \text{ - draft of ship}$$

or UKC – 200%

as the separation between deep and shallow waters

The curve of the ship resistance versus ship speed reveals a characteristic local maximum at a certain speed (Fig. 2). The speed corresponding to this local maximum is called ***critical speed***. The critical speed in shallow water of unlimited width could be calculated by the formula:

$$v_{crit} = \sqrt{gh} \approx 3.13\sqrt{h} \ \text{[m/s]} = 6.1\sqrt{h} \ \text{[knots]}$$

The critical speed and other phenomena in shallow water are often related to non-dimensional parameter – Froude number relative to water depth:

$$Fr_h = \frac{v}{\sqrt{gh}}$$

Then the critical speed corresponds to $Fr_h = 1$. The critical speed is related to the propagation of waves– it is the maximum speed at which waves can propagate in shallow water.

All phenomena that exists when the ship is sailing in shallow water exist also when the ship is sailing in a fairway with restricted cross-section – a canal or a river. However the phenomena are more pronounced. The critical speed in the canal is similar phenomenon as the critical speed in shallow water. It may be calculated using similar formula, but instead of the water depth, a hydraulic radius of the canal has to be used instead. Hydraulic radius $R_H$ is defined. by formula and as shown in Fig 1

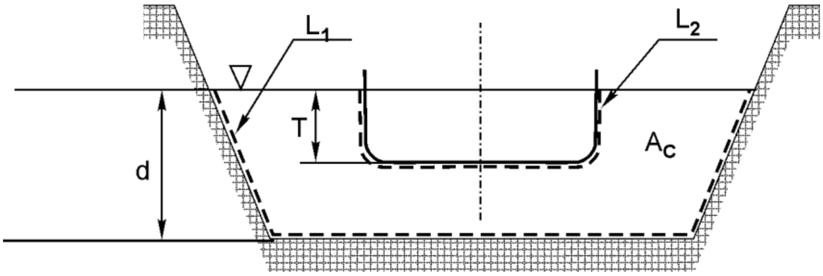



**Fig. 1. Definition of the hydraulic radius of a canal**

$$R_H = \frac{A_C}{L_1 + L_2}$$

With this definition of the hydraulic radius, critical speed in the canal would be:

$$v_{crit} = \sqrt{gR_H} \quad [m/s]$$
$$v_{crit} = 6.1\sqrt{R_H} \quad [knots]$$

The critical speed $v_{crit}$ in the canal is therefore related to the hydraulic radius $R_H$ of the canal.

When the ship is sailing in shallow water or in a canal then its resistance is increasing rapidly with increasing of the speed reaching local maximum at speed roughly corresponding to the critical speed (Fig. 2), therefore usually the sea-going ships could not sail faster than about 60 to 70 percent of the critical speed. Only high-powered small ships can reach the critical speed and exceed it. Once the critical speed is exceeded then the ship resistance could be even smaller than in a deep water, so the ships may accelerate rapidly (see Fig. 2).

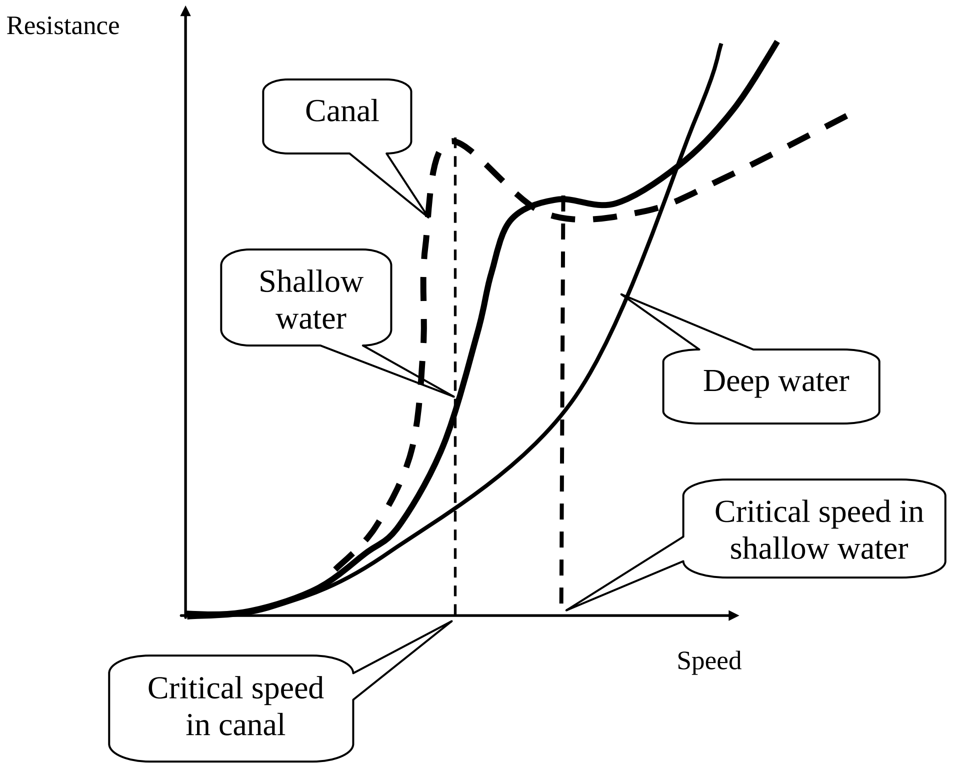



**Fig.2. Resistance curve in deep and shallow water and in the canal**

Fig. 3 shows the effect of the shallow water (two water depths – 12m and 16 m) on the effective power required for the propulsion and the trim for an example RO-RO ship (L=210m, T=9.05). The effect of the increased demand for the effective power when the ship approaches the critical speed is clearly seen. The effective power limits the ship speed to about 18 knots for 16 m depth, and to about 15 knots for the water depth of 12 m.

The above figure shows also the effect of shallow water on trim. Generally, with decreasing depth of the water and when the ship is approaching critical speed it is more trimmed to the stern and trim is increasing. The same happens when the ship is sailing in a canal.

The main parameters governing those effects are:

- depth of the water,
- depth of the water over draft of the ship ratio,
- form of the hull,
- speed of the ship.
- hydraulic radius or blockage coefficient of the canal

Those parameters should be taken into account in the simulation.

Because of restricted cross section of the flow around the ship hull in shallow water or in the canal return or back flow is created. Return or back flow is created in opposing direction to the ship motion in shallow water or in a canal. According to [2] velocity of the return flow could be calculated with the formula:

$$u = \left[ \frac{\frac{z}{h} - \frac{mh}{b}\left(\frac{z}{n}\right)^2 + \frac{1}{n}}{1 - \frac{z}{h} + \frac{mh}{b}\left(\frac{z}{h}\right)^2 - \frac{1}{n}} \right] v$$

Where; $h_C$ (mean depth of the canal , $h_C = A_C / b$

b – width of the canal on the water surface

m – slope of the bank

n - reverse blockage coefficient $n = \frac{A_C - A_S}{A_S}$

z- water level lowering

v- ship speed

The main effect of the back flow is that ship speed against the water is not equal to the ship speed against the bottom of the canal.

Fuehrer and Roemisch [2] proposed different formulea for calculation of the velocity of return flow:

$$u = v\left\{ \left[ 1 + \alpha\left(\varepsilon^2 - 1\right)\left(\frac{0.08 g h_C}{\varepsilon^2 - 1}\right)^{-0.85} v^{1.7} \right]^{0.5} - 1 \right\}$$

Where: ε coefficient

$$\varepsilon = \frac{1}{0.95 - k} \quad \text{and} \quad k = \frac{1}{n}$$

μ – coefficient depending on B/b, where B- witdh of the ship and b – width of the canal

$$\begin{array}{ll} B/b \le 2.5 & \mu = 1 \\ B/b > 2.5 & \mu = 0.114 \cdot B/b + 0.715 \end{array}$$

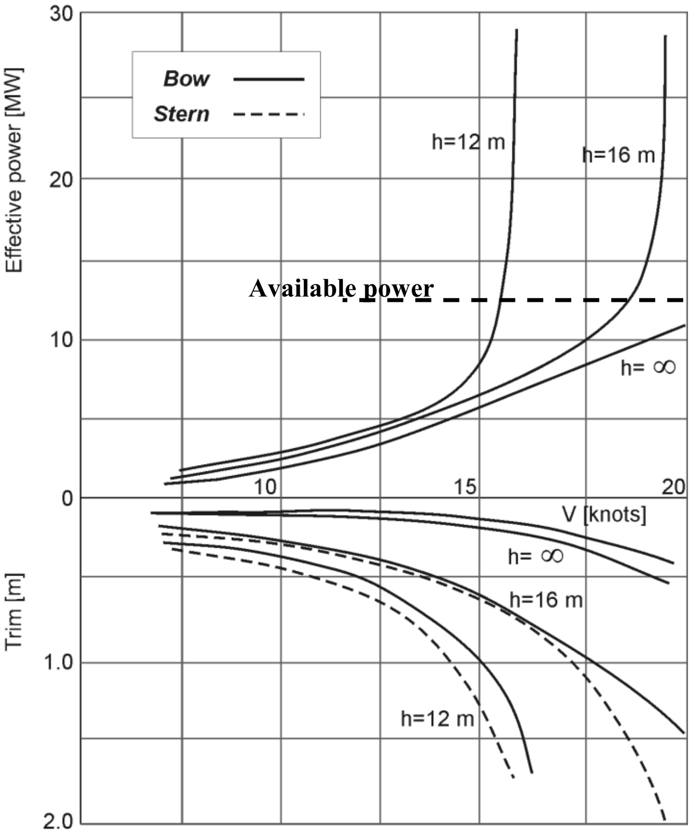



**Fig. 3. Effect of limited water depth on resistance and trim**

### 1.1.2. Bank or wall effect, surface and submerged channel effects

When the ship is moving close to a solid wall or bank or shore line then there is a reduction of the flow cross section area between the ship and the bank. This is result of application of continuity law. On the other side of the ship the flow cross-section area is not reduced and the water velocity does not change (when comparing to the open-water situation). If the water velocity increases, then according to the Bernouilli's law the dynamic pressure increases and in consequence static pressure is reduced causing the water level sinkage. The difference of pressures on both ship sides creates a force that is directed from the higher static pressure area towards the lower static pressure area. This is the suction force drawing the ship closer to the bank. (Fig. 4).

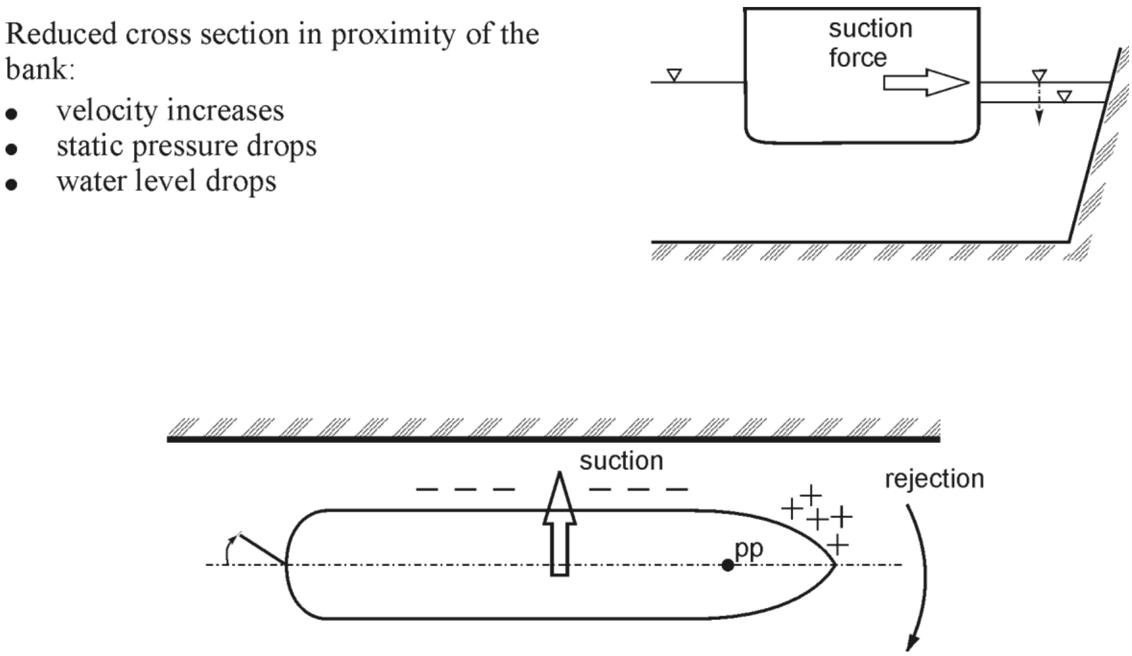



**Fig. 4. Effect of proximity of the bank**

The suction force is proportional to the speed of the ship squared and inversely proportional to the distance from the bank.

The bow of the ship on the other hand is rejected from the bank because of the increased pressure around the bow of the moving ship and proximity of the bank. As a result a lateral or suction force will act on the ship, mostly directed towards the bank and a yawing moment is pushing its bow away from the bank. However, the propeller race caused by the working propeller may affect the flow around the ship in situation when the under keel clearance (UKC) is very small (less than 35%) and the suction force towards the bank may actually became repulsion force pushing the ship away from the bank. Bank effect was investigated by several authors, as for example by Norrbin [7,8], Eloot et al [3], Lataire et al [1,10] and others.

In particular Lataire et al [1, 10] investigated effect of bank slope and of the bank with platform submergence on suction force and yaw moment acting on the passing ship. Table 1 provides overview of the tested banks. They included two types of banks, namely (Fig. 5):

- Surface piercing banks, characterized by a constant slope from the bottom up to the free surface,
- Banks with platform submergence, composed of a sloped part with height $h_o$ and a horizontal submerged platform at a depth $h_1(= h - h_0)$

Bank I is a vertical wall analogous to berthing quay wall. Bank VII is surface piercing with slope of 1/3 which is common slope in man made canals. Slope 1/8 is very common for natural river banks.

Three models were used in the test programme : container, LNG and tanker ship. All tests were executed in the towing tank belonging to Flanders Hydraulics Research- Ghent University.

For each loading condition models were tested in different values of UKC, usually 10, 35 and 100% of the draft.

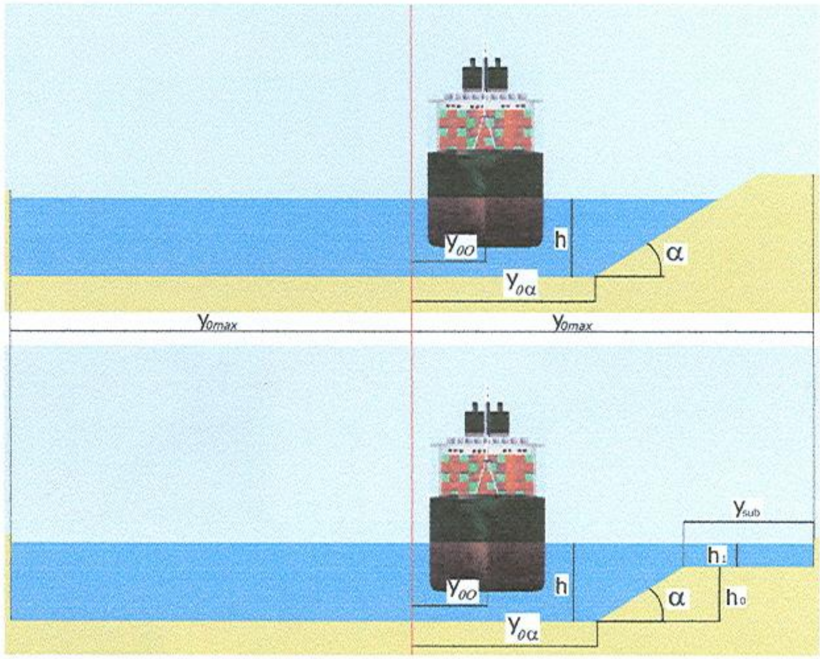



**Fig 5. Surface piercing bank and bank with platform submerged used in the test programme by Lataire et al [10]**

*Table 1. Overview of the tested banks*

| **Name** | $h_0$ | α | define$y_{0\alpha}$ | $y_{sub}$ |
|---|---|---|---|---|
| **I** | Surface piercing | Vertical wall | 2.830m | - |
| **II** | Surface piercing | 1/5 | 0.530m | - |
| **III** | 0.120m | 1/5 | 0.530m | 2.370m |
| **IV** | 0.150m | 1/5 | 0.530m | 2.220m |
| **V** | Surface piercing | 1/8 | 0.530m | - |
| **VI** | 0.150m | 1/8 | 0.530m | 1.770m |
| **VII** | Surface piercing | 1/3 | 2.230m | - |

Bank or wall effect will only be felt if the distance between the ship and bank is sufficiently small. The distance at which bank effect is practically felt may be defined as the boundary between restricted and unrestricted water. This distance is defined as horizontal reach. (Lataire [10]). Systematic tests conducted by Lataire [10] allowed to develop expression for horizontal reach, that depends on ship speed. Those tests were performed with a tanker model at combination of speed and water depth and the results were plotted as shown in Fig.6. In this figure three ranges were determined:

- If the distance to the bank is sufficiently large, the influence of the closest bank on the ship is negligible (△)
- Close to the bank, a significant influence is generated ( □ )
- In between, the influence is measurable, but not significant (◇) )

The division in three ranges was carried out for all UKC-speed combinations and shown in (Fig.6) as a function of the non-dimensional parameter defining the distance between the bank and the ship's side relative to the ship's beam and the water depth related to Froude number. The dividing line between combinations with significant influence and without significant influence shows dependency on the Froude number related to water depth:

$$y_{\inf l} = B(5Fr_h + 5)$$

The above value may be considered as the half width of the influence zone for bank effects.

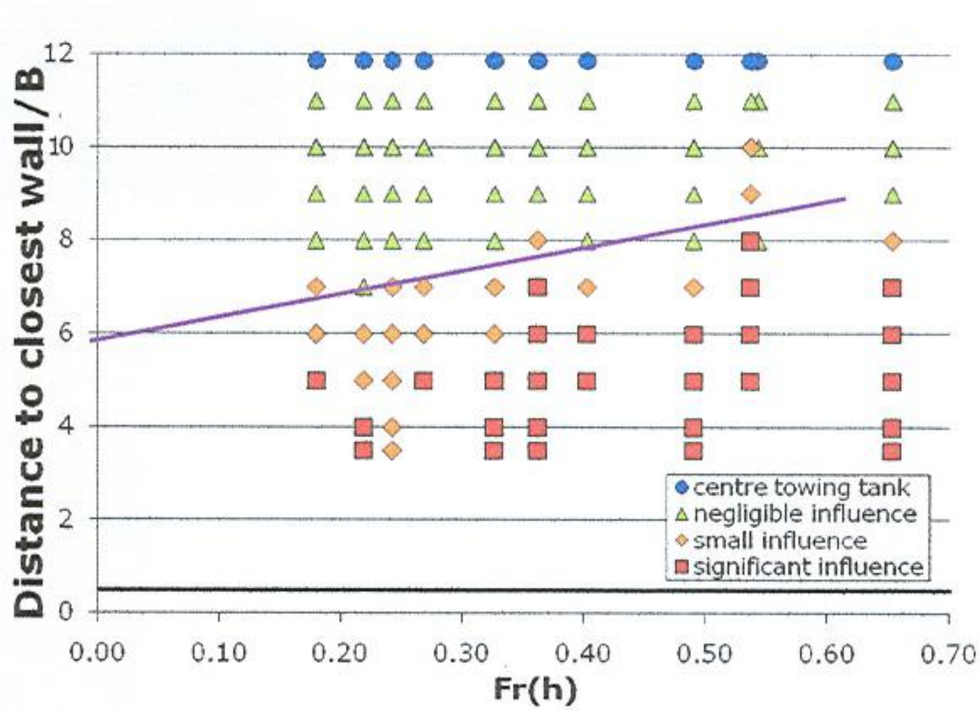



**Fig.6 Horizontal reach and $Fr_{(h)}$ with the three influence regions (Lataire et al [10])**

Fig. 7 shows the relation between the distance between the ship and the bank and the suction (sway) force for seven different bank geometries tested by Lataire and al [1, 10] for the container ship at 10 knots speed (full scale), with 100% UKC .The distance from the bank is defined at half draft of the ship as shown in Fig.8.

Lataire et al [1, 10] presented also the mathematical model simulating the above described phenomena, however this model has severe limitations. The model is not valid for extreme situations where the ship is very close to quay or wall or where it is going aground. The mathematical model is based on the results of model tests where the ship is moving forward with positive propeller revolutions (quadrant 1) therefore is not applicable to manoeuvring situations and therefore only with limited possibilities to be used in manoeuvring simulators.

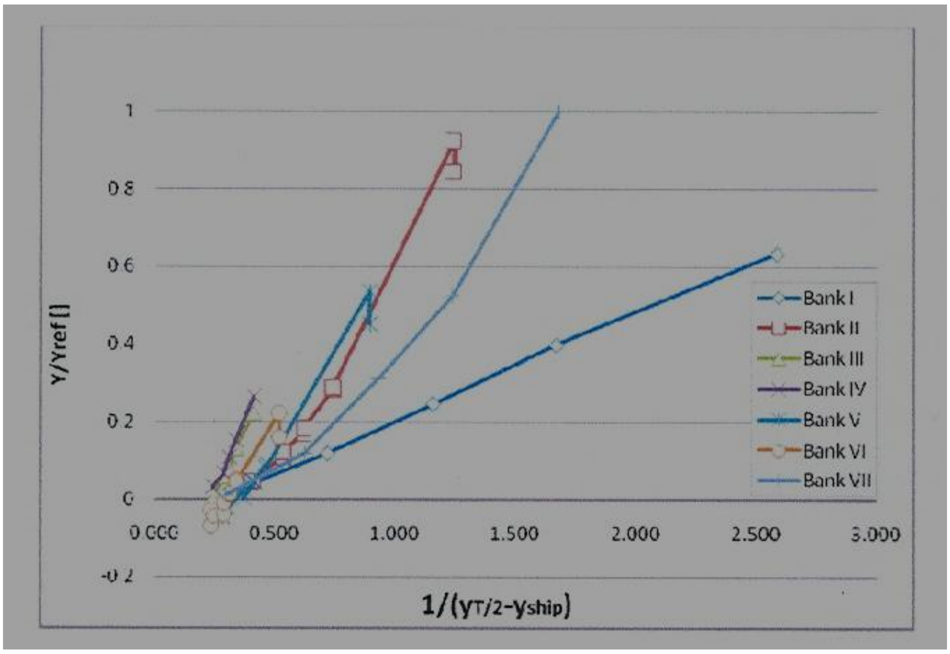



**Fig. 7. Suction (sway) force as a function of lateral distance from the bank. (Lataire et al [10])**

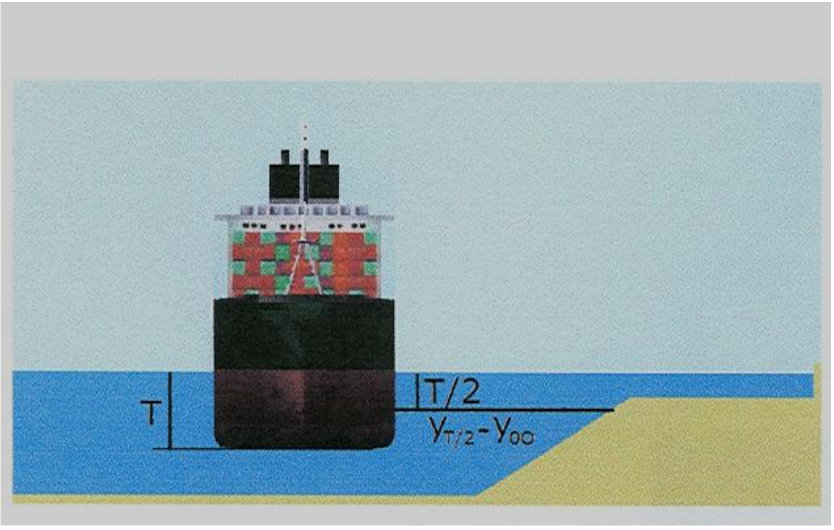



**Fig.8. The distance between the centre of the ship and the bank at half draft (Lataire et al [10])**

All the above considerations apply to ships fitted with conventional propulsion. There are virtually no data available with regard to pod driven ships. The forward rotating propeller changes the flow pattern around the ship body and different propulsion devices may affect those changes to some extent, although it seems that this effect is small and possibly even negligible. There are, however, no data available to proof this conclusion.

In the mathematical model developed by Lataire et al [10] calculation of speed increase is based on propeller thrust and propeller diameter and for pod driven ships the same method could be used

### 1.1.3. Manoeuvrability of a ship navigating in shallow water

It is well known fact that shallow water affects considerably manoeuvring characteristics of ships. First of all the parameter influencing this effect is Froude number relative to water depth

$$Fr_h = \frac{v}{\sqrt{gh}}$$

And the other parameter is $h/T$ .

If $Fr_h < 0.6$ or $h/T > 3$ the effect of shallow water on manoeuvring characteristics of ships is negligible and the water may be considered deep..

Otherwise in shallow water all manoeuvring characteristics of the ship are affected. Resistance of the ship increases, and with the same number of revolution of the engine ship speed drops down and the loading of the propeller increases. Therefore all hydrodynamic coefficients in equations describing motion of the ship including added mass, characteristics of the propulsion device in straight line and curvilinear motion, ship-propulsor interaction characteristics change. In reality all hydrodynamic coefficients increase and in particular damping coefficients increase considerably with reduction of the parameter $h/T$ or increase of the Froude number $Fr_h$

In general the most remarkable effect of shallow water is on turning characteristics. Turning circles become larger in a shallow water, because the hull yaw damping moment becomes larger, while the rudder moment almost does not change, so the drift angle decreases considerably in shallow water.

Decrease of the drift angle in shallow water results in reduction of speed of the ship when turning.

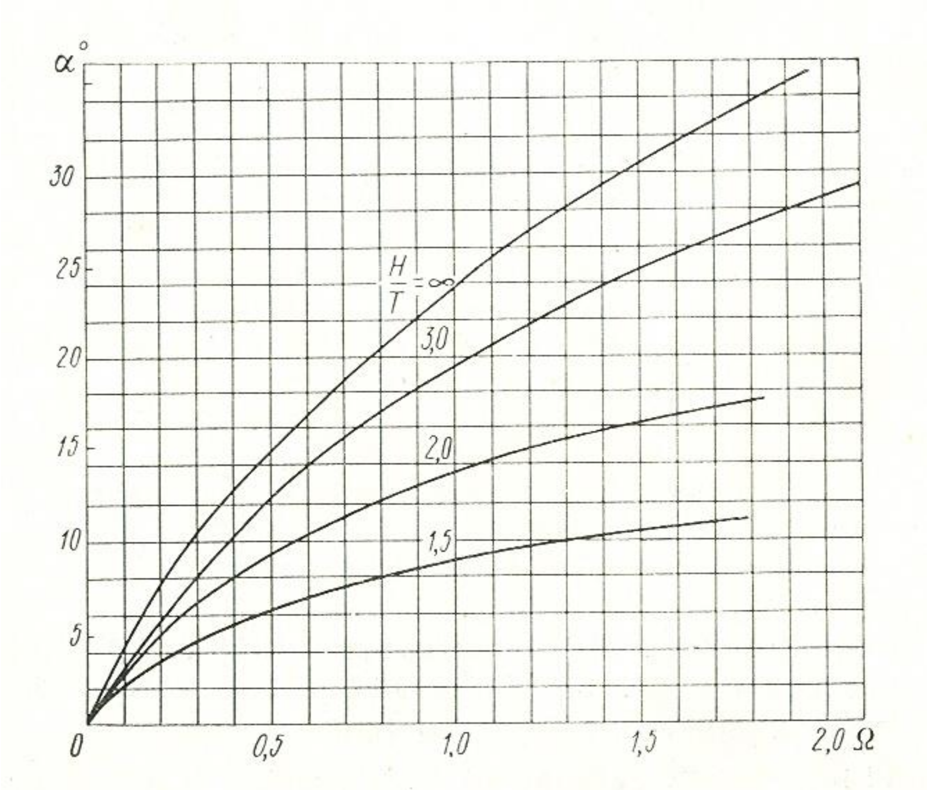



**Fig. 9. Drift angles versus parameter Ω for different h/T ratios. From Gofman [5]. (Remark: in this diagram α is**

**drift angle).**

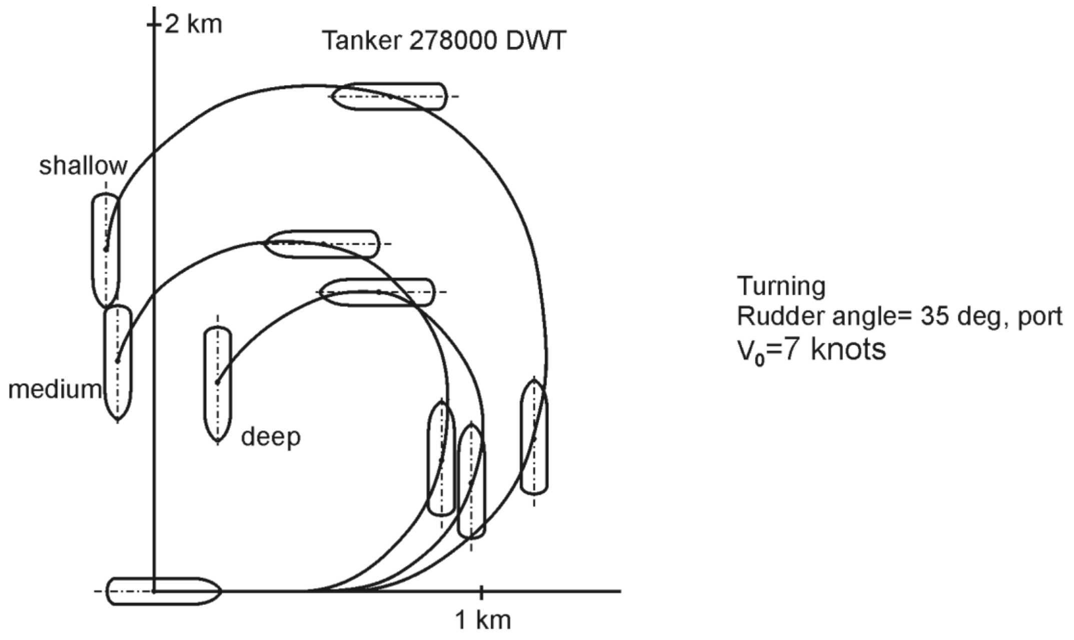



**Fig.10. Turning circles in deep and shallow water. Tanker 278 000 tdw, full-scale tests**

This effect is seen in the diagram in Fig. 9. (from Gofman [5]) where the drift angles for different ratios $h/T$ are plotted against the parameter $\Omega = 2\frac{L}{D}$. for the inland waterways ship used on Wolga river. From the diagram it is seen that for the same value of Ω drift angle is smaller with smaller values of $h/T$ .

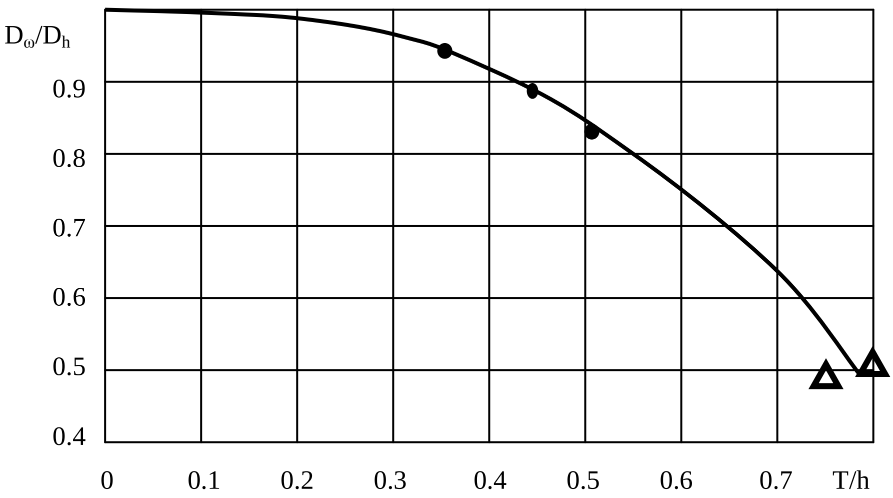



**Fig. 11. Effect of shallow water on turning circle diameter**

**● ship trials Δ model tests**

The effect of shallow water on turning characteristics is illustrated also in fig. 10 where results of turning circles tests of the full-scale tanker 278 000 tdw in deep and shallow water of different depths are shown.

Gofman in his book [5] provided also the diagram (Fig.11) showing the relation between the ratio of the turning diameter in deep and shallow water $D_\infty / D_h$ and the water/depth ratio $T/h$

The diagram shows how with increasing ship draft to water depth ratio (i.e. with decreasing water depth) turning circles are increasing (i.e. ratio $D_\infty / D_h$ is decreasing). This diagram may be used for the purpose of assessing this effect.

Fig .12. (from Gofman [5]) shows relation between the parameter $\Omega = 2\frac{L}{D}$ and rudder angle (in this diagram marked β) for different h/T ratios for an directionally unstable ship. In this diagram increase of turning diameter in shallow water with decreased depth for the same rudder angle is clearly seen.

The initial part of the diagram shown if Fig. 12 (for small rudder angles) allows to reach conclusion, that degree of instability in shallow water increases, therefore course keeping ability also decreases. This in a way is contradicting the common rule, that with the decrease of turning ability the course keeping ability increases. Apparently this is not so in shallow water.

There is little information available on the effect of shallow water on stopping ability. Fig 13 shows results of full-scale tests of stopping ability of a tanker 278 000 tdw (the same as in Fig. 10) in deep and shallow water. It may be concluded that stopping ability (head reach in crash stop) may be slightly larger with decreased water depth/ship draft ratio.

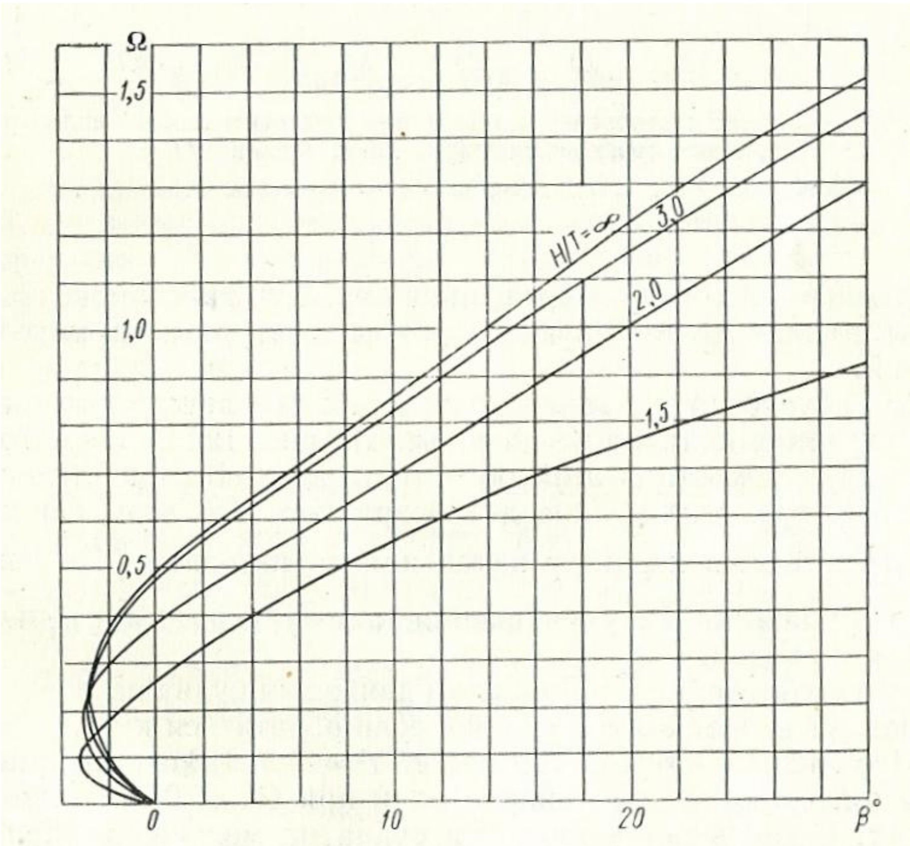



**Fig. 12. Turning diameter parameter versus rudder angle for different h/T ratios.**

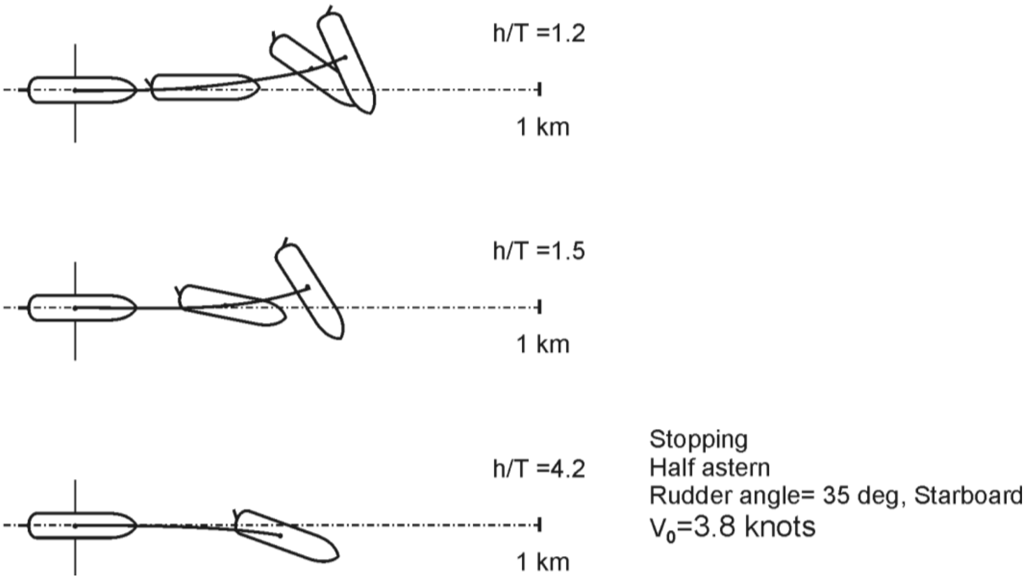



**Fig. 13. Stopping distances in different water depths. Full-scale tests. tanker 278 000 tdw**

Data on the effect of shallow water on manoeuvring characteristics of pod driven ships based on model tests or on tests of full-scale ships were not found, but there are available some data based on simulation. Table 3 shows comparison of basic manoeuvring parameters for four pod driven cruise liners in deep and shallow water simulated by TRANSAS (QUEEN MARY 2, RADIANCE OF THE SEAS, LIRICA AND VOYAGER OF THE SEAS) . The basic data for ships considered are included in Table 2.

*Table 2. Basic data for simulated ships*

| Hull and Pod parameters | Quen Mary 2 | SHIP 1 Radiance of the Seas | SHIP 2 Lirica | SHIP 3 Voyager of the Seas |
|---|---|---|---|---|
| Displacement [t] | 76499 | 44809 | 29060 | 64220 |
| $L_{PP}$ [m] | 344 | 263.5 | 222.3 | 297.2 |
| B [m] | 41 | 32.2 | 28.8 | 38.6 |
| T [m] | 10 | 8.15 | 6.8 | 8.6 |
| Number of pods | 4 | 2 | 2 | 3 |
| Design speed [kn] | | 24.5 | 21.9 | 23 |
| Shallow water h/T=1.5 | 15m | 12.23m | 10.2m | 12.9m |

Few data from simulator trials with the model of Lirica ship performed by Maritime Institute of Technology, Linthicum Heights, MD are also available., and excerpt of some data from the full report of these tests where comparison between some manoeuvring characteristics in deep and shallow water was possible are shown in Table 4.

Comparison of the results achieved by simulation for deep and shallow water reveals that at least in one case of Queen Mary 2 ship the turning circle characteristics for 35 deg rudder (advance, transfer and tactical diameter) in shallow water are actually smaller than in deep water. This is in contradiction to the general rule that turning characteristics in shallow water are worse than in deep water. The reason for this discrepancy is not clear and impossible to explain, because only simulation data are available and they cannot be compared with results of full scale or model tests.

The large discrepancy of result of simulated zig-zag tests for the ship LIRICA by TRANSAS and MIT should be also noted. In general, however, the overshoot angles in zig-zag simulation tests in shallow water are larger than in deep water which is in accordance with the general rule that course keeping ability in shallow water is worse than in deep water. This rule may be then equally applicable with regard to ships with conventional propulsion as well as to pod driven ships.

*Table 3.Comparison of simulated manoeuvring characteristics in deep and shallow water for four ships shown in Table 2.*

| | | Quen Mary 2 | | SHIP 1 | | SHIP 2 | | SHIP 3 | |
|---|---|---|---|---|---|---|---|---|---|
| | | Deep | Shallow h=15m | Deep | Shallow h=12.23 m | Deep | Shallow h=10.2m | Deep | Shallow h=12.9m |
| Turning circle $35^0$ STB | Advance | 927.2 | 863.5 | 700.0 | 827.0 | 491.5 | 533.1 | 1019.3 | 1140.7 |
| | Transfer | 383.6 | 431.5 | 270.0 | 462.8 | 229.7 | 281.0 | 438.5 | 553.6 |
| | Tact. diameter | 810.6 | 820.0 | 580.0 | 890.1 | 544.6 | 621.8 | 1106.1 | 1278.8 |
| Turning circle $20^0$ STB | Advance | 1430.5 | 1429.0 | 964.0 | 1300.7 | 676.4 | 758.6 | 1289.9 | 1527.6 |
| | Transfer | 849.0 | 999.0 | 473.0 | 918.4 | 406.8 | 495.6 | 678.8 | 893.8 |
| | Tact. diameter | 1713.1 | 2048.0 | 922.0 | 1880.6 | 942.1 | 1081.3 | 1675.3 | 2020.6 |
| Accel turn | Advance | 173.0 | 158.0 | 141.0 | 133.0 | 144.3 | 141.0 | 281.9 | 275.6 |
| | Transfer | 84.1 | 60.0 | 67.0 | 54.6 | 77.6 | 72.1 | 128.6 | 118.8 |
| | Tact. diameter | 337.0 | 253.0 | 265.0 | 221.8 | 305.0 | 287.4 | 593.9 | 557.9 |
| Z/Z 10/10 | 1st overshoot[$^0$] | 1.33 | 0.68 | 3.0 | 0.97 | 2.79 | 1.75 | 5.74 | 2.13 |
| | 2nd overshoot[$^0$] | 1.70 | 0.95 | 4.66 | 1.17 | 3.16 | 2.31 | 10.05 | 3.71 |
| Z/Z 20/20 | 1st overshoot[$^0$] | 3.98 | 2.09 | 7.6 | 2.7 | 5.95 | 4.88 | 8.36 | 5.38 |
| | 2nd overshoot[$^0$] | 5.33 | 2.63 | 10.9 | 3.22 | 5.97 | 5.39 | 7.06 | 4.89 |
| Initial. turning | Head reach | 703.0 | 658 | 500.6 | 606 | 268.8 | 313.6 | 508.1 | 645.7 |
| Pull-out | Stable/not | Yes | Yes | Yes | Yes | Yes | Yes | No | Yes |

In simulation of manoeuvring characteristics of a ship in shallow water the effect of UKC is mostly taken into account by adding UKC related coefficients to the set of hydrodynamic derivatives valid for deep water.

In the most common MMG mmathematical model the effect of shallow water (UKC) may be taken into account using the following expression proposed by Kijima [24]:

$$D_{shallow} = f\left(\frac{h}{T}\right)D_{deep}$$

Here D= a hydrodynamic derivative and

f =correction factor

$$f\left(\frac{h}{T}\right) = \frac{1}{\left[1-\frac{T}{h}\right]^{n}}\frac{T}{h}$$

Where the exponent n is function of ship hull parameters.

*Table 4. Some simulated manoeuvring characteristics for pod driven ship LIRICA*

| Test | Particulars | Deep water | | Shallow water | |
|---|---|---|---|---|---|
| Acceleration test | Full ahead up to 170 RPM | Advance 5895m | Speed 21.5 kn | Advance 7989m | Speed 18 kn |
| | Half ahead up to 120 RPM | Advance 3223m | Speed 15.6 kn | Advance 2360 m | Speed 11.3 kn |
| Crash stop | | Head reach 1930m | | Head reach 2310m | |
| Turning circle STB | | Advance: 614<br>Transfer: 774<br>Tactical Diameter: 719m | | Advance: 678<br>Transfer: 934<br>Tactical Diameter: 860m | |
| Zig/zag testt | $10/10^0$ | 1st overshoot $6^0$<br>$2^{nd}$ overshoot $9^0$ | | | |
| | $20/20^0$ | 1st overshoot $11^0$<br>$2^{nd}$ overshoot $15^0$ | | | |

Li and Wu [25] proposed formula for added inertia coefficients in shallow water:

$$\frac{m_{22\,shallow}}{m_{22deep}} = 1 + \frac{f\left(\frac{B}{T}\right)}{\left(\frac{H}{T}-1\right)^{0.82}}$$

Gronarz (26) expressed the shallow water effect on the hydrodynamic derivatives as follows:

$$f = C_0 + C_n\left(\frac{T}{h}\right)^n$$

Where $C_0$, $C_n$ and n must be determined experimentally.

### 1.1.4. Manoeuvrability of a ship navigating in a canal

Moving in the centreline of the canal. Saturation speed.

In the narrow canal or in the river the bottom and the banks restrict the flow around the hull and, as a consequence, the ship squats closer to the bottom than in shallow water only (without side restrictions) and suction forces act on both sides of the ship (Fig.14). Large waves are formed around the ship if the ship sails closer to the critical speed.

The critical speed in a canal is equal to:

$$v_{crit} = \sqrt{gR_H}$$

Where: $R_H$ hydraulic radius (see Fig.1).

Then the parameter influencing manoeuvrability in the canal is Froude number relative to the hydraulic radius

$$Fr_C = \frac{v}{\sqrt{gR_H}}$$

And the other parameter is blockage coefficient defined as:

$$F_B = \frac{A_S}{A_C} \quad or \quad F_{B1} = \frac{A_S}{A_C - A_S}$$

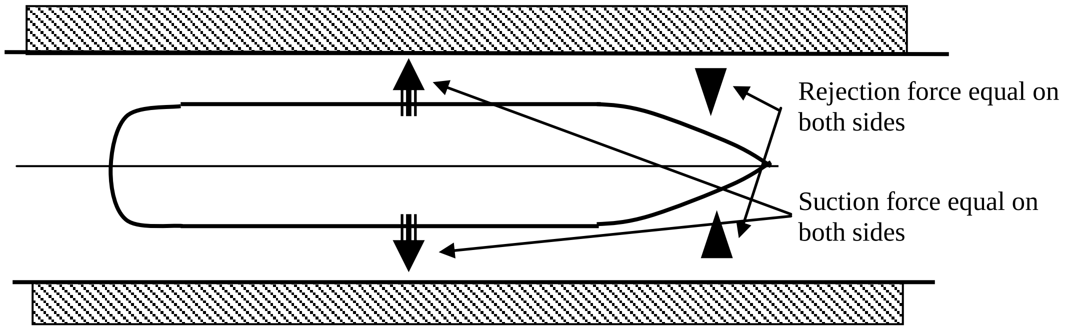



**Fig.14. Rejection and suction forces for the ship sailing in the canal**

Apart from the critical speed in the canal, that is related to the dimensions of the canal two other speed may be defined: saturation speed and sustainable speed. If the ship is moving in the canal too fast, then the bow wave becomes more steeper and the wake larger. If the bow is pushed away from the starboard side of the canal, introducing the yaw to port, then even a full rudder to starboard might be not sufficient to stop the sheer. The bow has the tendency to be sucked toward the port side and stern is sucked to the starboard side, increasing the sheer. The vessel comes across the canal and it will most probably go aground on a port shoal or her stern will hit the starboard bank. (Fig.15).

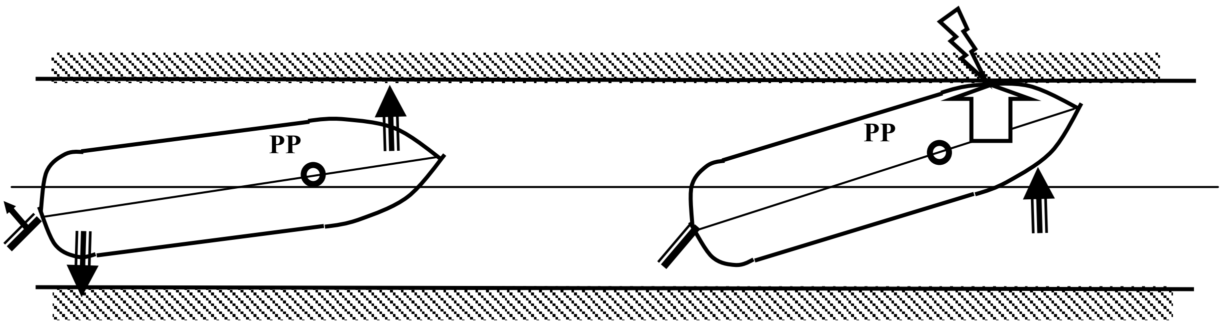



**Fig.15. Behaviour of the vessel at saturation speed.**

This happens at saturation speed i.e. at speed at which the ship becomes uncontrollable due to the repulsion force of the bow cushion and stern suction force

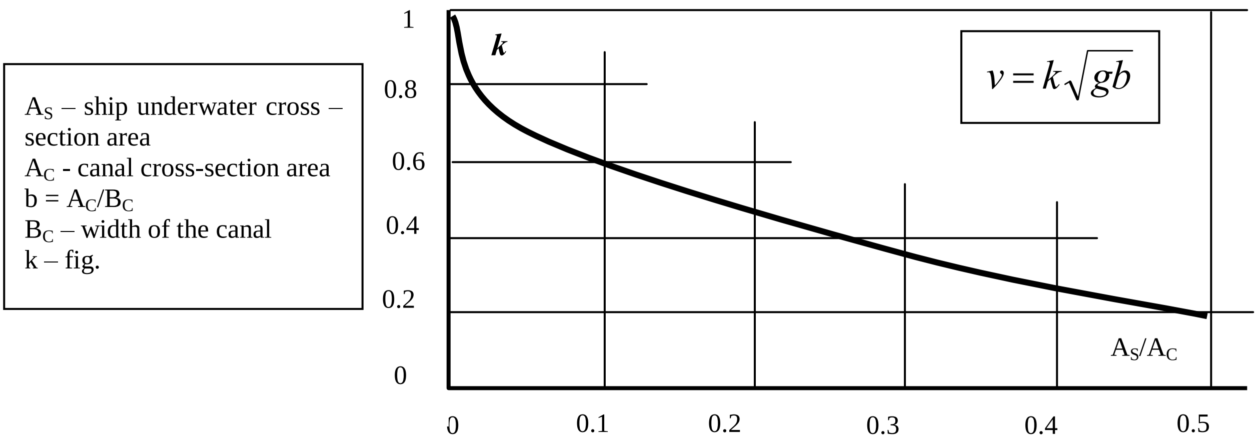



**Fig.16. Diagram for estimating coefficient *k***

Saturation speed may be calculated by the formula:

$$v_{sat} = k\sqrt{gb}$$

Where the coefficient k could be taken from the diagram shown in Fig.16.

Sustainable speed in the canal is the speed at which sharp increase of resistance starts. It is usually estimated from the diagram proposed by Schijf (see Kulczyk [6 ]).

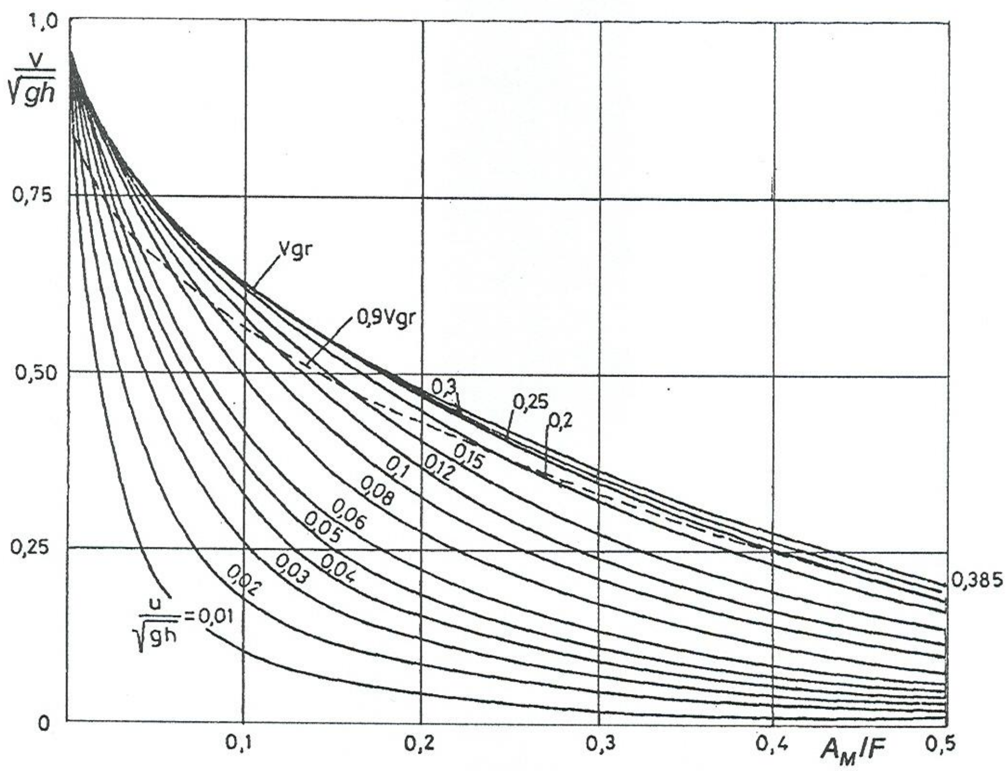
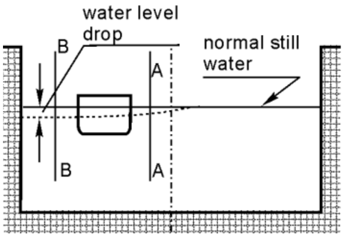
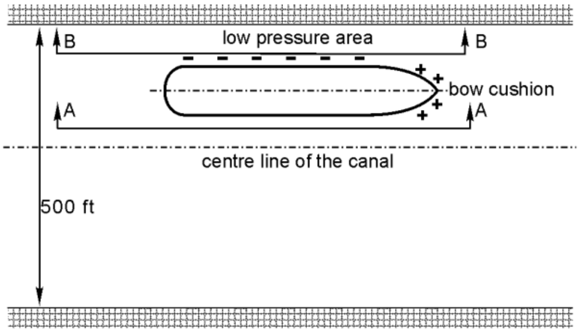



**Fig 16 Diagram for estimating sustainable speed in a canal.**

<u>Moving off-centre of the canal</u>

When the ship is moving off-centre (hydraulic) of the canal, closer to one canal's side, a low pressure area is created between the bank and the ship. The water level drops down - more in the space that is closer to the bank and less on the other side of the ship as shown in Fig. 17. Suction forces are now non-symmetric and the rudder has to be used to counter the swing. Those effects are discussed under the heading "Bank effect"



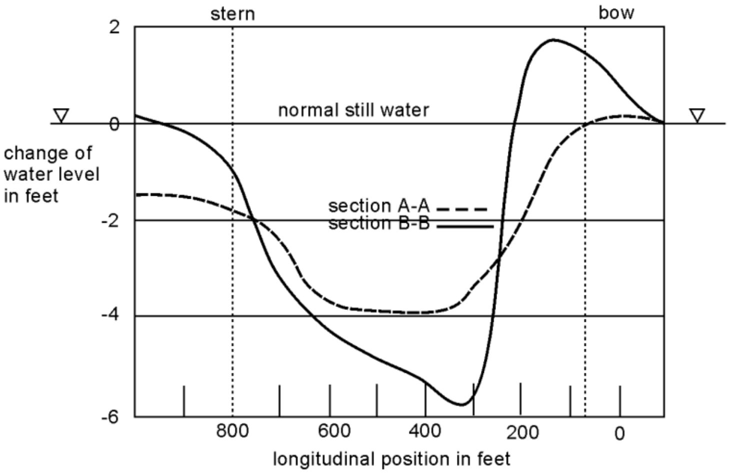




**Fig. 17. Ship moving off-centre of a canal**

## 1.2 Survey of ship to ship interaction

### 1.2.1. Interaction in meeting or overtaking manoeuvre

Strong interaction effects occur between two ship meeting or overtaking each other. When two ships are close together, either on the same course or in the opposite course, there is a restricted space between them. In this space there is an accelerated flow and in consequence the fluid and static pressure drop. When moving on the same course suction forces tend to bring the ships closer, but the bow cushions have the tendency to push the bows apart. This is shown diagrammatically in fig. 5-8.

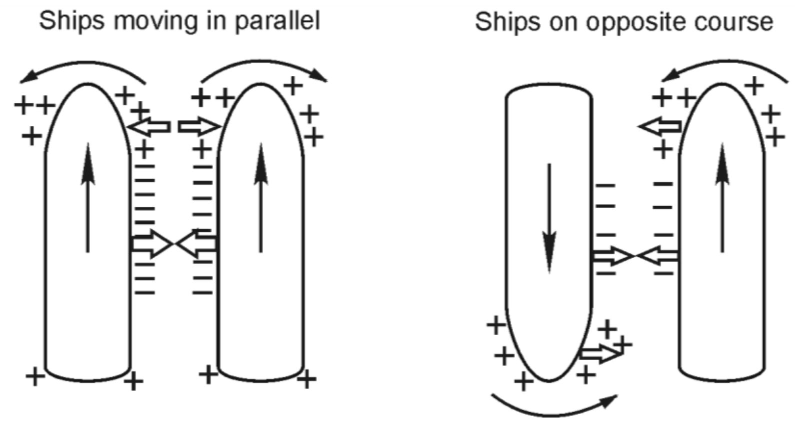



**Fig.18. Sketch showing interaction forced in meeting and passing manoeuvre**

Passing manoeuvre between two ships is rather safe because the passing time is comparatively short and because of inertia the suction forces are little time to develop shifting of the ship and, moreover, in the last phase of this manoeuvre suction forces tends to bring the ship to the original course. Overtaking manoeuvre is more risky because of the longer time when both ships are in parallel, particularly when the difference between speeds of both ships is small. Because of that all interaction forces have enough time to develop.

The hydrodynamic interaction forces develop also when one ship is passing another ship moored and in general those interactions have large impact on behavior of both ship. The interaction problem was studied by several authors, and some references to their work are given below:

Dand [17]. has produced a large data base of experimental data regarding interaction forces and moments between two ships in a channel. Data on hydrodynamic forces acting between moving ship and stationary ship in a shallow canal were provided also by Kyulevcheliev [20]. He investigated experimentally influence of speed, influence of depth of the canal, influence of spacing of both ships and also wave effects. Fig 19 shows the experimental set up and Fig 20 shows the effect of spacing on the interaction forces. of one example of recorded forces

Theoretical investigation using modelling the flow and computation pressures around passing ships in narrow channel were conducted by Spencer et al [21] with the purpose to improve simulator programs. Beck et al. [18] considered the case of interaction in the dredged channel surrounded on both sides by shallow water. Kijima. and Yasukawa [19] investigated the behaviour of ships during meeting and passing in a narrow channel using slender body theory

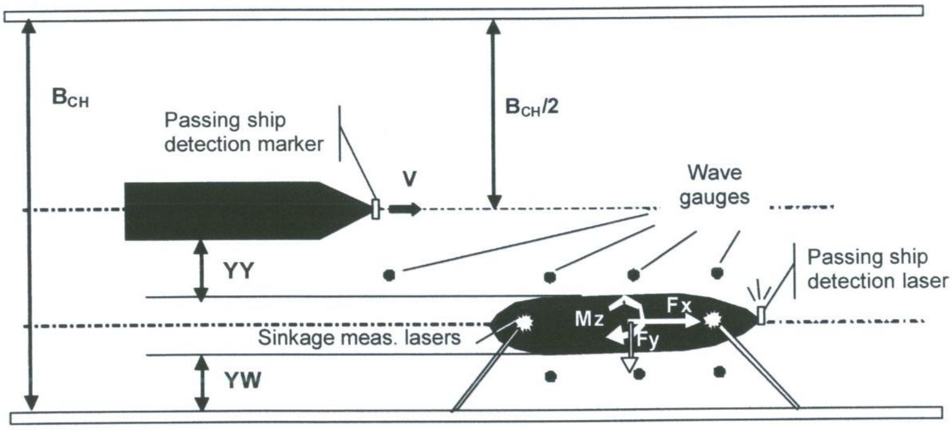



**Fig.19. Experimental set-up for investigating firces in overtaking situation.**

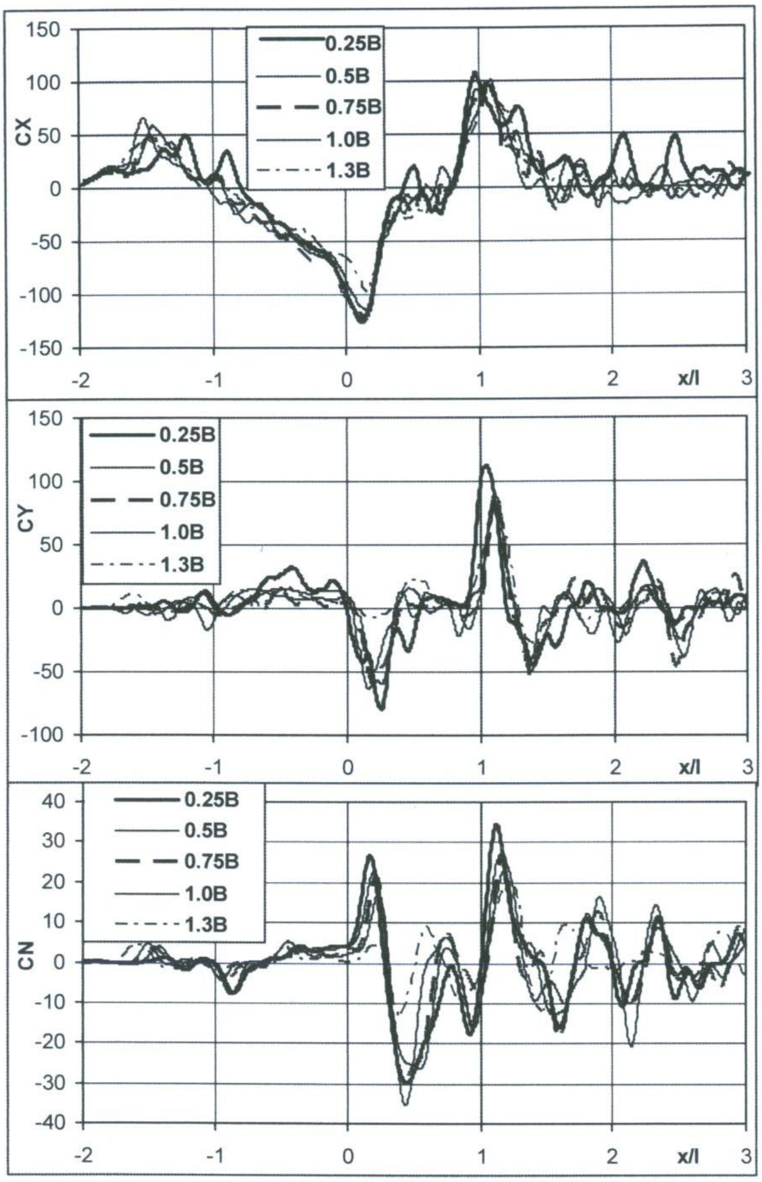



**Fig.20. Hull spacing effect on hydrodynamical loads (Kyulevcheliev [20])**

Varyani et al [15,16] developed empirical model and formulae for calculation of maximum peak of sway forces and yaw moments in overtaking or meeting manoeuvre in a channel. The basic formulae are repeated here. Co-ordinate system and nomenclature is shown in Fig 21.

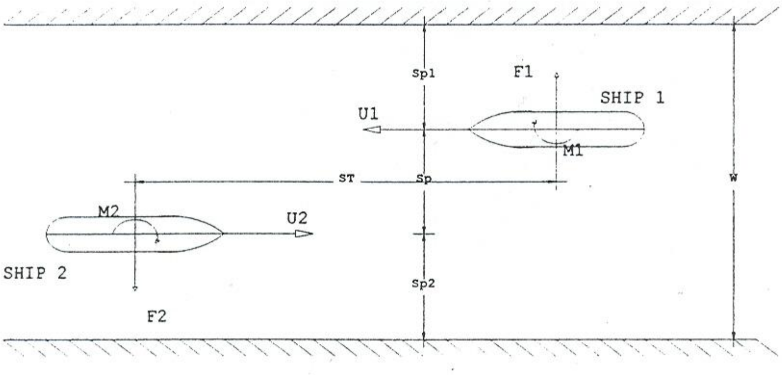



**Fig. 21. Coordinate system for two ships meeting**

The maximum repulsion force coefficients used at the bow-bow situation for a given Sp/L is:

$$C_{F\ Bow-Bow} = 1.2\left(1+\frac{Sp}{L}\right)^{-5.5}\left[1-0.85\frac{D}{H}\right]^{-0.9}\left(\frac{H}{D}\right)^{-0.9}$$

Where:

D- draught

H-depth of the water

Maximum attraction force coefficient at the midhip-midship situation and at the stern-stern situation are:

$$C_{F\ Mid-Mid} = -2.0\left(1+\frac{Sp}{L}\right)^{-4.8}\left[1-0.85\frac{D}{H}\right]^{-0.96}\left(\frac{H}{D}\right)^{-.0.96}$$

$$C_{F\ Ster-Stern} = 1.01\left(1+\frac{Sp}{L}\right)^{-6.0}\left[1-0.85\frac{D}{H}\right]^{-0.94}\left(\frac{H}{D}\right)^{-0.94}$$

Maximum yawing moment coefficient for bow-bow situation is:

$$C_{M\ Bow-Bow} = 0.305\left(1+\frac{Sp}{L}\right)^{-5.0}\left[1-0.85\frac{D}{H}\right]^{-0.75}\left(\frac{H}{D}\right)^{-0.75}$$

Maximum bow-in moment coefficient just before the midship situation is:

$$C_{M\ Fore-Fore} = -0.81\left(1+\frac{Sp}{L}\right)^{-8.0}\left(\frac{H}{D}\right)^{-1.0}$$

Maximum bow-out moment coefficient immediately after the midship-midship situation and at the end of encounter in stern-stern situation:

$$C_{M\ Aft-Aft} = 0.95\left(1+\frac{Sp}{L}\right)^{-10.0}\left(\frac{H}{D}\right)^{-1.2}$$

$$C_{M\ Stern-Stern} = -021\left(1+\frac{Sp}{L}\right)^{-5.0}\left[1-0.85\frac{D}{H}\right]^{-0.9}\left(\frac{H}{D}\right)^{-0.9}$$

All the above considerations refer to interaction phenomena occurring to ships with conventional propulsion. Data that refer to interaction effects between pod driven ships cannot be found. However, as interaction effects are caused mainly by disturbance of the pressure field around the ship body it may be concluded that the results are equally applicable also to pod driven ships.

A semi-empirical approach to estimating the suction forces and yawing moments acting in the overtaking manoeuvre was developed by Brix and Kleinwachter This method is presented in the book by Brix [11]. The short description of the method is presented here.

If one ship is overtaking the other ship at close distance, they are during some time on parallel courses and the pressure field around both ship is changing. This leads to developing suction forces $X_S$, $Y_S$ and Yawing moment $N_S$ as shown in Fig.22.

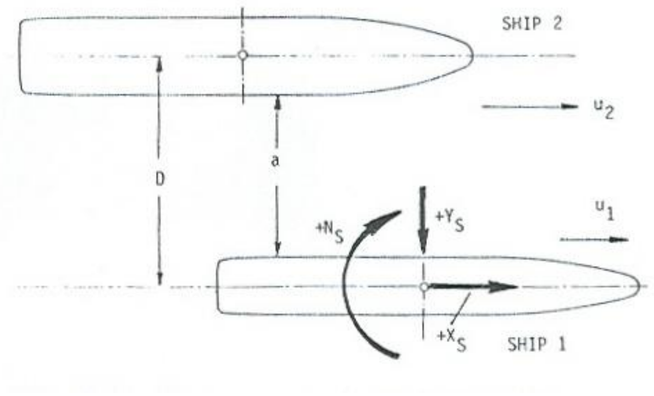



**Fig.22. Forces and moment components in overtaking manoeuvre**

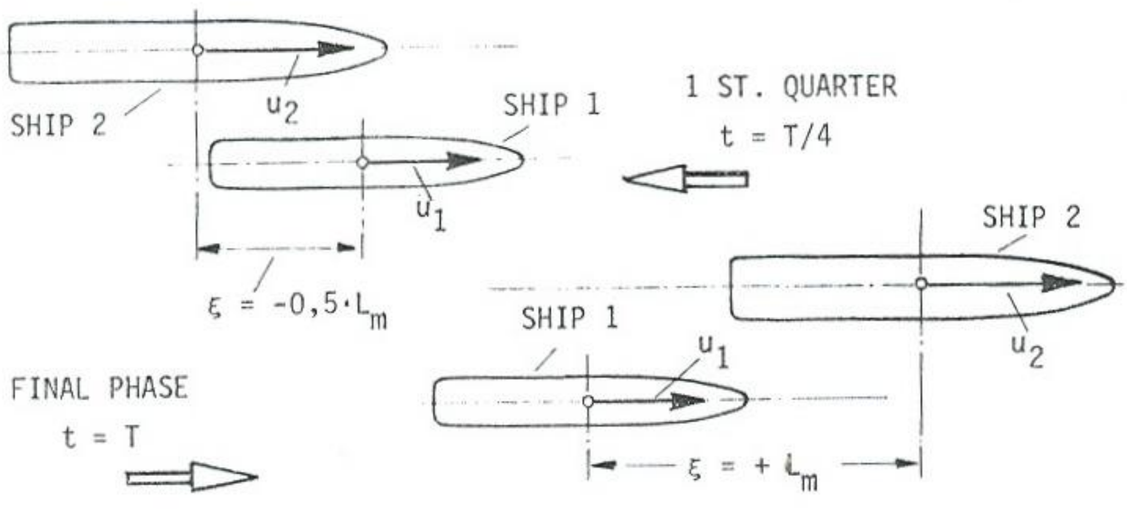



**Fig.23. Different phases of the overtaking manoeuvre. ξ – main section distance of ships 1 and 2**

Normally these interaction forces are measures in the towing tank model experiments where models of both ships are in a staggered position as shown in Fig 23.

Brix [11] recommends the following semi-empirical procedure for estimating those force components that was calibrated by extensive model tests.

If the ship lengths of both ships are not very different one may use the mean length

$$L_M = \frac{L_1 + L_2}{2}$$

And mean draught:

$$T_M = \frac{T_1 + T_2}{2}$$

The mean velocity is:

$$u_M = \frac{u_1 + u_2}{2}$$

The reference passing distance (see Fig 24) is:

$$D = a + \frac{B_1 + B_2}{2}$$

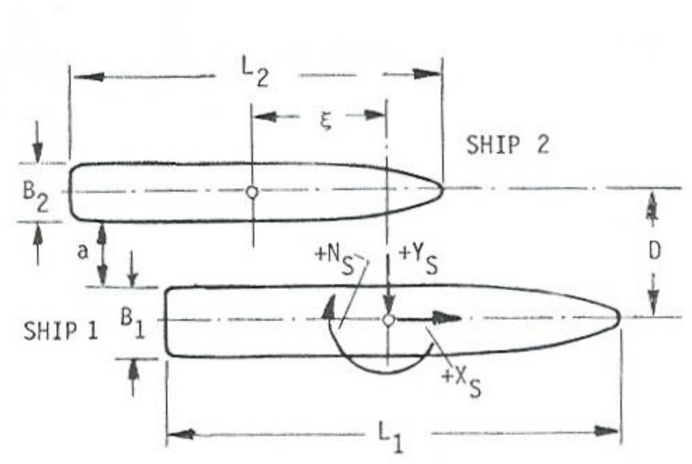



**Fig. 24 Definition sketch**

Coefficients for a reference to centerlines passing distance of

$D_0 = 0.35 \cdot L_M$ : $C_{XS}$max=0.014…..0.017

$C_{YS}$max=0.025 0.030

$C_{NS}$max=0.004 0.005

The reference ship data are $L_M$•$T_M$ for the forces and $L_M^2$•$T_M$ for the yawing moment. The smaller values to be used for $L_2/L_1$>~1.5 and $D_0$ the reference distance (see Table 5). The influence of various parameters and the centerlines spacing D may be drawn from Table 6 , geometry and sign convention from Fig. 24..

Generally suction forces $Y_S$ and yawing moments $N_S$ influence the overtaken ship to a larger amount than the overtaking ship.

In shallow water the forces and moments considerably increase.
The maximum values of the longitudinal force $\tilde{X}_S$max, the transverse force $\tilde{Y}_S$max and the yawing moment $\tilde{Y}_S$ max derived from previous equations by:

*'Table 5. Influence of ship dimensions and centrrelines sapcing on the suction forces $X_S$, $Y_S$ and the yawing moment $N_S$*

| Parameter | Influence on<br>Suction forces $Y_S$ | <br>Yawing moment $Y_S$ |
|---|---|---|
| Mean lemth $L_M$ | $\sim L_M$ | $\sim L_M^2$ |
| Maen draught $T_M$ | $\sim T_M$ | |
| Mean speed $u_M$ | $\sim u_M^2$ | |
| Spacing between centrelines: D rel. to $D_0$ | | |
| Small up to $\sim 0.6L_M$ | $\sim D^{-1}$ | |
| Medium $\sim 0.6\ L_M$……$\sim 1.0\ L_M$ | $\sim D^{-2}$ | |
| Large over $\sim 1.0\ L_M$ | $\sim D^{-3}$…..$D^{-4}$ | |

$$\tilde{X}_S \max,\ \tilde{Y}_S \max = C_{XS} \max,\ C_{YS} \max \cdot \left( \frac{\rho}{2} u_M^2 L_M T_M \right)$$

and

$$\tilde{N}_S \max = C_{NS} \max \cdot \left( \frac{\rho}{2} u_M^2 L_M^2 T_M \right)$$

They to be corrected for differences $D \neq D_0$ according to Table 5 in order to obtain the corrected values $X_S$max, $Y_S$max and $N_S$max.

The curves of these interactions as functions of the ship position $\xi$ may be constructed using Table 6 with the following:

$\kappa_0(-) = \xi / L_M$ for the relative stagger

$\kappa_1(-) = X_S / X_S \max$ for the longitudinak force $X_S$

$\kappa_2(-) = Y_S / Y_S \max$ for the transverse force $Y_S$

$\kappa_3(-) = N_S / N_S \max$ for the yawing moment $N_S$

From the above equations and Table 6 one obtains the suction forces:

$$X_S = \kappa_1 \cdot X_S \max$$

$$Y_S = \kappa_2 \cdot Y_S \max$$

and the yawing moment:

$$N_S = \kappa_3 \cdot N_S \max$$

For various staggered positions:

$$\xi = \kappa_1 \cdot L_M$$

The time T for the complete overtaking manoeuvre $-L_M \leq \xi \leq +L_M$ amounts to:

$$T = \frac{L_1 + L_2}{u_1 - u_2} = \frac{2L_M}{\Delta u}$$

The curves of ship/ship interactions thus obtained have the characteristics as shown in Fig. 25.

*Table 6 .Coefficients to be used for estimation suction forces and yawing moment*

| $\kappa_0$ | $\kappa_1$ | $\kappa_2$ | $\kappa_3$ |
|---|---|---|---|
| -1.00 | -0.289 | +0.298 | +0.264 |
| -0.75 | -0.690 | +0.345 | +0.706 |
| -0.50 | -1.000 | -0.060 | +1.000 |
| -0.25 | -0.850 | -0.595 | +0.837 |
| -0 | -0.2850 | -0.935 | +0.221 |
| +0.25 | +0.590 | -0.982 | -0.682 |
| +0.50 | +0.980 | -0.637 | -0.927 |
| +0.75 | +0.810 | -0.250 | -0.706 |
| +1.00 | +0.330 | -0.089 | -0.424 |

At larger $L_2/L_1$ relationship than about 2 this method is not reliable.

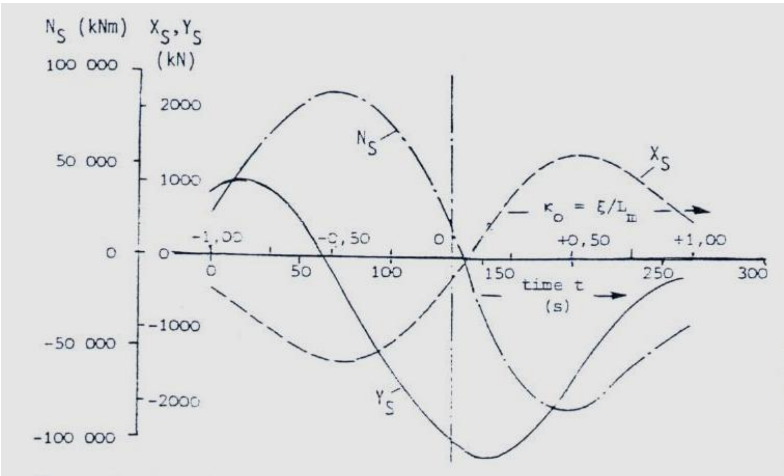



**Fig.25. Ship/ship interaction curves.**

### 1.2.2. Interaction effect when a ship is passing moored ship

Interaction effects are also present where one ship is passing the other one that is dead in the water, moored or at anchor. In this situation forces are acting on moored ship that may endanger the passing ship that may loose directional control or may cause that the mooring ropes broke. This case is covered by report provided by V.Ankudinov (TRANSAS) that with some editorial changes is reproduced below.

Survey of ship – to ship interaction effects –modelling and simulations.

Figure 26 below describe typical Ship to Ship Port Navigational Situation when Passing Ship moving along of a moored vessel. Similar analysis can be applied to cases when both ships are moving with their own speeds (like head-on encounter) and with the arbitrary relative heading angle to each other. For a typical Passing cruise ship of Displacement around 30,000 – 50,000 t these forces can reach value of 100-200 t and passed vessel can easily loose directional control or brake the mooring lines.

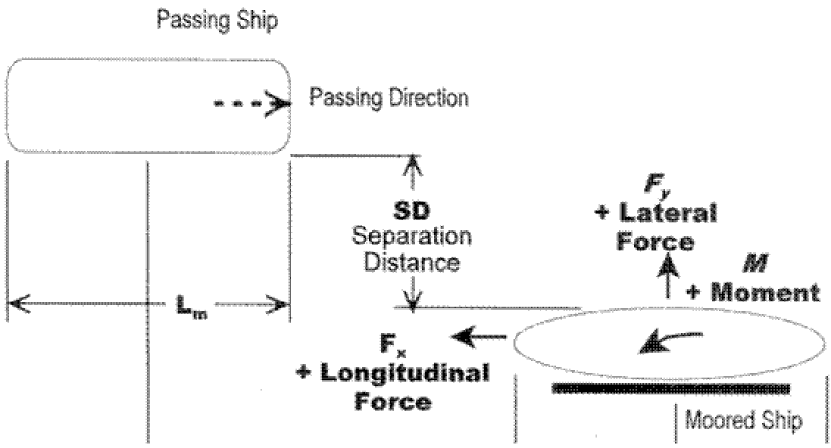



**Fig. 26 Nomenclature and Conventions used in this part**

Hydrodynamic Aspects of the Ship –Ship Interaction (SSI) Modeling and Simulation

A realistic description of the flow of water past. a ship will during a manoeuvrer in the proximity of other vessels or other water restrictions (channel, walls etc) pose one of the most complex problems encountered in the field of ship hydrodynamics.

Included to a significant extent are most of the phenomena which are associated with the mathematical model of the single ship, such as inertia and damping (viscous and wave) effects, propeller wake, hull/rudder/propeller interaction and waterway restriction effects The presence of another vessel qualitatively and quantitatively modify all these effects in time and space and, in effect, make hydrodynamic coefficients of the own vessel which are typically constant for specific ship and rudder/propeller complex, being now dependent on the geometry and kinematics of other vessels.

Perhaps a usage of the conventional simulation model with constant coefficients might be questionable Under the circumstances it is not surprising to see so many conflicting results on the subjects and in practical matters, a significant thrust towards to a generalization of some incomplete experimental observations. A fairly realistic description of the flow around conventional ship hull in the vicinity of another fine form hull can be formulated assuming the field is nonviscous, disturbance of the free surface are sufficiently small, the boundary layer is thin, and there is no large-scale separation or ventilation on both vessels. The modern numerical and hydrodynamic methods are available to solve this problem on various levels of accuracy and practical demand.

What is the most important in regard to proximity effect that this (non viscous) proximity forces are the major parts of the ship interaction forces with relatively small contribution from the propeller rudder and viscous wake. Also the "ship proximity" lateral disturbance and lateral velocities are relatively small (maximum values of the cross-flow angles due to the "ship proximity" are in the range from five to eight degrees with average values around 3 to 4 degrees). Because streamlines and boundary layers of the simulation model of he ship with constant coefficients seems to be approximately valid, and ship proximity forces can be added as an external disturbances at the specific time and location. This methodology will be utilized in he hydrodynamic analyses of two vessels in the proximity to each other.

Typical non-dimensional hydrodynamic SSI hydrodynamic forces in surge, sway and yaw are shown on Fig.27 below. Figure 28 provides the Narrative of the SSI Forces during the passage.

The non- dimensional SSI force coefficients are defined as follows:

X (int – non dimensional) = X (int)/ {0.5*rho*(Up)**2 *[ Cb * Bp *Tp]}

Y (int – non dimensional) = Y (int)/ {0.5*rho*(Up)**2 *[ Cb * Bp *Tp]}

N (int – non dimensional) = N (int)/ {0.5*rho*(Up)**2 *[ Cb * Bp *Tp * Lp]}

Here Cb , Bp ,Tp , Lp are block coefficient, beam, and length of the passing vessel.

Force and Moment Graphs for TRANSAS VIRTUAL SSI SHIP in relative water depth H/T = 1.15 and for relative passing distance between the hulls equal to 0.645*Bp. Other operational and waterway conditions can be easily derived using CORRECTION COEFFICIENTS**.**

For simple SSI simulations of the moored or slow moving vessel (like Cruise Ship arriving and then being moored a port Terminal) on PC-based Simulator the following simulation techniques can be used:

1. Simplified Ship – Ship Interaction Modeling can be used for the standard ship types (tankers, BC, containerships, etc.), for any shallow water conditions and channel, and perhaps for ship speeds less than 10 knots where wave making is relatively small. It also approximately accounts for the relative ship heading between hulls and relative passing distance (clearance) between the two ship hulls.

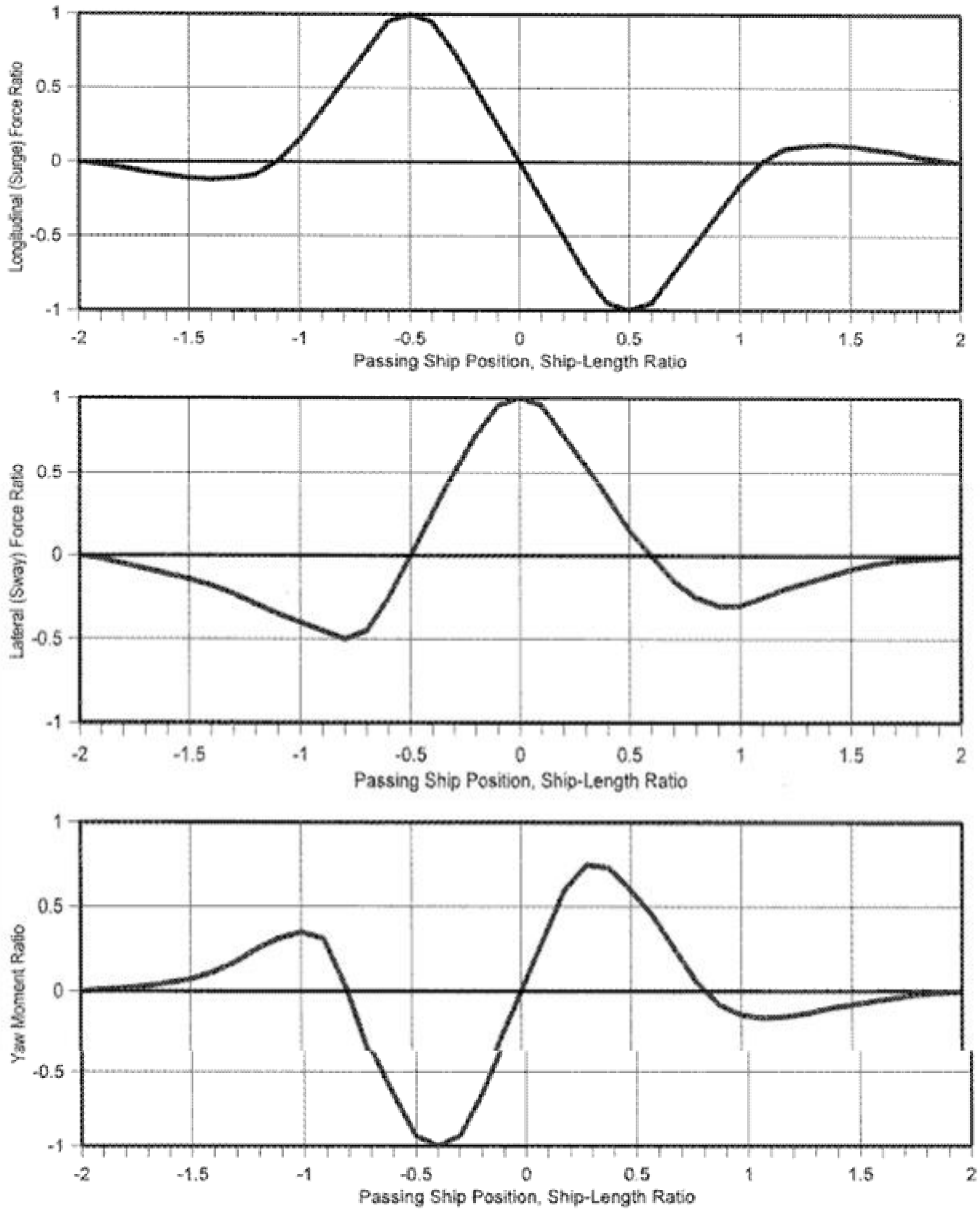



**Fig. 27. Non-dimensional passing ship poisition**

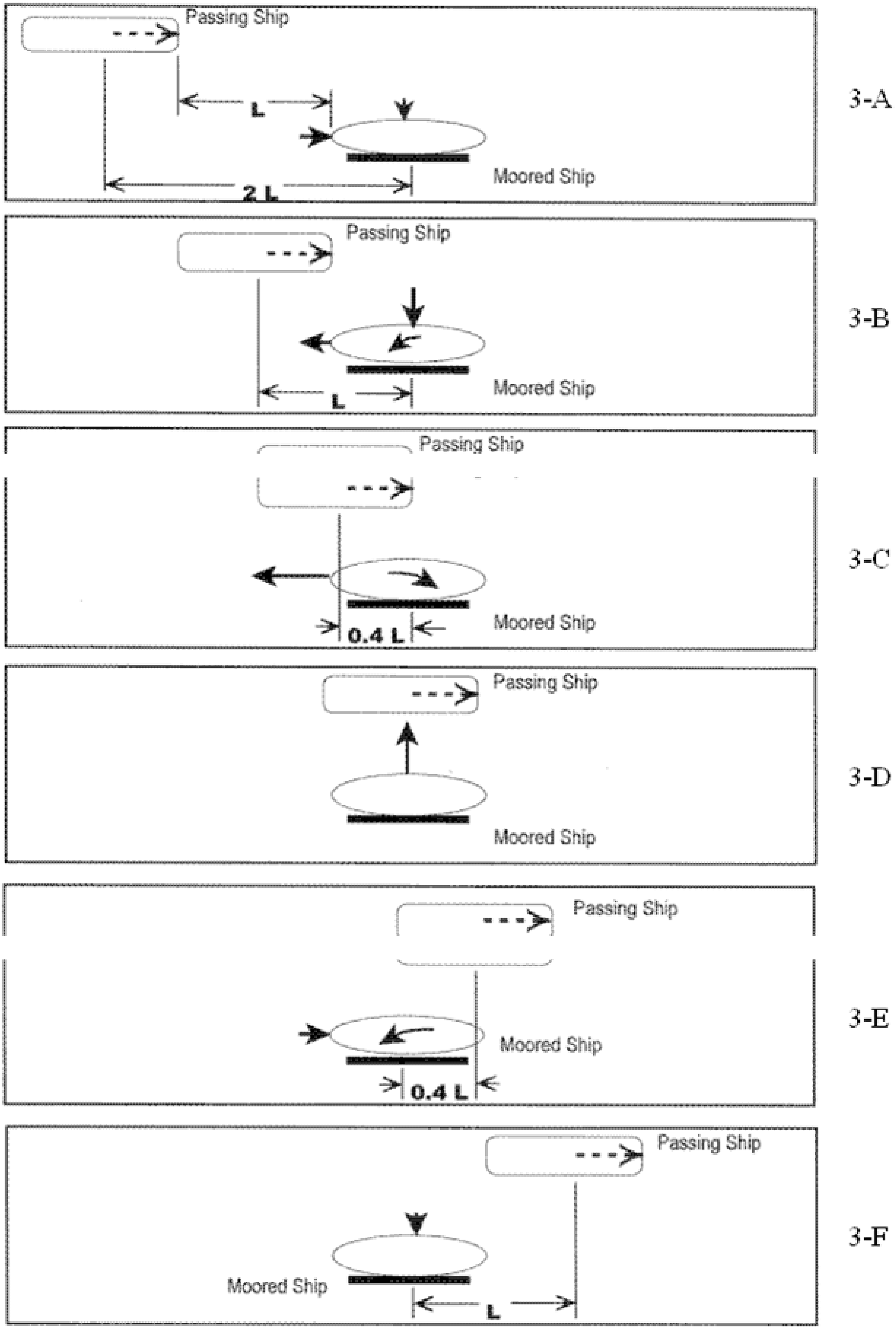



**Fig. 28: Narrative of Passing Ship Forces and Moments during Passage**

2.The SSI hydrodynamic force components in surge, sway and yaw will be computed from the following expressions:

a.X SSI force = X (non-dimen) x { [2 – Uown/Up]} x {0.5*rho*(Up)**2 *[ Cb * Bp *Tp]} x COS (rel. head)x

{ 1/ [1 – 0.95/(Hp/Tp)**2]} x {1 – 0.90/(Bchannel/Bp)**2]}

x {7 –(7/4)*[Dpass/Bp]}.

b.Y SSI force = Y (non-dimen) x { [2 – Uown/Up]} x {0.5*rho*(Up)**2 *[ Cb * Bp *Tp]} x COS (rel. head)x

{ 1/ [1 – 0.95/(Hp/Tp)**2]} x {1 – 0.90/(Bchannel/Bp)**2]}

x {7 –(7/4)*[Dpass/Bp]}.

c.N SSI force = N (non-dimen) x { [2 – Uown/Up]} x {0.5*rho*(Up)**2 *[ Cb * Bp *Tp* Lp]} x COS (rel. head)x { 1/ [1 – 0.95/(Hp/Tp)**2]} x {1 – 0.90/(Bchannel/Bp)**2]} x {7 –(7/4)*[Dpass/Bp]}.

d. X (non-dimen), Y (non-Dimen) and N (non-dimen) are non-dimensional X, Y, N of the SSI hydro forces taken from curves shown on Figure 27 Non dimensional Passing Ship Position vs. Force and Moment Graphs for TRANSAS VIRTUAL SSI SHIP in relative water depth H/T = 1.15 and for relative passing distance between the hulls equal to 0.645*Bp. Other operational and waterway conditions can be easily derived using CORRECTION COEFFICIENTS as explained below.

e.The next term { [2 – Uown/Up]} is the speed corrections where Uown is the ship speed of the own ship and Up is the ship of the passing vessel. If two ships are passing each other (U own has the same sign as Up) the SSI forces will be reduced. It is easy to see that for the mooring ship (Uown = 0) the second term will be also zero. However in head on encounter (Uown and Up will be of the opposite signes) the second term will be of positive value and we will get the largest SSI forces – it is well established and confirmed physical phenomenon.

f.The term {0.5*rho*(Up)**2 *[ Cb * Bp *Tp]} controls speed and main hull parameters of the passing vessel. It means that the main source of SSI forces is due to the moving pressure distribution of the passing vessel.

g. The term COS (rel. head) is cos function of the relative heading angle between two hulls. For the heading angles equal 90 and 270 degrees (and close to these angles) there will be no SSI forces – also well- established fact.

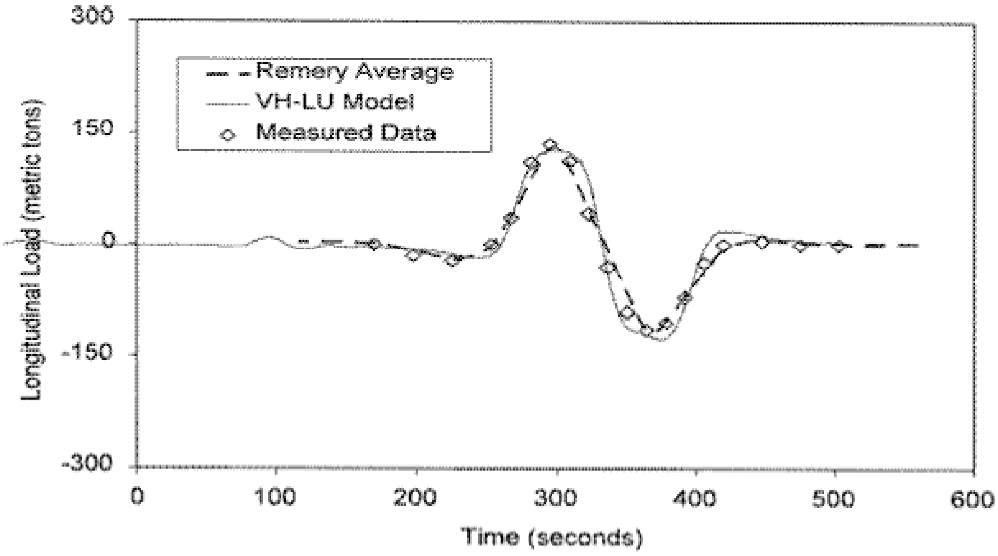
h.The next term {1/ [1 – 0.95/ (Hp/Tp)**2]} describes the shallow water effect on SSI forces. Details and values are given below.

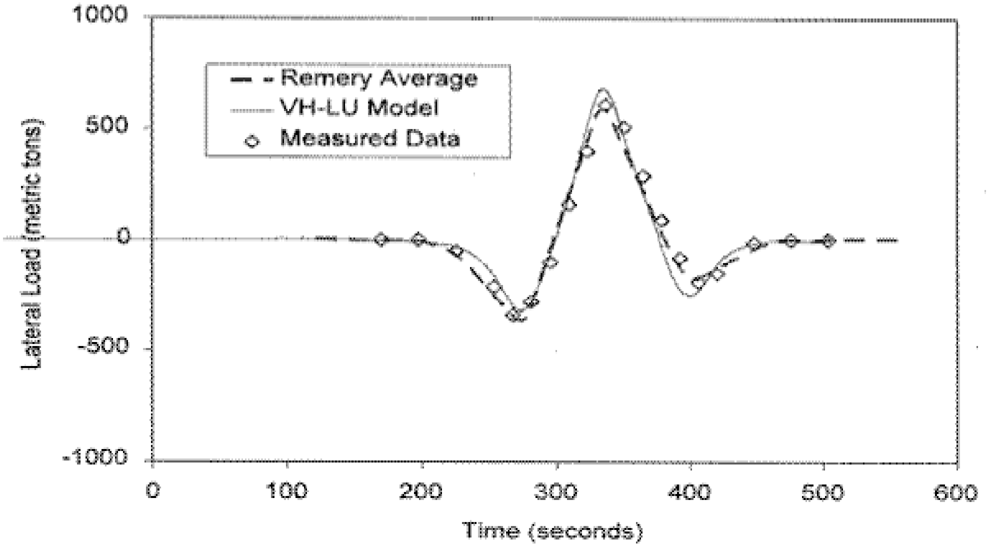
i.The term {1 – 0.90/(Bchannel/Bp)**2]} describes the channel effect. See below also for more details.

j.The last term describes the effect of the lateral distance (clearance) Dpass between the hulls. At Dpass =0 (both hulls are skin to skin) the SSI reach the maximum values.

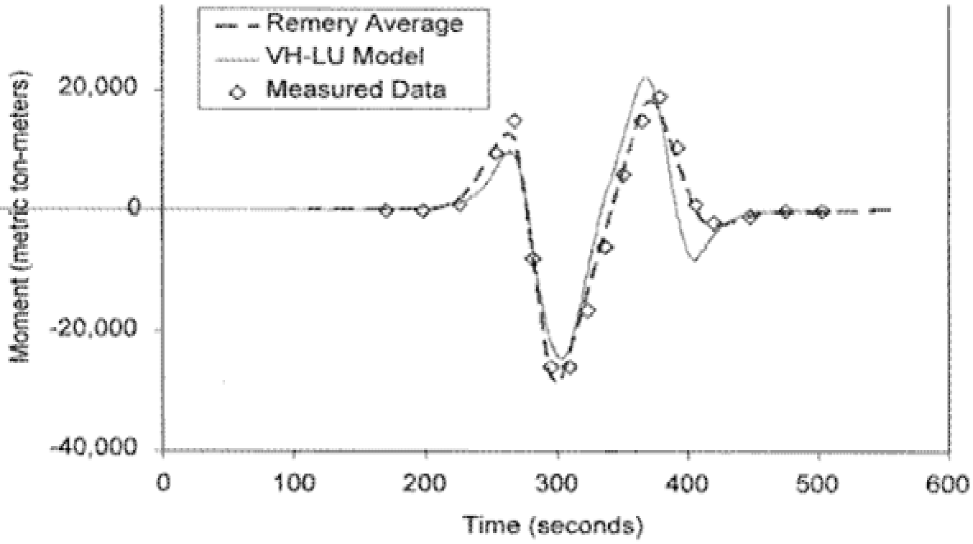
1.2For more general SSI cases in passing or head–on- encounter a relatively sophisticated numerical methods based on modeling both hulls by so-called source distributions are used.

1.3Example of these computations and comparison with the test data are shown in Fig. 29.


300
150
0
-150
-300
Longitudinal Load (metric tons)
Remery Average
VH-LU Model
Measured Data
0
100
200
300
400
500
600
Time (seconds)


1000
500
0
-500
-1000
Lateral Load (metric tons)
Remery Average
VH-LU Model
Measured Data
0
100
200
300
400
500
600
Time (seconds)


40,000



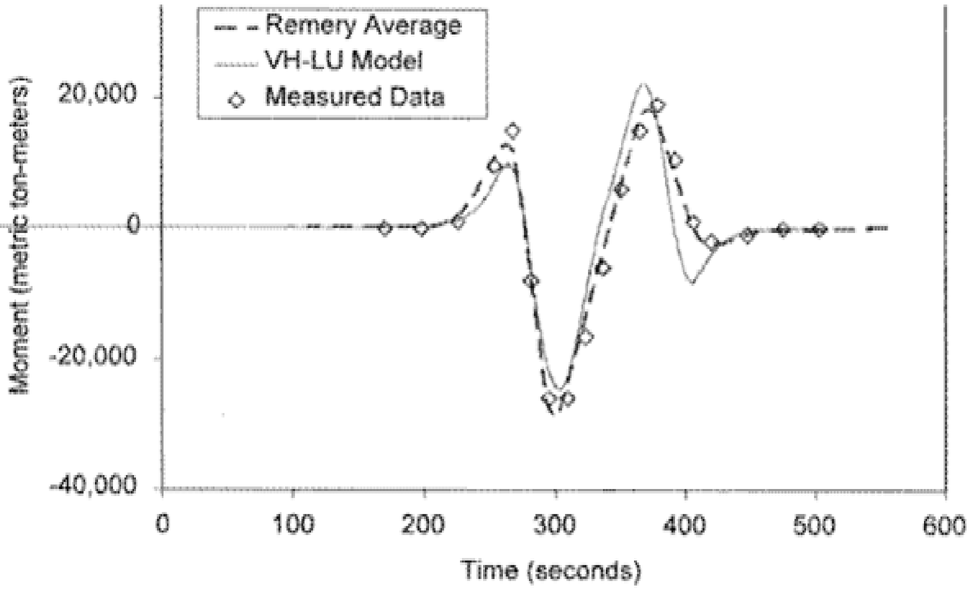




**Fig. 29. Comparison of Wagenigen Model Test Data (Case 2: 100,000 t moored cruise ship being passed by 130,000 t cruise ship) and VH-LU computations.**

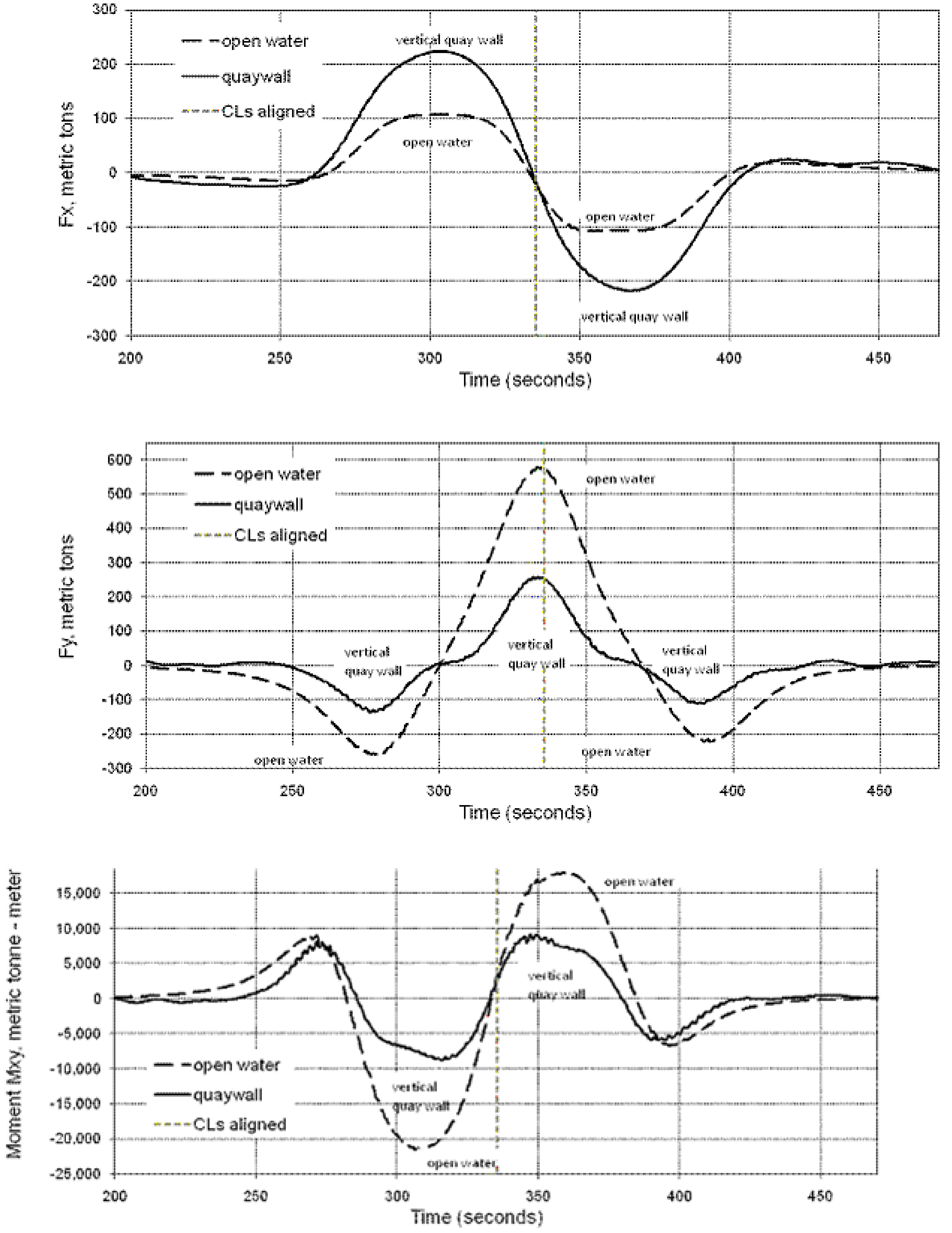
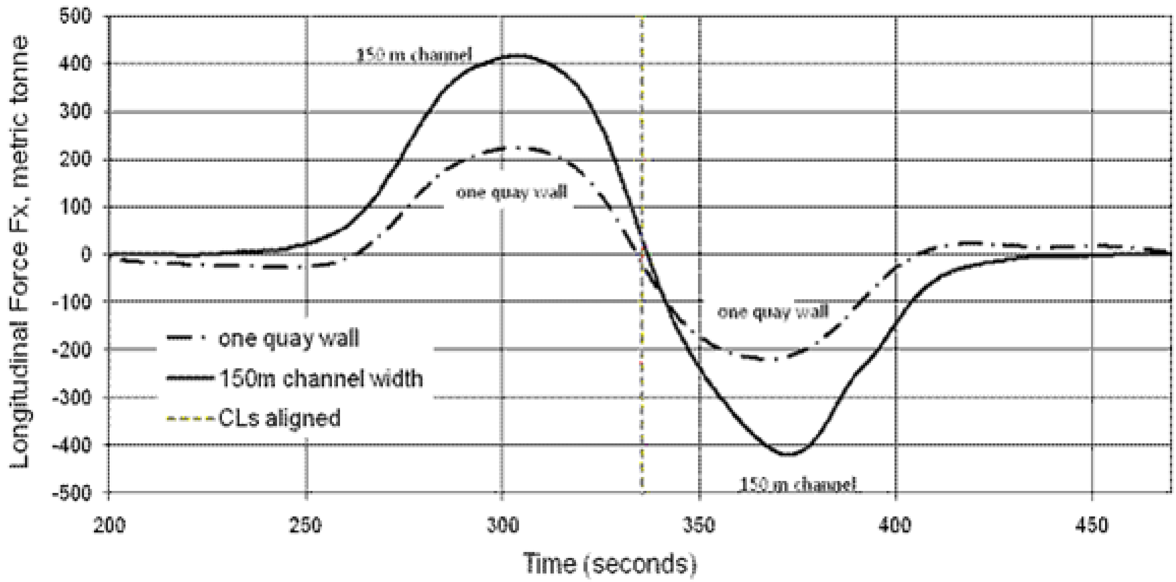
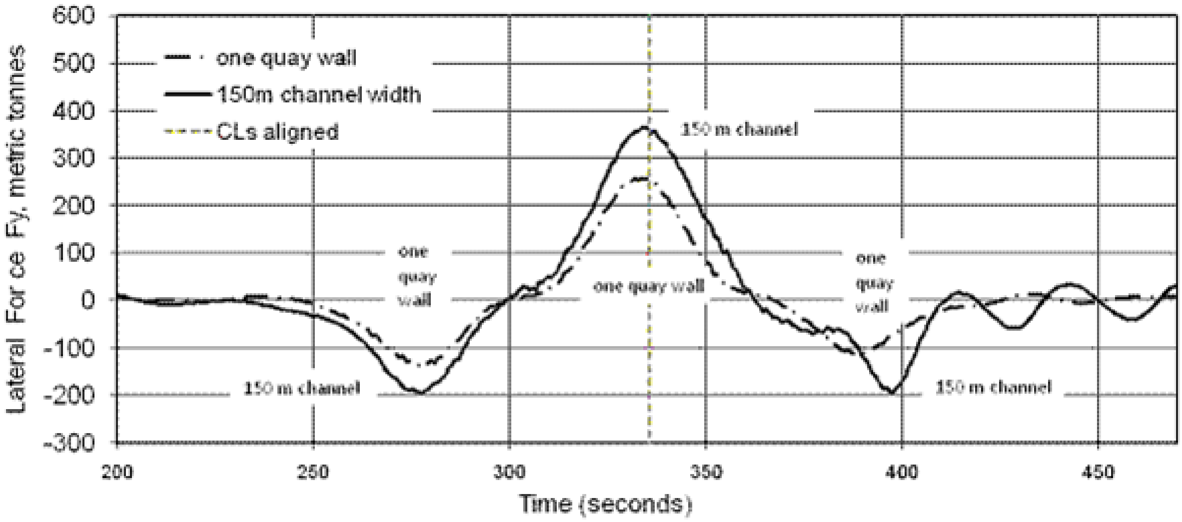
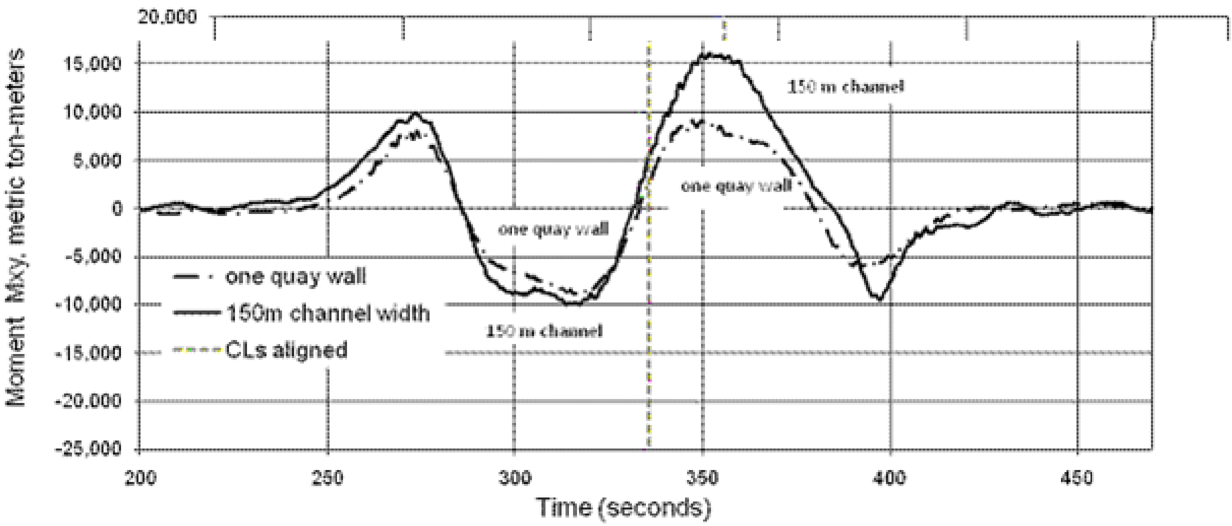



**Fig. 30. Effect of Adjacent Vertical Quay Wall on Passing Ship Effects**


Longitudinal Force Fx, metric tonne
Time (seconds)
150 m channel
one quay wall
one quay wall
150 m channel
one quay wall
150m channel width
CLs aligned


Lateral For ce Fy, metric tonnes
Time (seconds)
one quay wall
150m channel width
CLs aligned
150 m channel
one quay wall
one quay wall
one quay wall
150 m channel
150 m channel


Moment Mxy, metric ton-meters
Time (seconds)
150 m channel
one quay wall
one quay wall
150 m channel
one quay wall
150m channel width
CLs aligned

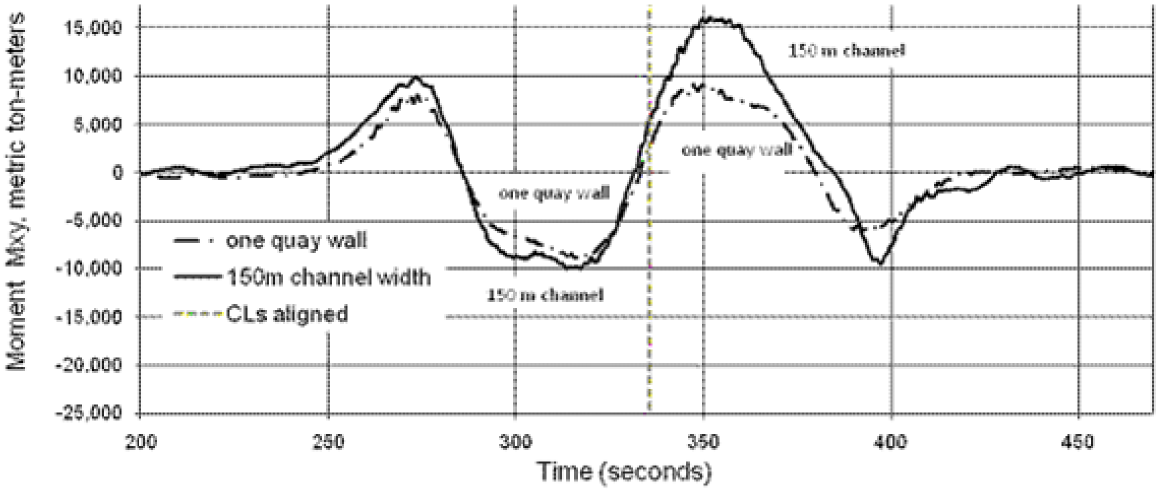



**Fig. 31. Comparison of Quay Wall With and Without Restricted Channel Width**

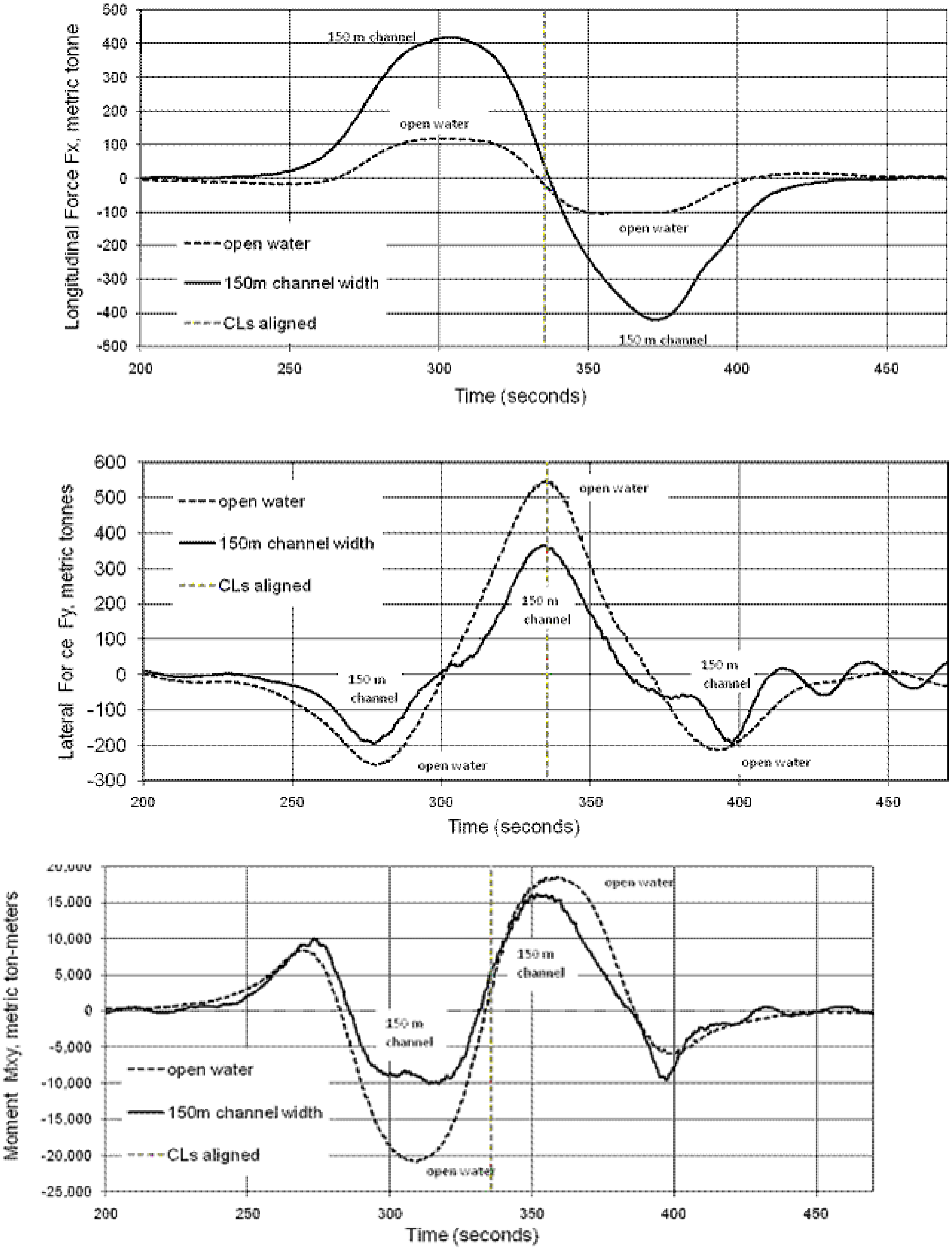



**Fig. 32. Comparison of Open Water with Effect of Channel Walls**

The next four Figures (Figs. 33-36) show the Simulation results on the US Army Maneuvering Simulator along with the UK and Dutch model test results

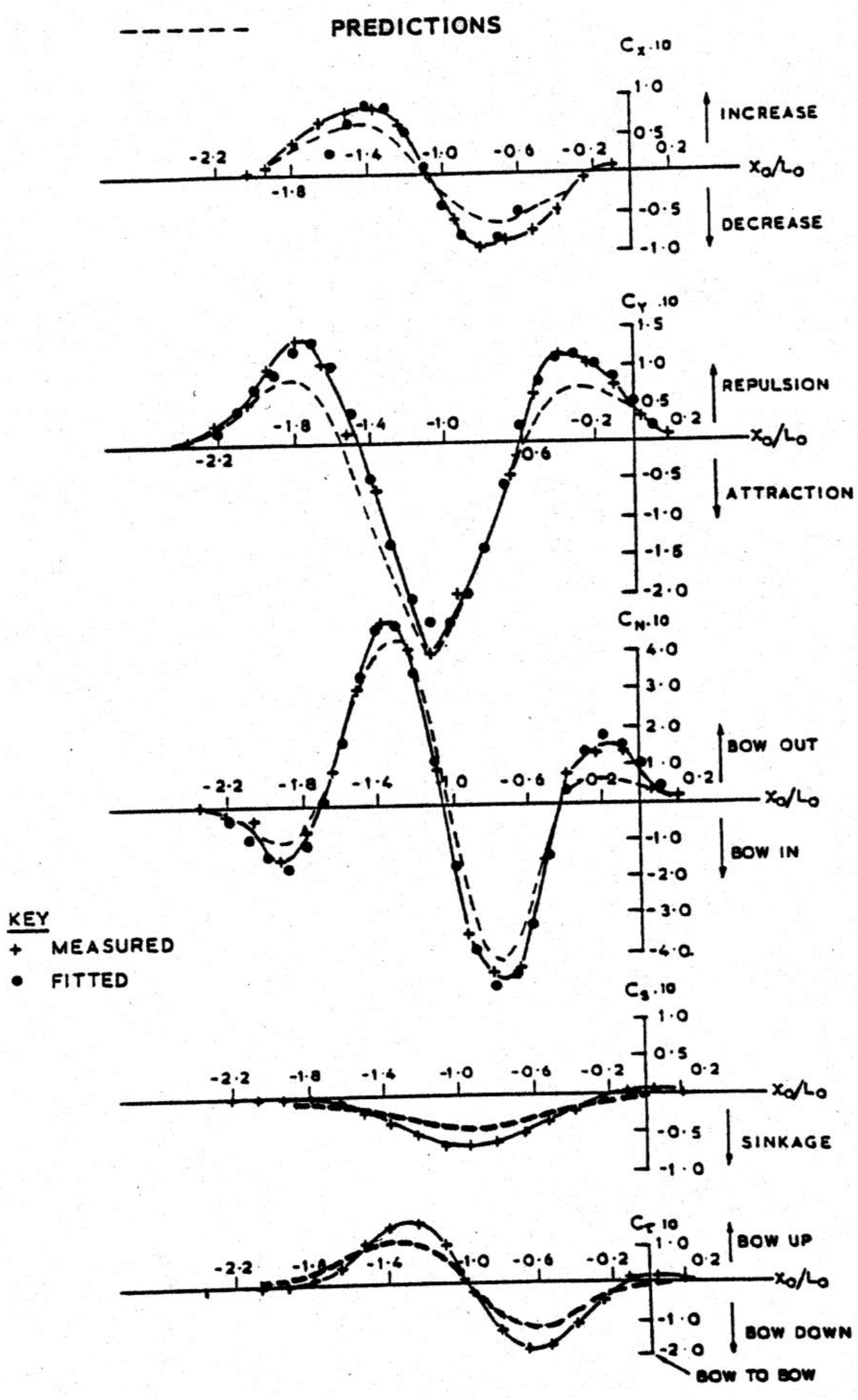
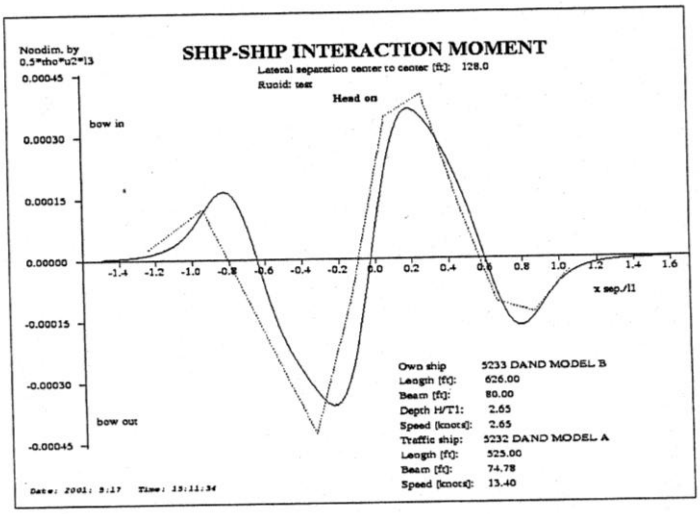



**Fig.33. Head-on encounter. comparison of the predictions and measurements. depyh/draft ratio is 2.6, relative separation of hulls is 1.6 ship beam.**




Mooring Forces Induced by Passing Ships, G.F.M. Remery, Netherlands Ship Model Basin, Offshore Technology Conference, 1974 (OTC 2066)

A tanker is passing another moored tanker (head to head)

Own ship
Ship length = 843.0 feet
Ship beam = 120.67 feet
Ship draft = 51.53 feet
Displacement = 119,819.4 LT
Speed = 0 knots

Traffic ship
Ship length = 991.0 feet
Ship beam = 152.57 feet
Ship draft = 52.53 feet
Displacement = 187,327.4 LT
Speed = 7 knots

**Fig. 34. A tanker passing another moored tanker. (head to head)**

The theoretical method has been adjusted to make as good a fit the model test available from literature. Some of the most careful model tests were carried out by Ian Dand: Some Measurement of Interaction between Ship Models Passing on Parallel courses, NMI R 108, August 1981.

**Meeting / Head on ships**

The own ship is DAND B (5233)
Length = 626 feet
Beam = 80 feet
Draft = 34 feet
Fnh = 0.25
Speed = 8 knots

Traffic ship is DAND A (5232)
Length = 525 feet
Beam = 74.78 feet
Draft = 26.5 feet
Fnh = -0.421
Speed = 13.40 knots

Water depth draft ratio = 2.65 – which is deep water
Water depth = 90.1 feet
Separation = 128 feet, Yo / Beam = 1.6

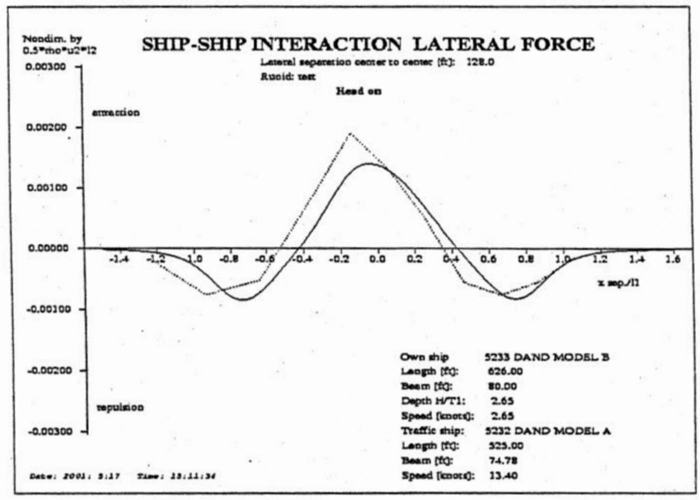



**Fig. 35. Ship-ship interaction lateral force**

Own ship DAND B (5233)
Same ships in shallow water
Separation = 128 feet
Own ship water depth draft ratio = 1.2
Fnh = 0.402
Own ship speed = 7.87 knots

Traffic ship DAND A (5232)

Fnh = -0.518
Traffic ship speed = 10.14 knots

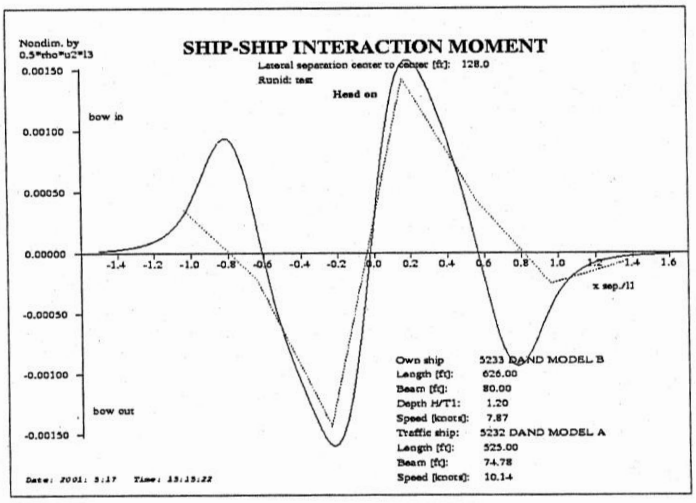



**Fig.36. Ship-ship interaction moment**

## 1.3. Effect of soft bottom and mud

Little is known on the effect of soft bottom and mud effect on ship behavior and manoeuvring characteristics in shallow water or in a canal.

However, as the effect of bottom on the manoeuvring characteristics and first of all on squat depend on pressure field under and around ship hull, that is modified when the ship is moving in shallow water or in a canal, the effect of muddy bottom must be strong, because the pressures are dissipated in the layer of mud that is mixture of water and earth of different densities.

Extensive research work on the effect of muddy bottom on manoeuvrability was performed by Delafortrie and was published as the doctor thesis by Ghent University under promotorship of professor Vantorre [21].

The research included experimental part where models were tested in the towing tank where at the bottom a layer of artificial mud was created, then the mathematical model of ship behavior in muddy waters was developed an finally simulation runs were carried out.

To assess the effect of muddy bottom on the possibility of navigation the concept of nautical bottom was introduced. According to International Navigation Association the definition of nautical bottom is as follows:

*"The nautical bottom is the level where physical characteristics of the bottom reach a critical limit beyond which contact with ship's keel cases either damage or unacceptable effects on controllability and manoeuvrability"*

This definition implies that in muddy waters ship bottom may touch the upper layer of mud and still navigate safely. The criterion of the nautical bottom depend on the availability of tugs and the author of the dissertation together with Zeebrugge pilots proposed a critical limit of 1.2 t/m$^3$ if at least 2x45 ton bollard pull tug assistance is available. Penetration of mud layers of lower density is restricted to:

- - 0% UKC if tugs of 30 tons bollard pull and less are available
- - 7% UKC for assistance of 2 tugs of 45 tons bollard pull
- -12% UKC for assistance of 2 tugs of 60 tons bollard pull

Formation of the mud layer is a complicated process. Fig. 37 (from Delafortrie [21]) shows the mud cycle where process of consolidation at the bottom layer, liquefaction and fluidization – medium layer and deposition –upper layer are shown. In the uppermost layer concentrated suspension of particles occurs. These processes are time dependent and are influenced by water flow caused by current and waves.

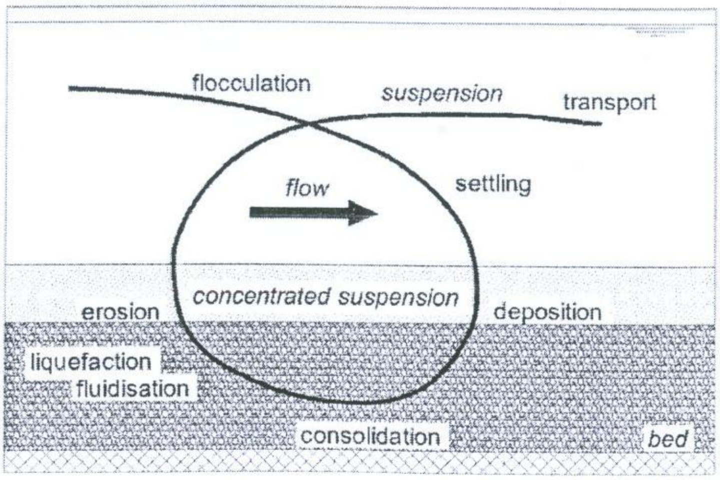



**Fig. 37. The mud cycle.**

Typical density profile of muddy bottom for Zeebrugge harbour is shown in Fig.38.

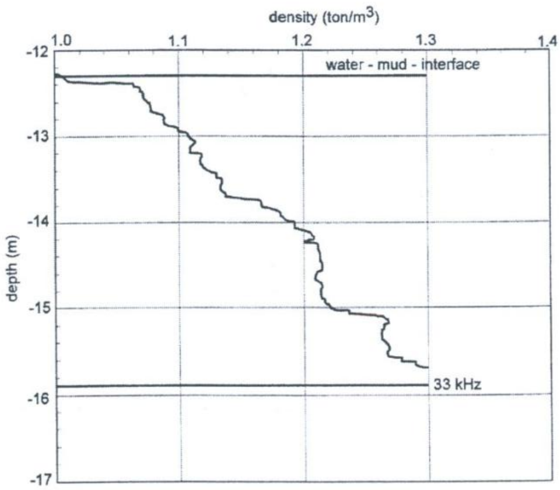
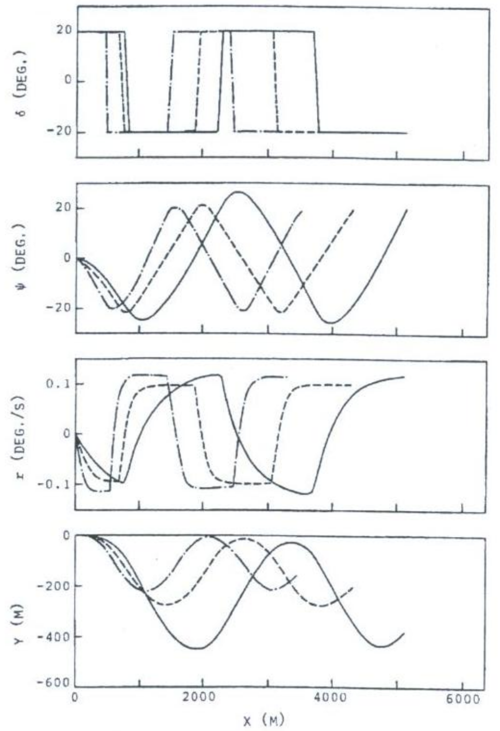



**Fig. 38. Density profile in function of the water depth**

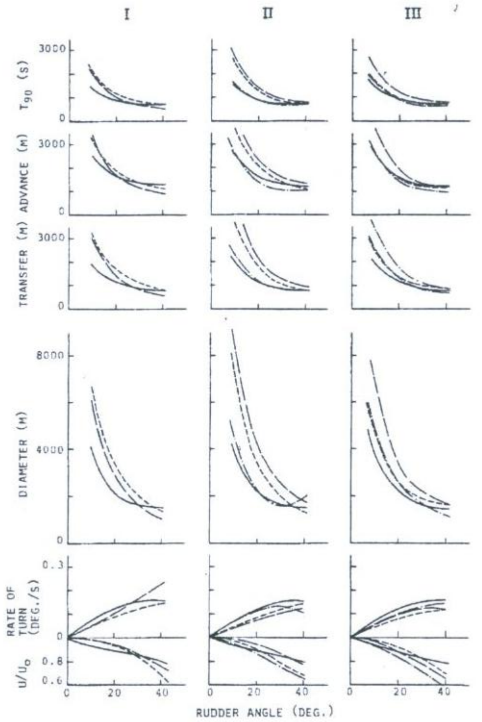
Apart from the dissertation by Delafortrie results of manoeuvrability tests executed in MARIN in muddy waters were published by Sellmeijer et al [23]. Results of zig-zag tests and of turning circle tests of a tanker from this publication are included here (Fig 28 and 29.)

The tests performed at Flanders Hydraulic Research and also at Sogreah and also some full scale tests showed that undulations of the water-mud interface occur when passing ship and those undulations have an effect on the manoeuvring behaviour of the ship. But in general the scope of the tests was rather limited and general conclusions could not be drawn. Artificial mud was used in the experiments.




| Type of line | Mud | | Keel clearance | |
|---|---|---|---|---|
| | Density (kg/m$^3$) | Thickness | To mud | To bottom |
| ———— | 0 | 0 | – | 0·20 T |
| ------- | 1140 | 0·13 T | 0·10 T | 0·23 T |
| —·—· | 1240 | 0·13 T | 0·10 T | 0·23 T |

**Fig.39. Zig-zag test.**




| Group | Type of line | Mud | | Keel clearance | |
|---|---|---|---|---|---|
| | | Density (kg/m$^3$) | Thick-ness | To mud | To bottom |
| I | ———— | 0 | 0 | – | 0·20 T |
| | --------- | 1140 | 0·07 T | 0·15 T | 0·22 T |
| | — — — | 1140 | 0·13 T | 0·10 T | 0·23 T |
| II | ———— | 0 | 0 | – | 0·20 T |
| | --------- | 1140 | 0·07 T | 0·10 T | 0·17 T |
| | — — — | 1140 | 0·13 T | 0·03 T | 0·16 T |
| | —·—·— | 1140 | 0·20 T | −0·03 T | 0·17 T |
| III | ———— | 0 | 0 | – | 0·20 T |
| | --------- | 1140 | 0·07 T | 0·03 T | 0·10 T |
| | — — — | 1140 | 0·13 T | −0·03 T | 0·10 T |
| | —·—·— | 1140 | 0·20 T | −0·10 T | 0·10 T |

**Fig.40. Turning circles: effect of mud thickness**

The wider program of model tests was arranged by Delafortrie [21].

Within the scope of the programme first of all the undulations of the mud-water interface were tested. The typical undulations that occur at the interface are shown in Fig. 41. To assess the amount of undulations may be important in order to estimate UKC.

The main conclusions from these tests are as follows:

- The rising increases with increasing speed
- The increase is limited. Once the limit has been reached the rising can decrease again as it is with low density mud layers.
- When the vessel navigates above mud layer the rising will increase faster when the density and viscosity of the mud layer are small. With thinner mud layers the rising becomes significant once the viscosity drops below a certain value,

- A significant undulation is always observed when the ship navigates in contact with the mud layer. The rising is mostly located amidships for higher density mud layers. For lower density and viscosity the rising is located abaft.
- When navigating ahead a positive propeller rate only influences the rising when it is located near the stern.
- Navigating astern with reversed propeller has little to zero influence on the rising.

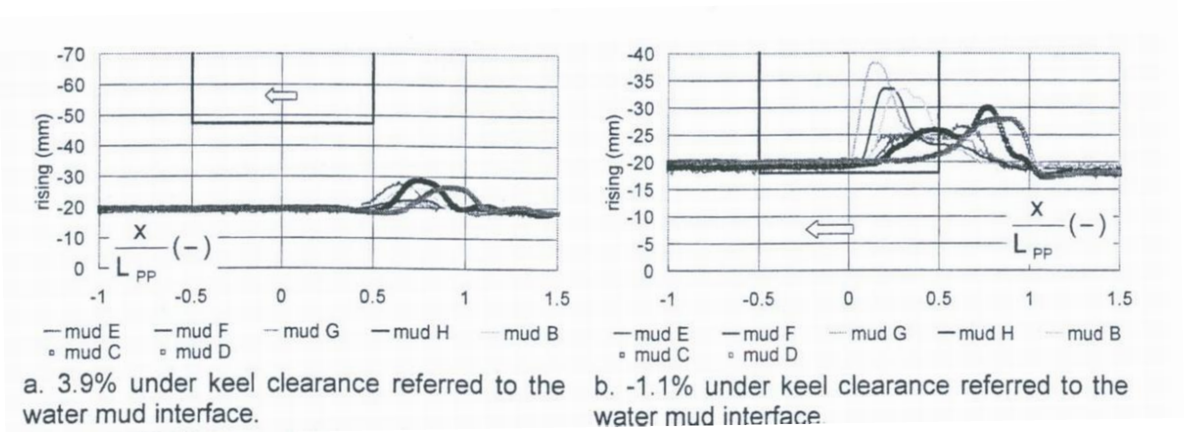



**Fig.41. Undulations of the surface at mufo 1, no propeller or rudder action. Ship speed = 0.6 m/s. Thickness of the mud layer 20mm.**

Important part of model tests was measuring hydrodynamic derivatives of hull, propeller induced and rudder induced forces on PMM in towing tank where mud layer of different characteristics was at the bottom..The model used was a typical single screw. container vessel.

Apparently all results of measured hydrodynamic hull derivatives are applicable also to pod driven vessels apart propeller induced and rudder induced forces. The propeller behind the hull in pod driven ship will be operating in another flow environment closely to water mud interface where mud particles are suspended. The difference will be caused by the form of the stern that in general may be different, by interaction between left and right pod and because pods are also acting as rudders creating propulsion and control force in different directions. Therefore results of the tests performed with single screw vessel may not be applicable to pod driven ships in this respect. But relevant data on propeller-rudder induced forces for pod driven ships in shallow water and mud layer are not available.

A series of fast-time simulations has been carried out by Delafortrie [21] in order to assess manoeuvring characteristics of tested ship (6000TEU container ship) in muddy areas. The manoeuvres simulated were:

- Acceleration test
- Turning circles
- Zig-zag tests
- Crash stop
- Tug assistance
- Course change
- Course keeping in current
- Back and fill

Some conclusions from these simulation are given below.

Turning circles

Turning ability of the vessel which is already small at 30% UKC above the solid bottom compared to deep water will further decrease when navigating above mud layer. The tactical diameter reaches a maximum at extremely small positive UKC. The less effective propulsion is probably due to rising mud water interface near propeller that is working in different environment. This conclusion must be applicable also to pod driven ship. The same apply to transfer and advance characteristics.

Zig-zag tests

The overshoot angles are larger above the solid bottom and decrease significantly in muddy areas. A local minimum of overshoot time and angle can be observed at extremely small positive UKC.

Crash stop

At small positive UKC above mud layer stopping ability is better because of larger damping. When the ship navigates in contact with mud layer of higher density the time to stop is relatively large but still acceptable.

The hydrodynamically equivalent depth

As the effect of shallow water with solid bottom on manoeuvring characteristics of ships is fairly well known, the useful concept for muddy water areas is hydrodynamically equivalent depth i.e. the corresponding depth above the solid bottom that leads to the same forces as above the solid bottom without mud layer

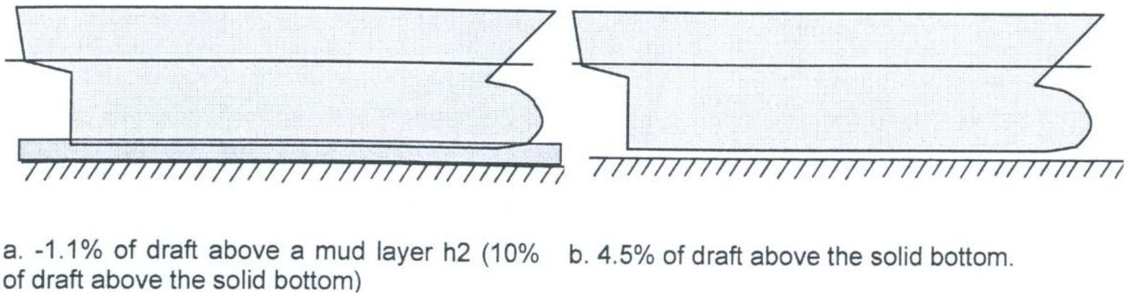



**Fig. 42. Example of hydrodynamically equivalent depth. Both conditions are hydrodynamically equal**

The hydrodynamically equivalent depth is (Fig. 42.)

$$h^* = h_1 + \Phi h_2 \leq h$$

Where:

$h_1$= the height of upper lying water layer

$h_2$ =the thickness of the mud layer

$\Phi$ = parameter, $\Phi$=0 (hard layer of thickness $h_2$)

$\Phi$ =1 (watery layer of thickness $h_2$)

## 1.4.Steering when towing or under tow

In the towing operation, when one vessel is towing the other one that has no rudder action and no engine working, heavy yawing motion will usually appear endangering the towed vessel and in general the whole traffic around as well. The yawing motion is causing reduction of speed and make towing operation difficult or even impossible.

The tendency for self excited yawing motions of is increasing with (Brix [11]):

- Increasing L/B ratio
- Increasing draught T
- Very short or very long hawser
- Shifting the hawser attachment aft
- Trim by the head

Increasing L/B ratio of the towed vessel has a negative effect on course stability and the towed vessel may develop heavy yawing motion of large amplitudes. This is actually in contradiction to the common observation that self propelled free running ships with increasing L/B ratio have better course keeping stability. This conclusion is supported by tests of the towed pontoon with L/B =2 that was found stable as reported by Brix [11] after Dawson. Also increased draught of the towed ship has negative effect on the towed vessel stability. The same negative effect is observed when the towed ship is trimmed to the bow.

Length of the hawser is important factor affecting yawing characteristics of towed vessel. Generally course instability of towed vessel is expected when moderate or long lengths of hawser are used and towed vessel became more stable with short length of hawser. Brix [11] pointed out , however, that the experience of American and Canadian barge shipping shows that some of the barges towed at short lengths are prone to instability but became stable on longer hawsers. This effect was also observed in tests of towing performed in Hamburg Towing Tank.

There is the criterion of course stability related to the length of towing hawser developed by Strandhagen [12] based on consideration of equations of motion (although linear). This criterion could be use if hydrodynamic derivatives for the towed vessel are known.

Shifting the point of attachment of the towing hawser has an important effect on course stability. This point should be shifted towards the bow as far as possible; this will increase course stability.

Skegs or fins fitted at stern have important stabilizing effect. This is similar effect as with adding resistance increment at stern. Brix [11] pointed out that hydrodynamic cross forces of skegs or fins at stern are of main interest for course stabilization, although the resistance component contributes to the course stabilizing steering moment.

Distribution of force on stern skegs is shown in Fig.43 (Brix [11]). In this figure the lift forces L are of equal amount and opposite direction , whereas the drag forces D are parallel. The hydrodynamic cross forces C and the resistance components W in longitudinal direction are obtained via the resulting forces R.

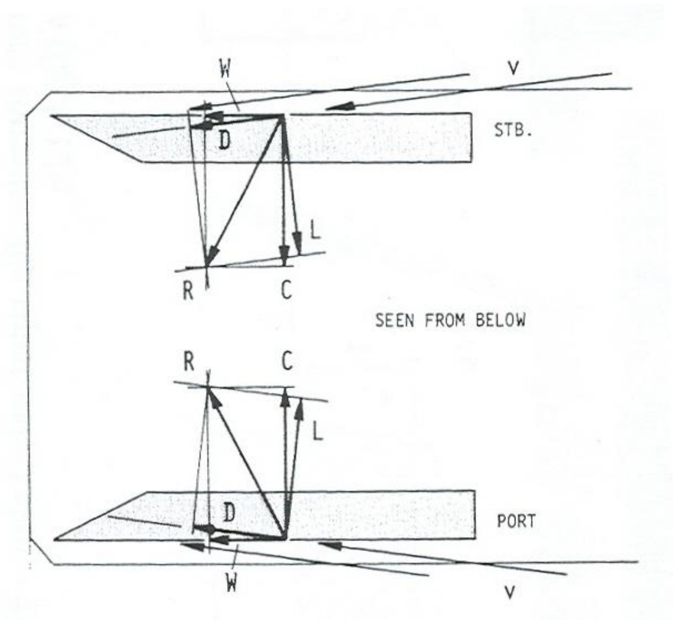



**Fig. 43. Distribution of forces on stern skegs (Brix, [11])**

The effect of skegs on course stability of pod driven ship was investigated by Kobylinski [13] and by Kobylinski & Nowicki [14]. In this investigation large manned model of the gas carrier was used. The test were performed in open water (lake) and the model used for testing POD propulsion was manufactured in the 1:24 scale. The model was fitted either with single pushing POD or twin PODs with pulling propellers. The experiments comprised *inter allia* standard manoeuvrability tests such as turning circle tests, pull-out tests and zig-zag tests. Tests were performed following recommendation of IMO (IMO 2002), however the range of rudder (POD) angles was extended up to $90^0$. The test showed clearly important stabilizing effect of skegs. The model without skegs was unstable and became more stable with increased areas of fins or with combination of fins and skegs. The report containing all relevant results of these test see the references.

Fig.44. (Brix [11]) shows record of course keeping behaviour of a towed barge in function of time. The tests were performed in a towing tank and the towline was attached to the fixed point on the carriage, therefore towed vessel-tug interaction was neglected. Four damping characteristics were shown:

- a) immediately damped yawing motion D=1
- b) damped yawing motion D<1
- c) zero damping D=0
- d) negative damping D<0

Record a) shows optimum damping of the yawing motion after a deflection d. Record b) indicates small damping, whereas from record c) and d) zero or negative damping constant or increased amplitudes of yawing are seen.

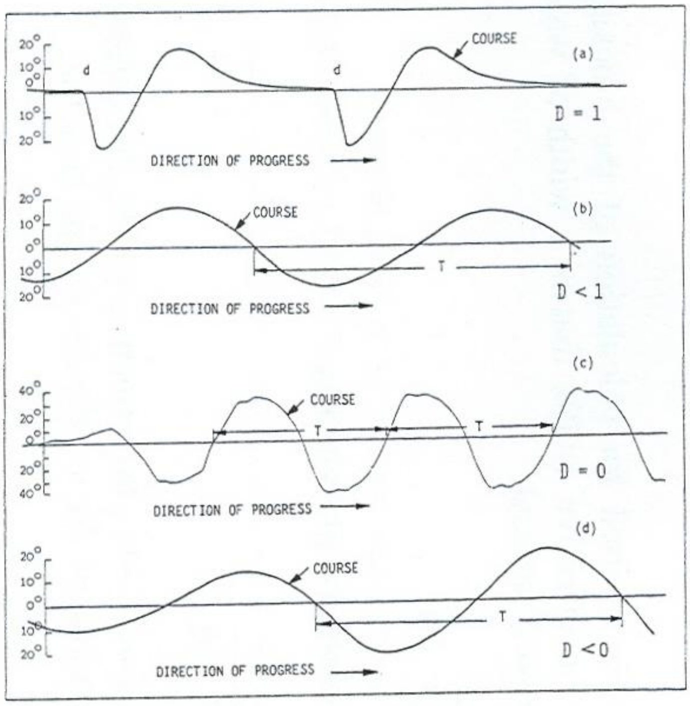



**Fig.44. Damping characteristics of towed ship**

As there are no data available on the motion behavior of pod driven ships when towing or being towed, special tests were arranged in the SHRTC in June 2010. Two large manned models were used for this purpose: the single screw tanker (WARTA) and the gas carrier (DORCHESTER LADY) driven by two pod propulsors. Data for both models are shown in the Table 7.

*Table 7. Data for models used in towing tests*

| | Tanker | LNG carrier |
|---|---|---|
| $L_{PP}$ [m] | 12.21 | 11.33 |
| B [m] | 2.0 | 1.80 |
| T [m] | 0.64 | 0.50 |
| $C_B$ | 0.844 | 0.79 |
| D [t] | 12.49 | 8.21 |
| TDW(ship) | 148 000 | 140 000 $m^3$ |
| Model scale | 24 | 24 |

Three lengths of hawser were used, namely 21, 14 and 7 m in both situations, where pod driven ship LNG carrier was on tow or was towing. Towing speed was about 7 knots (corresponding to full scale)

Fig.45 shows yaw angle (course deviation) versus time for the situation where LNG carrier was towing, and Fig.46 shows the situation where LNG carrier was on tow.

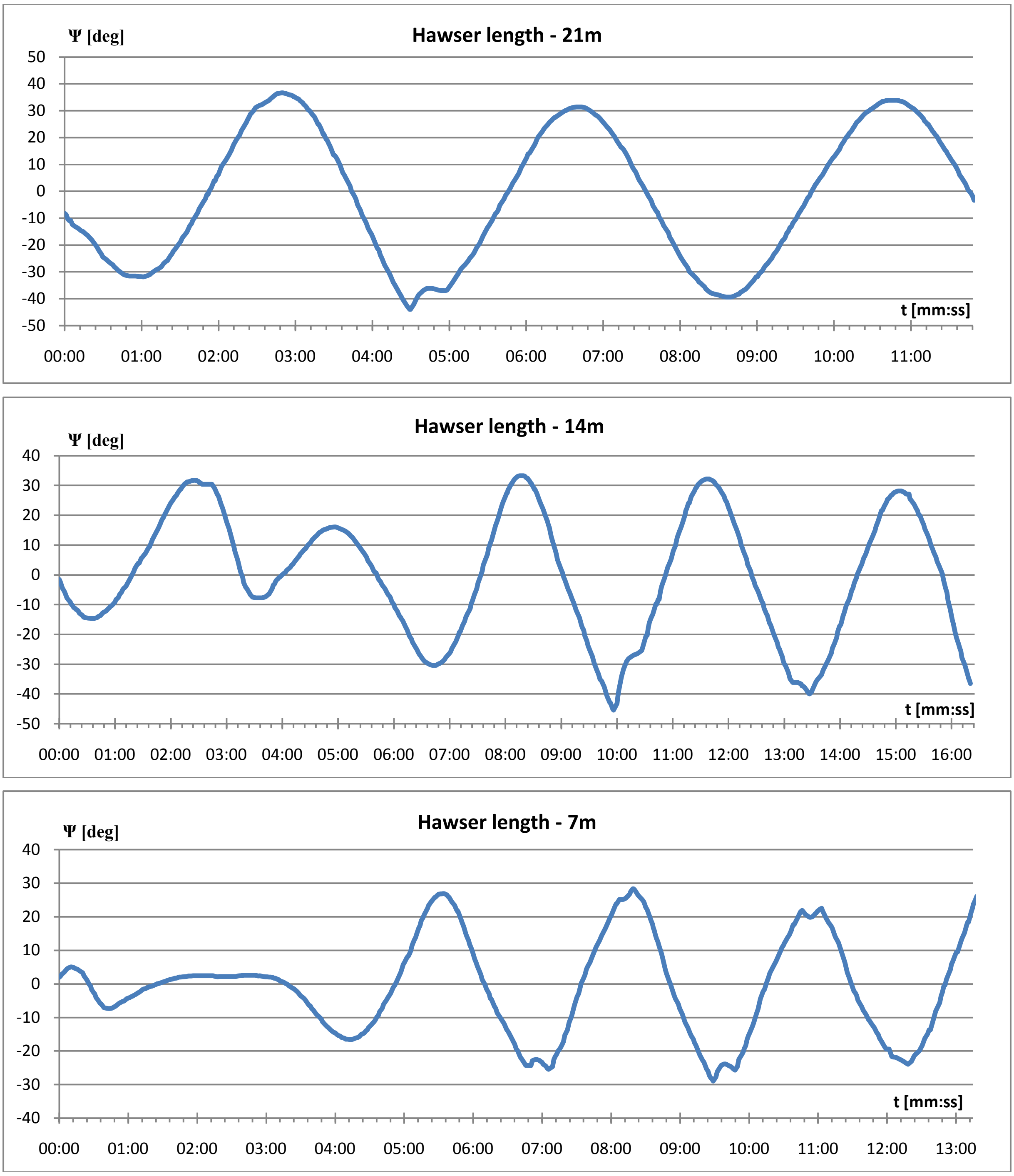



**Fig.45. Record of yaw angle versus time for tanker on tow**

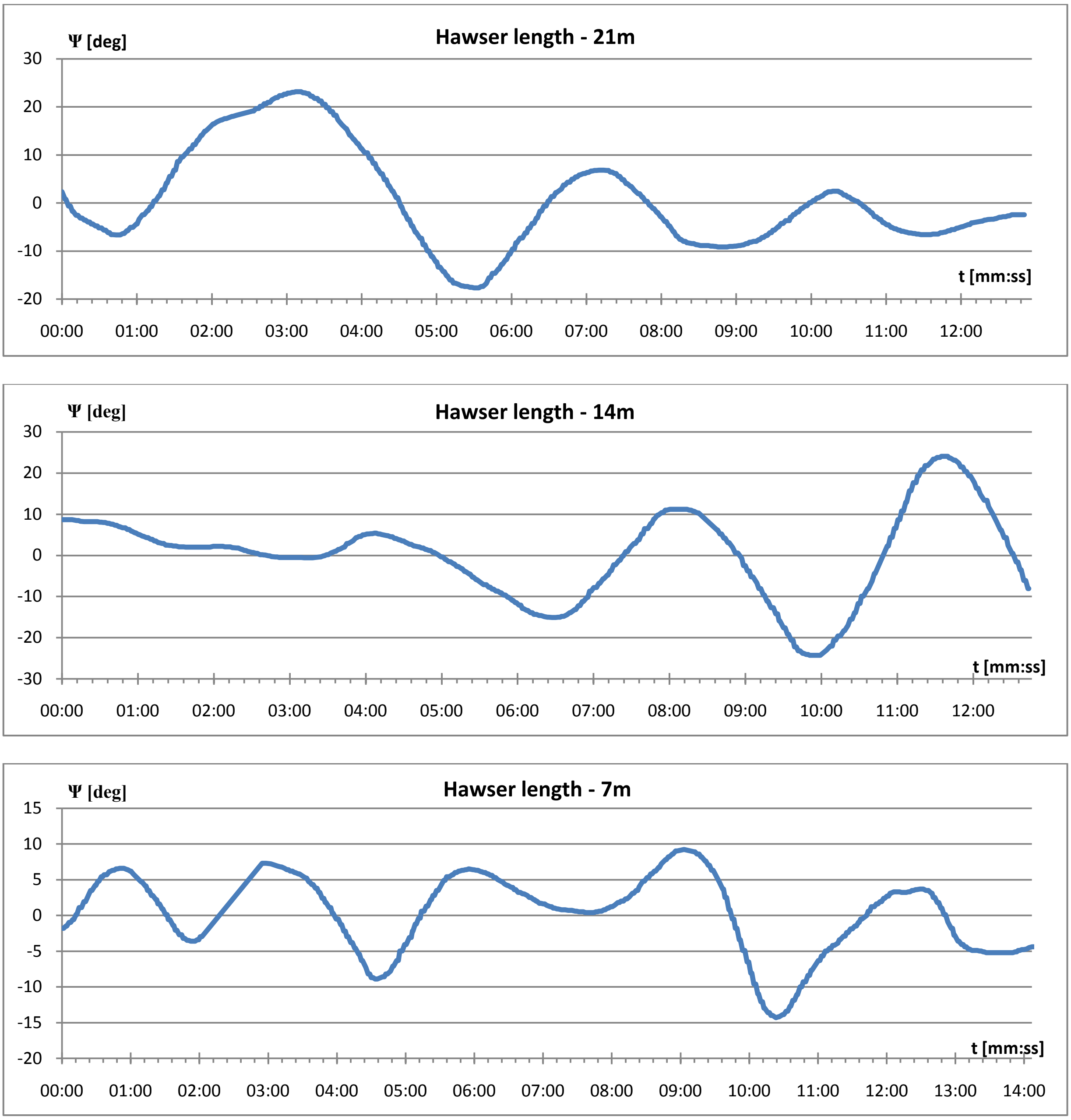



**Fig.46. Record of yaw angle versus time for LNG carrier on tow**

Results of these test did show that when pod driven ship is towing there were no difficulties in keeping the ship on straight course in spite which length of hawser was used. When the pod driven ship was on tow, the best situation was with the longest hawser – the damping was positive and the yawing motion is reducing in time reaching almost stable situation.

With shorter length of hawser the LNG carrier was unstable with zero or negative damping. This was in spite of the fact that LNG carrier was fitted with rather large skegs and when used self propelled had good course keeping characteristics, although was directionally unstable.

The results of extensive test with model of this LNG carrier did show that the pod driven ships having large block coefficient are inherently unstable on straight course and only installation of a combination of skegs and fins did improve this situation.

The simulation tests of pod driven cruise ships showed that those ships are directionally stable (compare Table 3), but model tests or simulation test on how they behave on tow are not available.

## 1.5 Assisted braking including indirect mode

Escort operations performed over long distances and relatively high speeds require escort tugs. All escort tugs have omnidirectional propulsion, (Voith-Schneider, Schottel or azipod type) The main advantage of escort tugs is the possibility to quickly develop high steering and braking forces to a ship when needed.

Steering forces can be developed at high speeds exceeding 10 knots. In this case tugs are working in the indirect moder (in case of failure or human error.).

The distribution of forces acting in the indirect towing mode in escort operations is shown in Fig.47.

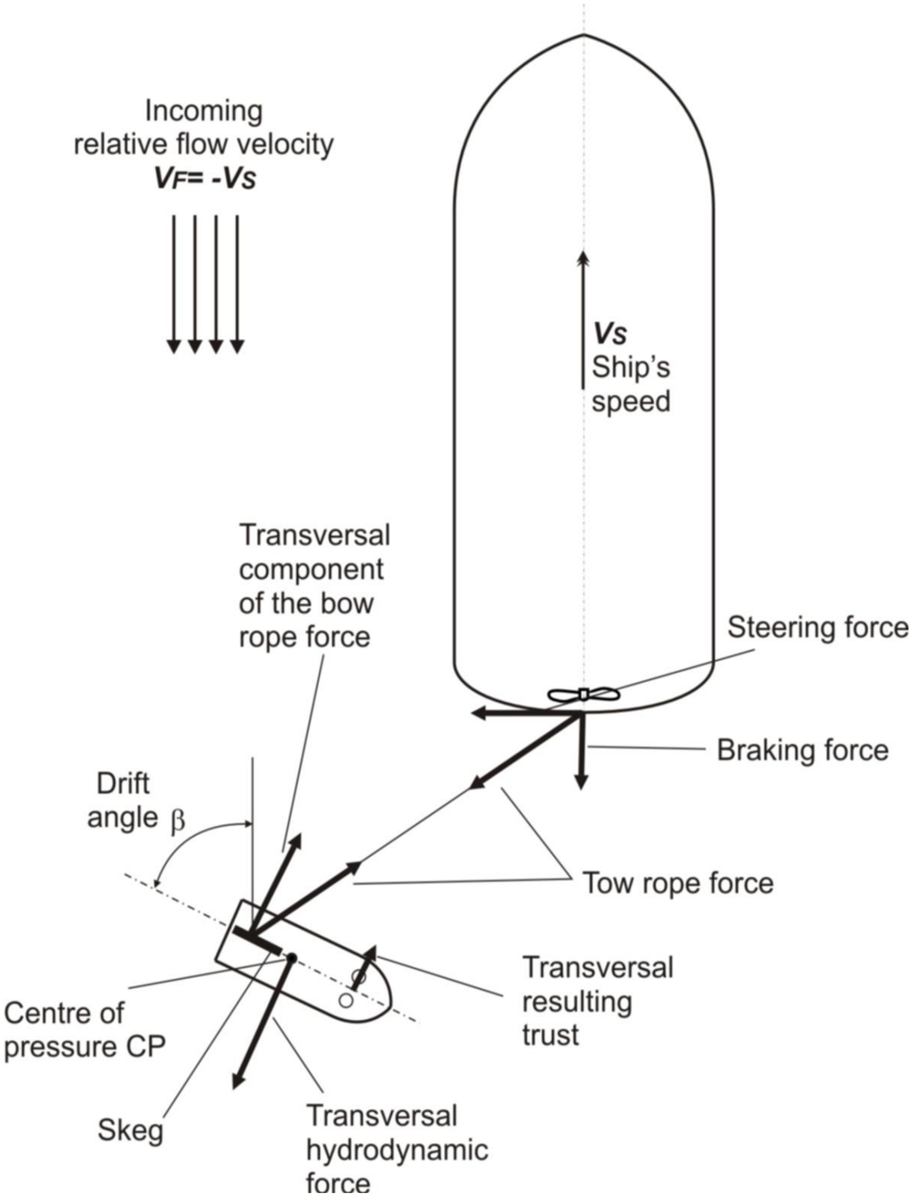



**Fig. 47. Distribution of forces in the indirect towing mode**

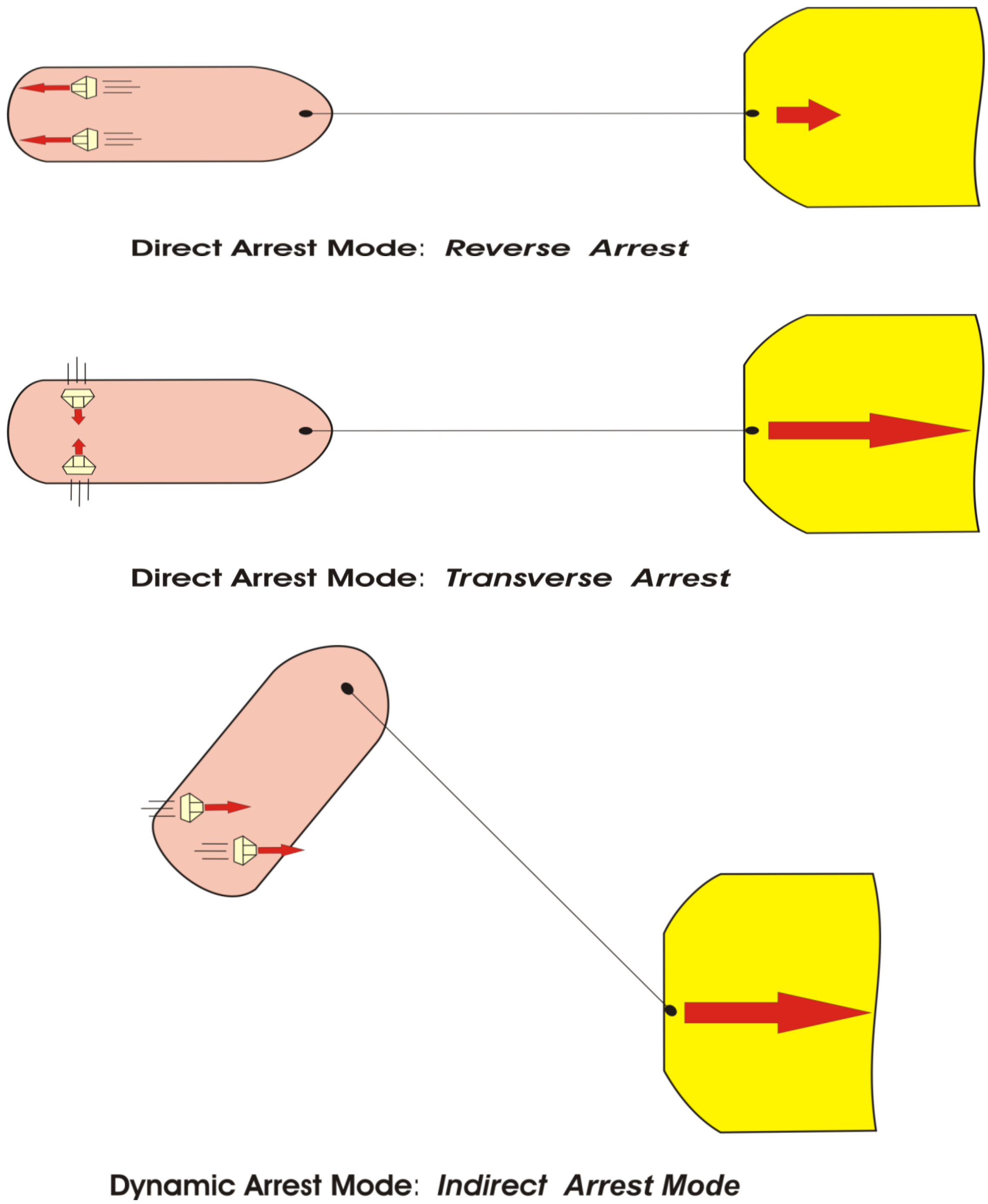



**Fig 48. Schematic presentation of different arrest modes.**

Fig. 48 shows schematic presentation of the three arrest (braking ) modes, direct and indirect with azipod driven tug. Figs. 49 and 50 show different phases of braking manoeuvre, first one, where rudder is blocked on SB and the ship is stopped by hard turn, the second with black-out occurred on board the ship and tug assists braking keeping straight course until ship stops. In both cases indirect mode is used at the initial phase of manoeuvre.

The paper by Capt. Dough Pine (MITAGS-PMI) “An introduction to reverse tractor Z-drive towing modes for pilots” is attached as Appendix 1.

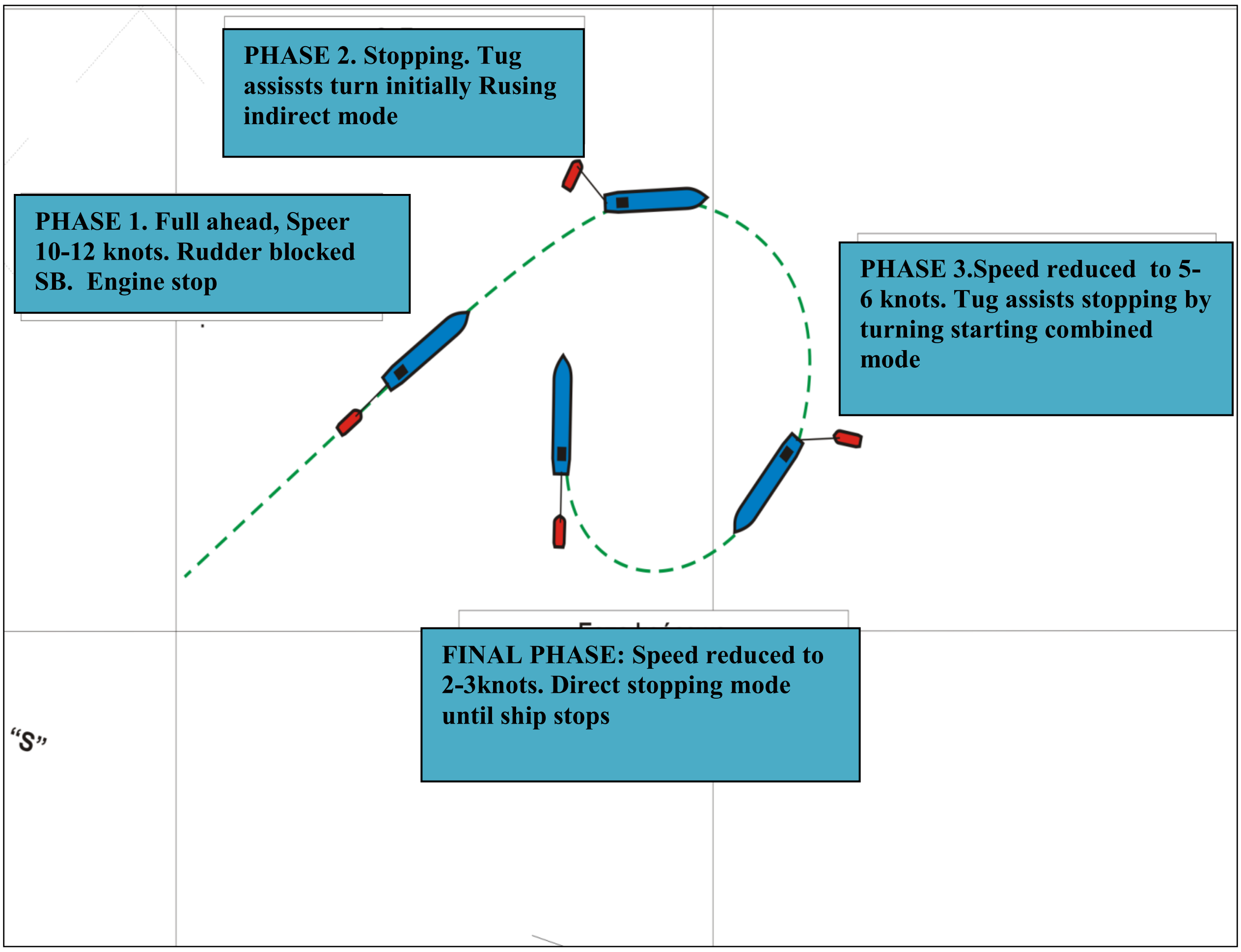



**Fig.49. Assisted braking using hard turn**

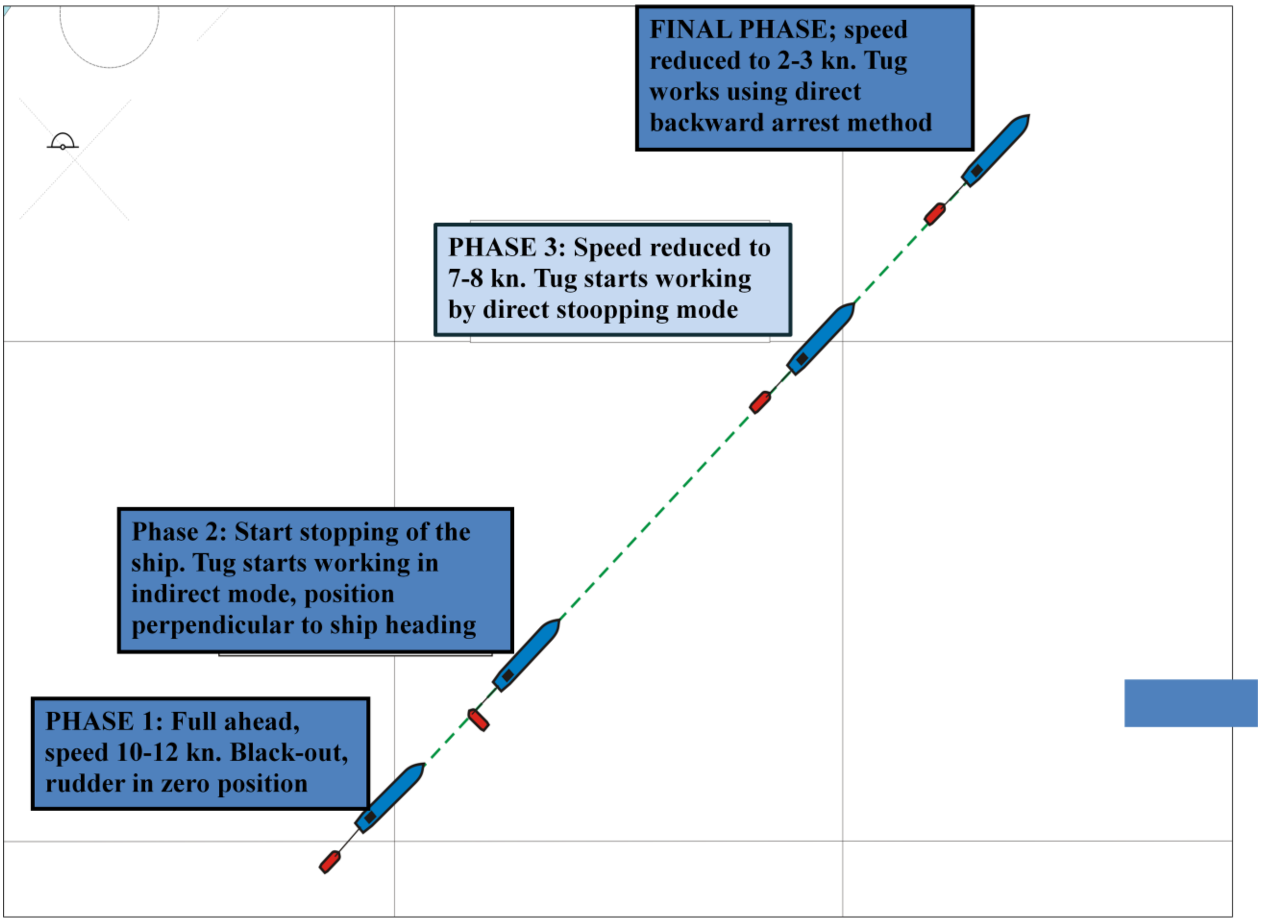



**Fig. 50. Assisted braking keeping the ship on straight course**

## 1.6. Tugs operating near the stern of pod driven ships

Little information is available on the very specific problem of tugs operating near the stern of pod driven ships. It is supposed that there would be not much difference between working conditions for the tug working near the stern of conventional or pod driven ships.

Because of lack of other relevant information the paper headed : "Tug operating near the stern of Pod-driven ships" is included as the Appendix 2.

# PART 2

**SURVEY OF CAPABILITIES OF EXISTING SIMULATORS, EITHER FULL MISSION BRIDGE SIMULATORS (FMBS) OR MANNED MODELS SIMULATORS (MMS) TO SIMULATE THE ABOVE EFFECTS**

## 2.1.Full mission bridge simulators (FMBS)

### 2.1.1. General remarks

Practically all Full Mission Bridge simulators capable to simulate manoeuvring and ship handling characteristics in the real time are also capable to simulate manoeuvrability of pod driven ships provided respective data on hydrodynamic derivatives of pod driven ships are available and fed into the computer programs. As shown in the Part 1 of this reports the majority of influencing factors affecting ship handling that were discussed are included at least in the most advanced FMBS.

General remarks on modeling of ship motions in restricted channels on marine simulators including ship to ship interaction effects presented by V. Ankudinov are given below:

This paragraph presents a short general outline, results and implementation on TRANSAS Maneuvering Simulation System of the novel modeling technique allowing a prediction of almost any obstacle effect on a maneuvering ship, including bank, channel and proximity to other vessels during passing maneuver effects.

The technique is based on the generalized flow/ pressure functional describing motion effects and variable pressure field of maneuvering ship in the restricted channel of variable bottom and banks in the presence of other stationary or moving ships. A very general structure of this functional includes effects of the ship forward and lateral velocities and yaw rate, as well as effects of the propeller (rotational velocities in ahead and astern motions and wake) and effects of the rudder and thrusters. The model structure, qualitative estimation of its principal geometric and hydrodynamic parameters have been developed on the basis of integrated theoretical and empirical approach using numerical analysis and results of specially designed model tests modeling a ship in the channel (or near vertical or inclined wall) with sloping banks and inclined bottoms.

This procedure allowed to quantify effects of the ship hull and channel geometry (with short or long banks of various configurations) and ship motion parameters, effects of ship heading, depth to draft ratio, Froude number effects (based on both, ship length and water depth), channel blockage effects and proximity of other ships or solid obstacles on flow field, pressure and resulting hydrodynamic forces. The technique comprises elements of system identifications and can be used in the range of the tested or numerically computed parameters.

The time dependant algorithm describing the hydrodynamic forces acting on a turning vessel due to the restricted channel is installed on the maneuvering simulators used for training and research. The hydrodynamic forces are estimated by integrating the pressure/ flow field along the hull in real time. During the simulation twelve or more velocity vectors along the ship hull are estimating the flow field (and then the pressure) between the underwater portion of the hull and channel boundaries. The pressure field and hydrodynamic forces will change if position and velocities of the ship hull, as well as parameters of the channel and other ships will change.

The developed technique is fairly complex and best suited for solid unmovable objects in the channel (walls, moored ships). The modeling of proximity of other maneuvering ships of various types moving with various heading angles and velocities needs further refinement, although simple cases of ships on parallel course are modeled fairly well.

There is, however, little direct information from FMB simulators on whether they currently have installed programs to simulate manoeuvrability of pod driven ships and, if so, which influencing factors are included.

Direct or indirect information on the capability to simulate manoeuvrability of pod driven ships taking account of the majority influencing factors is available from the following FMB simulators:[1]

- MITAGS
- TRANSAS (Annex 2)
- NS 5000 simulator by Rheinmetall Defense Electronics [27] (it is not known which simulator centres use this type of simulator)
- DMI -Danish Maritime Institute, Lyngby
- Australian Maritime College [9]

Special simulation programs of azipod driven tugs are available at following simulator centres:

- Maritime Institute of Technology, MITAGS, Washington Di, USA: 2 Full-Bridge 360 degree view Simulators and Tug simulator (Annex 3)
- Pacific Maritime Institute, PMI, Seattle, USA: 2 Full-Bridge Simulators and Tug Simulator
- Marine Engineering School, MEBA, Easton, Maryland, USA: 2 Full- Bridge Simulators and 2 Tug simulators
- Georgian Great Lakes Maritime College, Canada, 4 Full-Scale Bridge Simulators in Network. Bridge layouts allow simulation of practically any ship types including Tugs with all existing drives (FPP, CPP, Steering Nozzle, Pods, Voith – Schneder, etc), Tows, and many others.

The above lists are not complete and certainly many more simulator centres have capability to simulate manoeuvrability of pod driven ships taking account most of the above mentioned factors as well as of tugs action.

Detailed presentation of capabilities of TRANSAS simulator was prepared by V. Akudinov . This presentation was included as Annex 4.

Annex 5 contains presentation by MITAGS – Tractor tug familiarization.

There is even less information on the results of validation of computer programs used in FMBS and on comparison of results of simulation and full-scale tests.

[1] More information will be available after responses to the questionnaire send to simulator centres will be returned.

### 2.1.2.Validation of capabilities of simulation module ANS 5000

There is, however, one detailed information published by de Mello Petey [27] and by Heinke [28] on the simulation module ANS 5000 developed by Rheimetall Defence Electronics GmbH, Bremen, simulating manoeuvring capabilities of POD driven ships. This code takes into account the following:

- Propeller thrust
- Transverse propeller force
- Lift and drag forces of the POD body
- Interaction effects between different POD units
- Interaction effects between POD and hull, and
- Shallow water effects

The method of taking account interaction effects between two POD units and between POD and the ship hull is described in the preliminary report on task 2.3 and is not repeated here. The method of taking account of shallow water used in the simulation module referred to is not known.

The high level of accuracy achieved by the simulation module was proved by validation tests performed with pollution control ship ARKONA (L= 69.2m). The example of comparison of simulated and measured results of the stopping manoeuvre where at full speed both POD were commanded to zero RPM is shown in Fig. 51 (Taken from the reference [27]).

The tables 8 to 12 show a comparison between simulated and measured characteristics of turning circle tests and of zig-zag tests (remark: t90 is the time required for a $90^0$ heading change, to 180 for a $180^0$ change etc) of the passenger ship EUROPA (L=198.6m) Those test however, were conducted in the presence of Beaufort 5 to 6 wind which may influence the results. But in general the accuracy of simulation appears to be good.

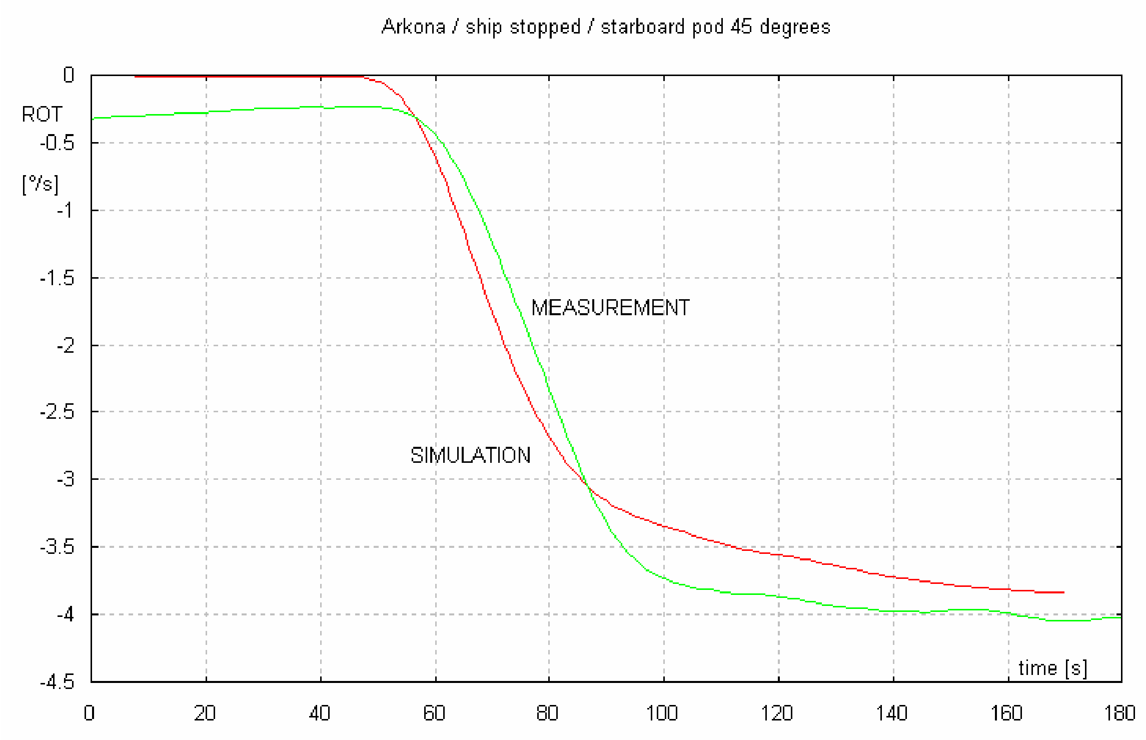



**Fig 51. Comparison of simulated and measured characteristics of stopping moeuvre ARKONA ship (Ref. 27 )**

*Table 8. Turning circle tests with both pods at an angle $35^0$ (EUROPA)*

| | Manoeuvre to port | | Manoeuvre to starboard | |
|---|---|---|---|---|
| | Simulated | Actual | Simulated | Actual |
| Starting speed [knots] | 21.40 | | 11.40 | |
| Engine[%] | 100 | | 60 | |
| Rudder angle [deg] | 35.0 | | -35.0 | |
| Adcance [m] | 404.0 | 379.6 | 333.0 | 364.0 |
| Transfer [m] | 165.0 | 159.1 | 167.0 | 164.3 |
| Tactical diameter [m] | 375.0 | 392.1 | 382.5 | 398.7 |
| Turning circle diameter [m] | 320.0 | 313.7 | 323.5 | 320.3 |
| Steady speed at turn [knts] | 6.40 | 6.59 | 3.90 | 4.38 |
| t90 [s] | 56 | 54 | 91 | 96 |
| t180 [s] | 117 | 120 | 182 | 203 |
| t270 [s] | | 192 | | 314 |
| t360 [s] | 260 | 264 | 397 | 425 |

***Table 9. Turning circle tests with starboard pod only at $35^0$*** *(EUROPA)*

| | Manoeuvre to port | | Manoeuvre to starboard | |
|---|---|---|---|---|
| | Simulated | Actual | Simulated | Actual |
| Starting speed [knots] | 10.50 | | 10.50 | |
| Engine[%] | 80 | | 80 | |
| Rudder angle [deg] | 35 | | -35 | |
| Adcance [m] | 399.0 | 430.6 | 402.0 | 434.0 |
| Transfer [m] | 205.0 | 210.5 | 201.0 | 210.7 |
| Tactical diameter [m] | 497.0 | 480.3 | 466.0 | 492.2 |
| Turning circle diameter [m] | 496.0 | 403.2 | 506.0 | 419.7 |
| Steady speed at turn [knts] | 4.80 | 5.04 | 4.80 | 5.06 |
| t90 [s] | 115 | 118 | 117 | 121 |
| t180 [s] | 217 | 234 | 244 | 242 |
| t270 [s] | | 356 | | 368 |
| t360 [s] | 471 | 478 | 529 | 494 |

*Table 10. Turning circle tests with starboard pod only at $60^0$ (EUROPA)*

| | Manoeuvre to port | | Manoeuvre to starboard | |
|---|---|---|---|---|
| | Simulated | Actual | Simulated | Actual |
| Starting speed [knots] | 10.50 | | 10.50 | |
| Engine[%] | 80 | | 80 | |
| Rudder angle [deg] | 60 | | -60 | |
| Adcance [m] | 309 | 377.5 | 322 | 376.1 |
| Transfer [m] | 133 | 143.6 | 136 | 139.3 |
| Tactical diameter [m] | 287 | 293.0 | 253 | 276.4 |
| Turning circle diameter [m] | | 53.3 | | 33.5 |
| Steady speed at turn [knts] | 1 | 0.46 | 2 | 029 |
| t90 [s] | 99 | 112 | 102 | 114 |
| t180 [s] | 191 | 206 | 198 | 207 |
| t270 [s] | | 294 | | 296 |
| t360 [s] | 402 | 377 | 423 | 382 |

*Table 11. Zig-zag test $10^0/10^0$ with both pods. (EUROPA)*

| | Simulated | Actual |
|---|---|---|
| $1^{st}$ overshoot [deg] | 6.5 | 6.8 |
| $2^{nd}$ overshoot [deg] | 8.1 | 9.0 |
| $3^{rd}$ overshoot [deg] | 7.9 | 8.3 |
| $t_{(1st\ overshoot)}$ [deg] | 36 | 29 |
| $t_{(2st\ overshoot)}$ [deg] | 94 | 76 |
| $t_{(3st\ overshoot)}$ [deg] | 146 | 135 |

*Table 12. Zig-zag test $10^0/10^0$ with starboard pod only. (EUROPA)*

| | Simulated | Actual |
|---|---|---|
| $1^{st}$ overshoot [deg] | 3.9 | 3.9 |
| $2^{nd}$ overshoot [deg] | 5.1 | 6.0 |
| $3^{rd}$ overshoot [deg] | 4.0 | 5.1 |
| $t_{(1st\ overshoot)}$ [deg] | 71 | 69 |
| $t_{(2st\ overshoot)}$ [deg] | 180 | 178 |
| $t_{(3st\ overshoot)}$ [deg] | 298 | 295 |

### 2.1.3. Simulation capability of simulating shallow water, bank effect and ship to ship interaction. Responses from 4 simulator facilities.

This part was prepared by Dr. Andreas Gronarz from DST

In order to compare the capabilities of the different simulators to deal with the special effects a questionnaire was developed where special manoeuvres were described. The results should be noted in tables which were processed in anonymous and nondimensional form.

The main page of this questionnaire described the strategy and the ship type which was used in the simulations.

Target group:Simulator Manufacturers (can be delegated to an operator of the simulator)

Review of existing ship simulator capabilities Task 2.2

Name of manufacturer: ……………………………………………….

**Note:** It is worthwhile asking either "Does your simulator consider the shallow water effect" or "Describe in detail, how your simulator considers the shallow water effect" because the answer always will be "yes" and no answer will be given to the details. To overcome this problem, the target group is asked for the outcome of simulations varying the water depth (or other specific functionalities).

Specific simulations with a loaded **Panamax container carrier** (or a similar vessel) shall be carried out and the results be given in tables. This shall lead to a quantitative assessment of the consideration of the certain effects.

Details of the vessel:

Length L [m]…………….Breadth B [m] …………….Draught T [m] …………….

Displacement [t] …………Speed V [kn]…………….

Later pages of the questionnaire contain special manoeuvres to be carried out in order to highlight the special hydrodynamic effects. The manoeuvres have been tested on a simulator to check whether they are easily executable and deliver results which allow a judgement of the special effects. This yields in the tables to be filled which are printed in the separate chapters.

The complete questionnaire was planned to be sent to all manufacturers of simulators listen in the data-collection "Basic groups of interest". Due to the fact that the circulation of the questionnaire and the return of the results exceeded the delivery date of the report partners of the project have been asked to perform the required simulations in advance. 3 of 4 manufacturers / simulator operating facilities have replied and delivered date for the comparison.

Shallow water effect

Theory

The reduction of the under-keel-clearance (UKC) results in a change of the manoeuvring behavior of the ships. This is caused by the change of flow underneath the bottom of the ship, which is hindered when the gap between keel and seabed is becoming closer. Using the main parameters water depth h and ship draught T the shallowness of the water can be described in various nondimensional expressions: h/T, T/h, (h-T)/h, (h-T)/T, h/(h-T) and T/(h-T). The first ones are mostly used and the expression T/h seems to be preferable, because it takes the value T/h = 0 for infinite water depth and T/h = 1 for the grounding condition. The reciprocal formulation has the disadvantage, that the deep water case takes the value of $h/T = \infty$, which is inconvenient in graphical representations of the effect.

Two main outcomes of the effect can be considered and will be compared for the different simulators.

- Speed loss with reduced water depth

  In deep water a ship can reach the highest velocity using constant revolutions of the propeller. When the UKC decreases the speed decreases due to the reduced flow under the keel in longitudinal direction. It can be expected, that with rising values of T/h the speed will be reduced slightly and the loss will be increased significantly when the UKC is rather small.

- Turning circle diameter change

  Due to the reduced cross flow in shallow water the ship build up dri9ft angles and yaw rates as high as in deep water. This will result in an increase of the turning circle diameter with reduced UKC.

Questionnaire

**Shallow water effect:**

Scenario:Unrestricted water without wind & current, 75% EOT, constant water depth.

| Variation | Straight ahead | Turning circle 20° rudder to port | | |
|---|---|---|---|---|
| T/h | Final speed [kn] | Diameter [m] | Final speed [kn] | Yaw rate [°/min] |
| 0 | | | | |
| 0.3 | | | | |
| 0.5 | | | | |
| 0.6 | | | | |
| 0.7 | | | | |
| 0.75 | | | | |
| 0.8 | | | | |
| 0.85 | | | | |

Analysis

The results of the simulations are presented in 2 figures. For the shallow water effect the cases A and B are from the same simulator but different ships have been used while C and D are from the other 2 different simulators.

- Speed loss with reduced water depth

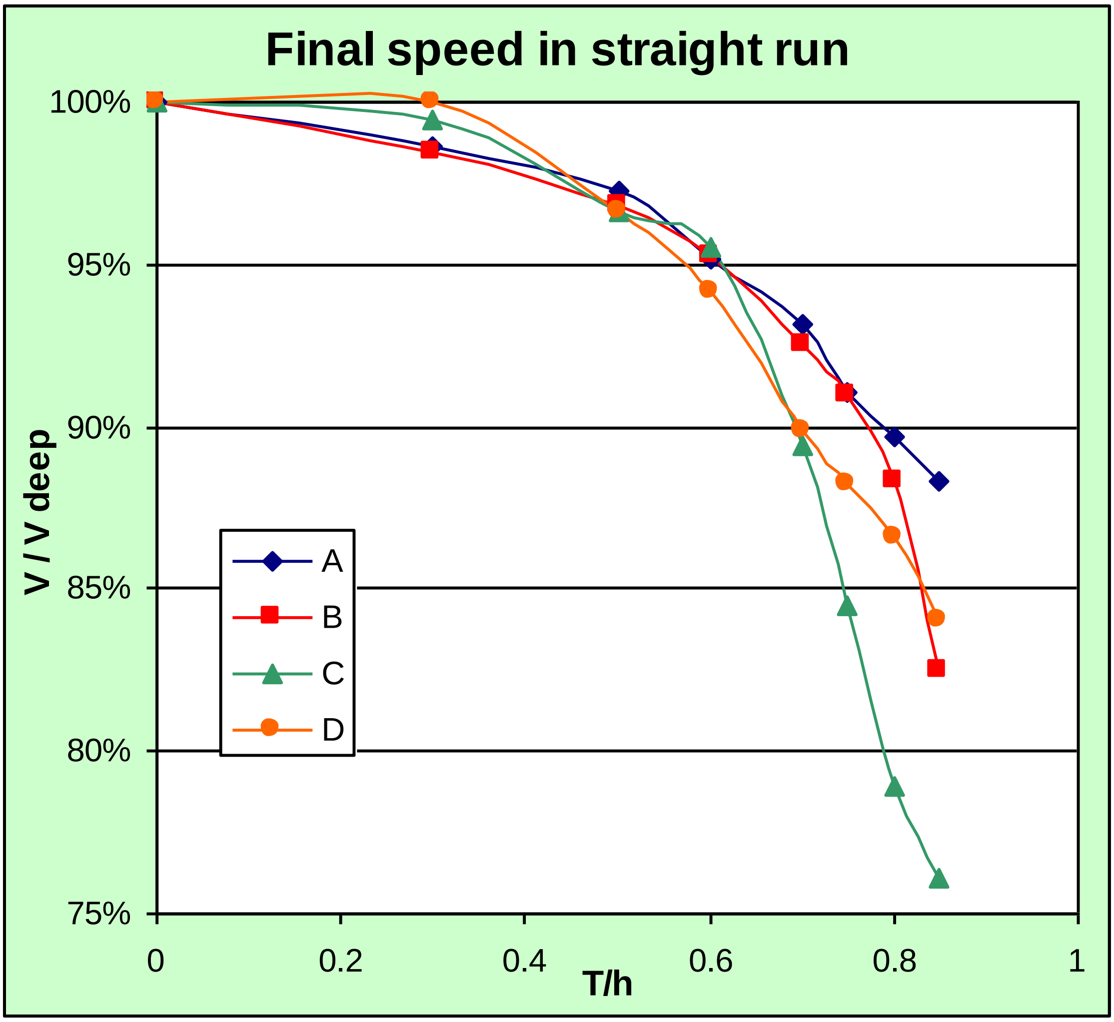



**Fig 52 Speed loss with reduced UKC**

In general all simulators show the expected loss of speed with increasing shallowness of the water as it can be seen in Fig 52. In detail there are some differences which have to be noted. The case T/h = 0.3 means, that the UKC is more than twice the draught. This gap is not close and the speed loss should only be marginal. This is the correct for C and D, but in A and B there is already at significant loss of the speed. On the other hand at very shallow water (T/H ~ 0.8) the speed loss of A and B is not great enough as it comes out for the other simulators, especially for C.

This means that for the simulator A & B the representation of the shallow water effect should be improved.

- Turning circle diameter change

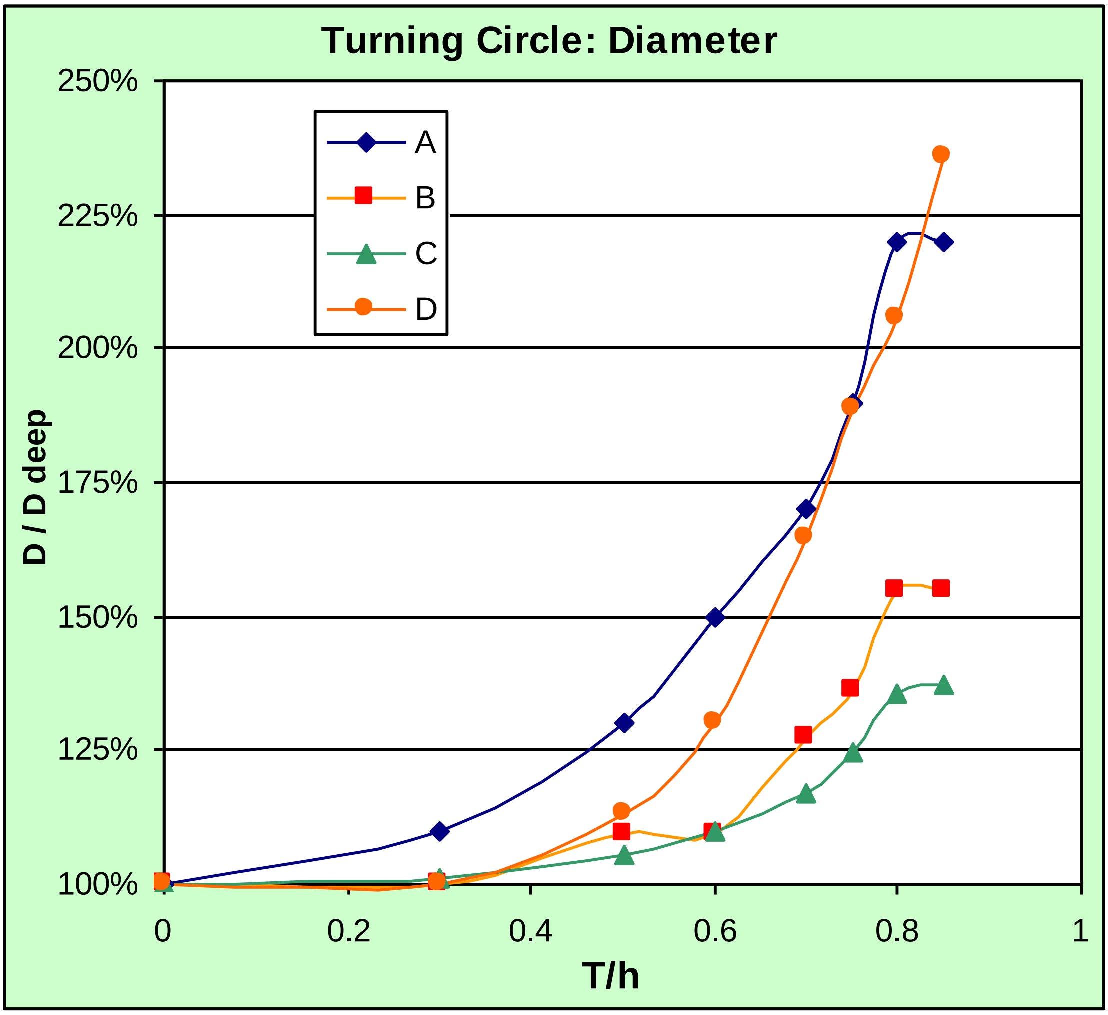



Fig 53. Increase of turning circle diameter with reduced UKC

Also the turning circle diameter shows the expected increase for all simulators. It is remarkable, that only A (not B) shows an increase in diameter with still rather deep water (T/H = 0.3).

Another observation is, that the increase in diameter is significantly different. The range of 35% (C) to 135% (D) increase seems unusual. Maybe the increase is too small for C.

It can be seen, that for A, B and C the curves show nearly constant values for T/h = 0.80 and 0.85. This cannot be expected from theory, which seems to be represented rather good by Simulator D. The reason seems to be a limitation in the shallow water effect in the mathematical motion model of some simulators. This limitation has been built in to prevent extreme motions in cases close to grounding.

## Bank effect

### Theory

A ship moving close to a bank or a vertical wall is affected by an unsymmetrical flow condition. This will result in a lateral force and a yaw moment. The lateral force will attract the ship to the wall following Bernoullis law which predicts a lower pressure for a higher flow speed. The yaw moment created by the unsymmetrical flow is normally an outturning one.

The closer a ship is to a wall or a bank the greater is the effect acting on that ship.

### Questionnaire

**Bank effect:**

Scenario:Deep water without wind & current, 75% EOT, vertical wall on starboard, parallel course, no rudder action

| Variation | After **30** sec | | After **60** sec | |
|---|---|---|---|---|
| Wall distance [m] (side of ship at midship measured to wall) | Wall distance [m] | Course deviation [°] | Wall distance [m] | Course deviation [°] |
| 200 | | | | |
| 100 | | | | |
| 50 | | | | |
| 30 | | | | |
| 20 | | | | |
| 15 | | | | |
| 10 | | | | |

The two values to be observed can be traced back to the lateral force and the yawing moment. While the course change is only dependant on the yawing moment the change in the wall distance in influenced by both. The lateral force creating the attraction to the wall is superimposed by the motion away from the wall due to the course change. This fact is the reason for the very small distance changes at 30 s.

Only for two simulators results are available, because for one type there was no vertical wall available to carry out the tests. Compared with the shallow water effect in chapter 0 the bank effect has been investigated with one ship (A) instead of two

### Analysis

The change in the wall distance and the course deviation are only presented for the time of 60 s – the values for 30 s are only smaller but give the same result

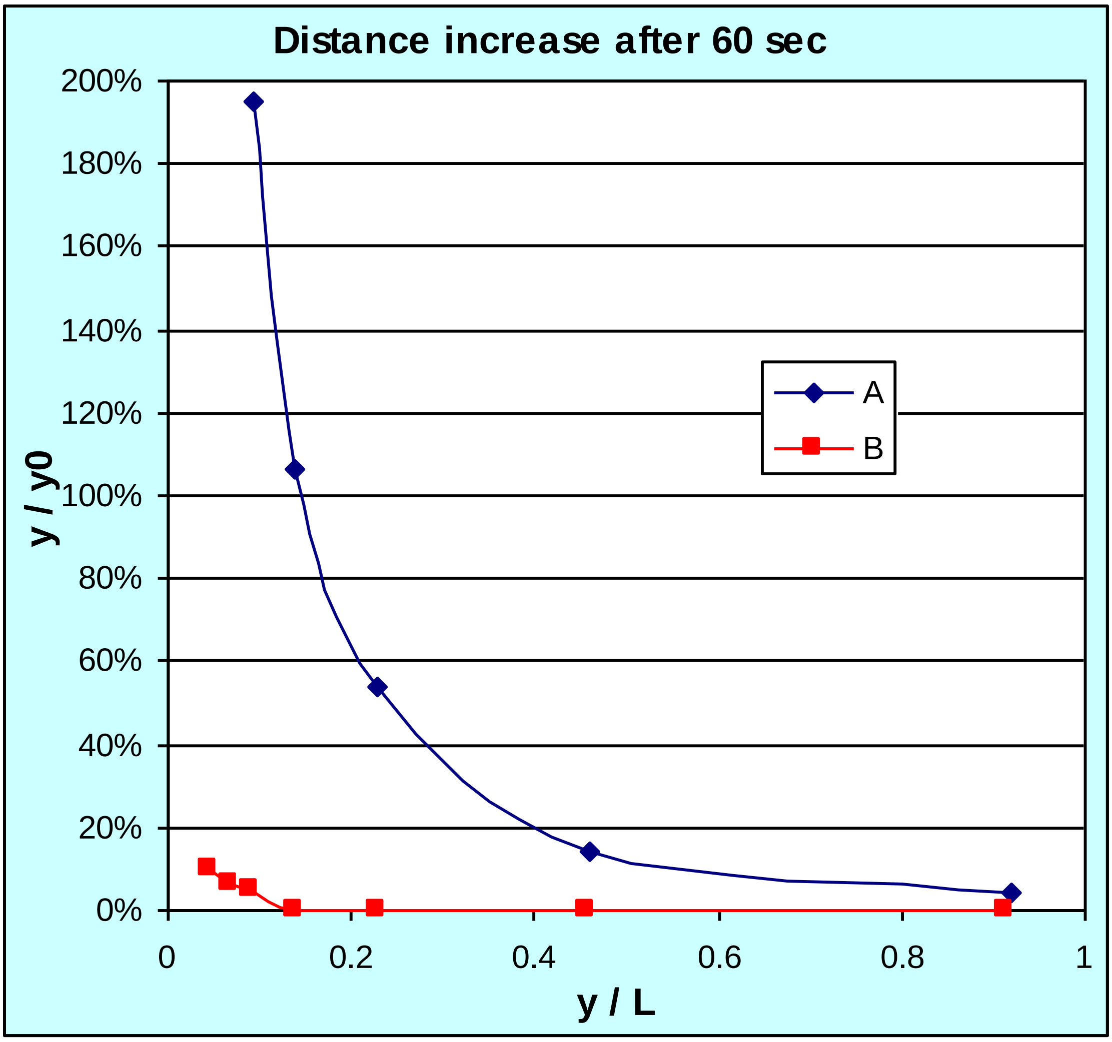



**Fig 54. Distance increase due to bank effect**

On both simulators the bank effect is implemented and results can be obtained. The major difference is the magnitude of the effect. While B only shows a lateral deviation very close to the wall it is ascertainable at already on ship length distance at simulator A. The effect is so strong in A, that no results could be obtained for the wall distances 15 m and 10 m for 60 s because the yawing moment caused the stern of the ship to touch the wall.

As the results were obtained in m without decimal fractions of it the results for B were 0 at the larger wall distances.

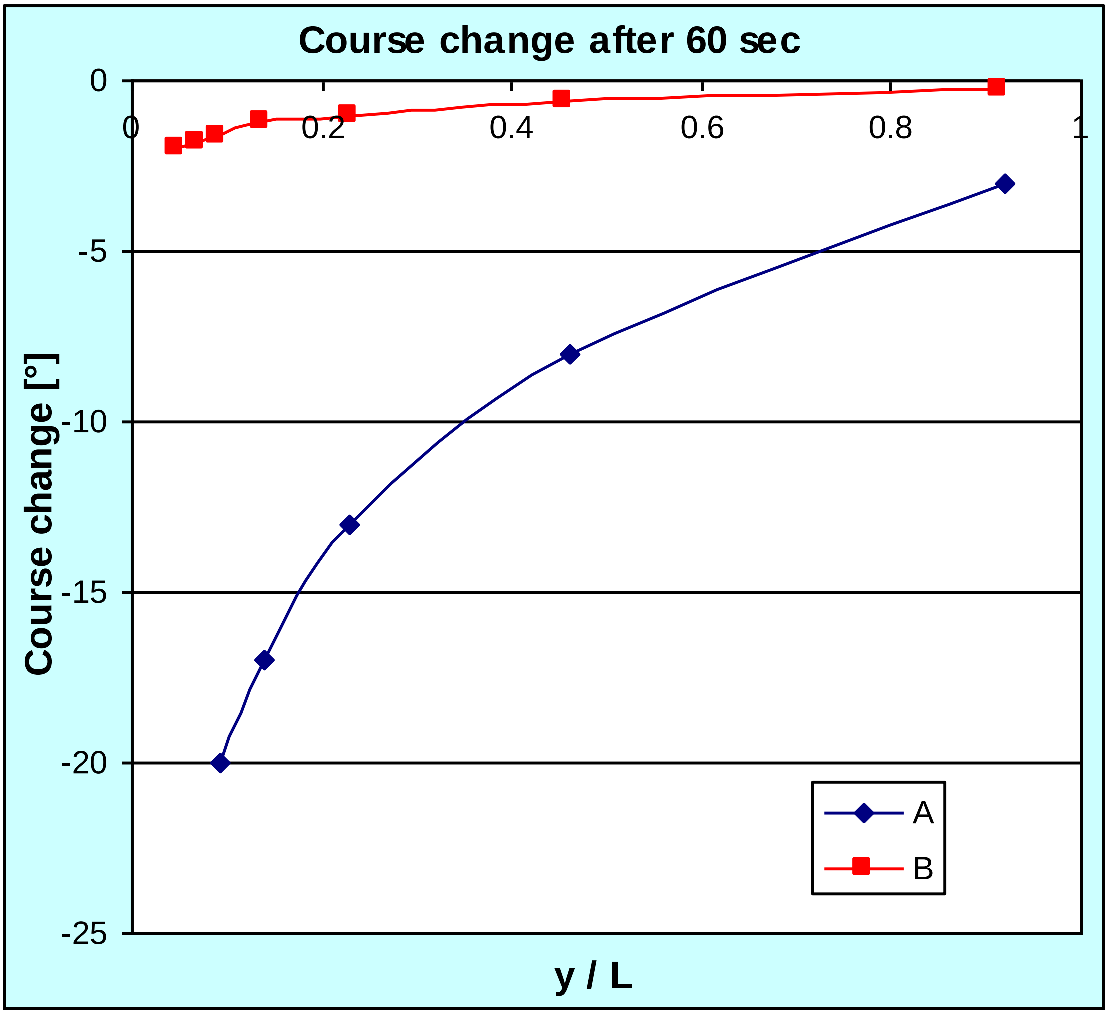



**Fig 55. Course change due to bank effect**

The course change after 60 s shows the same great difference in the magnitude of the bank effect between A and B. Numerically the difference is a factor about 10. Unfortunately there are only two tests available so it cannot be said whether A is too big or B is too small. But, as the bank effect is normally described with formulae and variable coefficients, this discrepancy can be adjusted by measurements or the experience of captains.

## Ship-ship-interaction

### Theory

When two ships pass (overtaking or encountering) also an unsymmetrical flow situation occurs. In comparison to the bank effect, which is constant, this forces and moments induced to the ships are changing with the time according to the variable positions of the ships to each other during the manoeuvre.

Mainly 3 situations are evident at a passing manoeuvre (here encountering):

- Bow to bow position (begin of the manoeuvre)
  Repulsion of the ship and outturning moment
- Midship to midship position
  Attraction of the ships, no yaw moment
- Stern to stern position (end of the manoeuvre)
  Repulsion of the ship and inturning moment

The smaller the distance is, the greater the effect will be.

### Questionnaire

**Ship-Ship interaction:**

Scenario:Deep water without wind & current, 75% EOT, same ship as own ship coming towards own ship, parallel course, no rudder action

| Variation | Bow-bow position | Midship position | Stern-stern position |
|---|---|---|---|
| Passing distance [m] (side of ship to side of ship) | Yaw rate [°/min] | Yaw rate [°/min] | Yaw rate [°/min] |
| 200 | | | |
| 100 | | | |
| 75 | | | |
| 50 | | | |
| 30 | | | |
| 20 | | | |
| 15 | | | |
| 10 | | | |
| 5 | | | |

As it is rather difficult to measure the lateral offset from the original course during an encountering manoeuvre the yaw rate has been chosen as a parameter for the judgement of the quality of the simulation.

Analysis

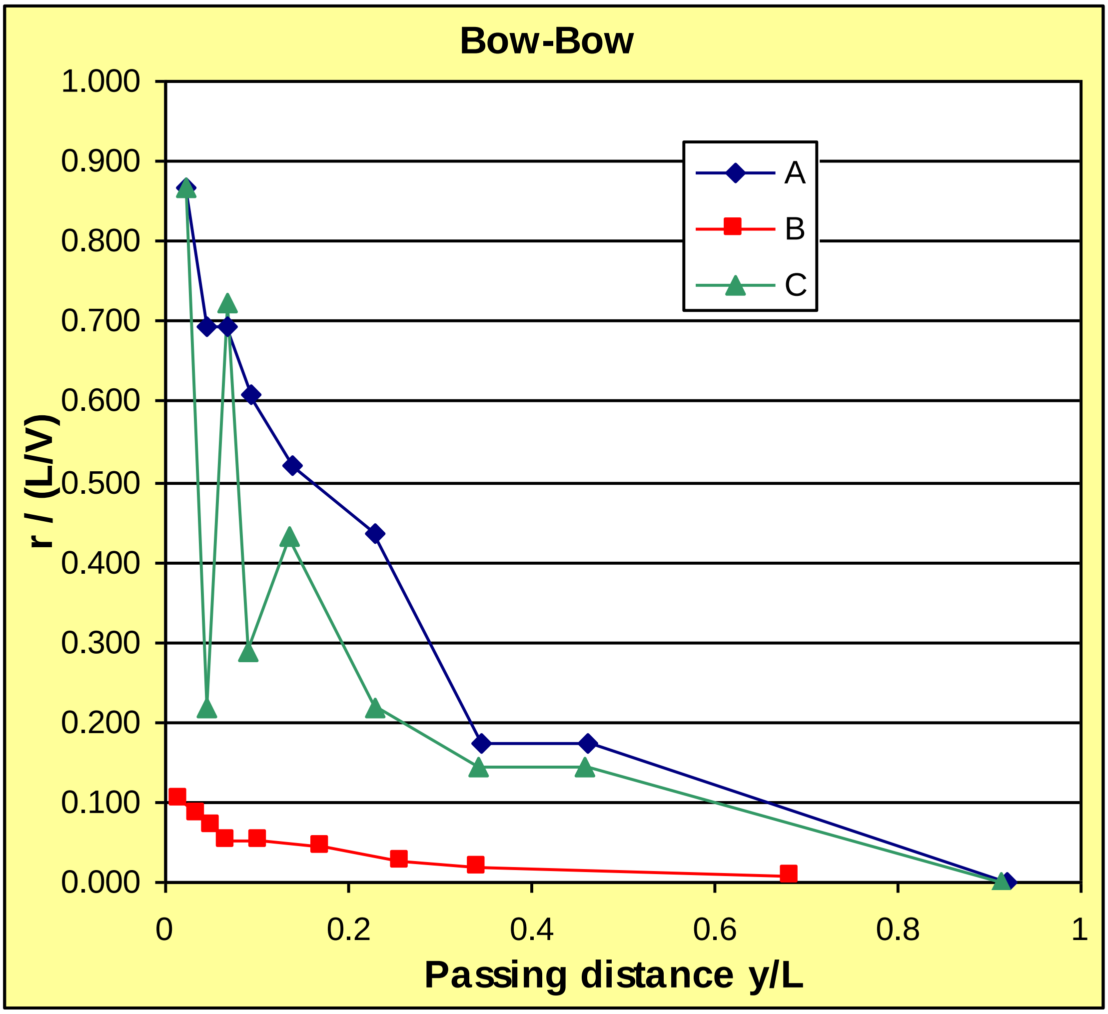



**Fig 16. Yaw rate at beginning of the encountering manoeuvre**

For the simulators A and C it can be said, that the results are quite comparable. At small distances there might be some errors in the numbers for C. The fact that there are equal values at A for two times comes from the accuracy of the date – only full numbers without decimalös ver available for A.

Simulator B gives rather small values. A reason for that might be the fact, that the ship was nearly 50% longer than that used in A and C.

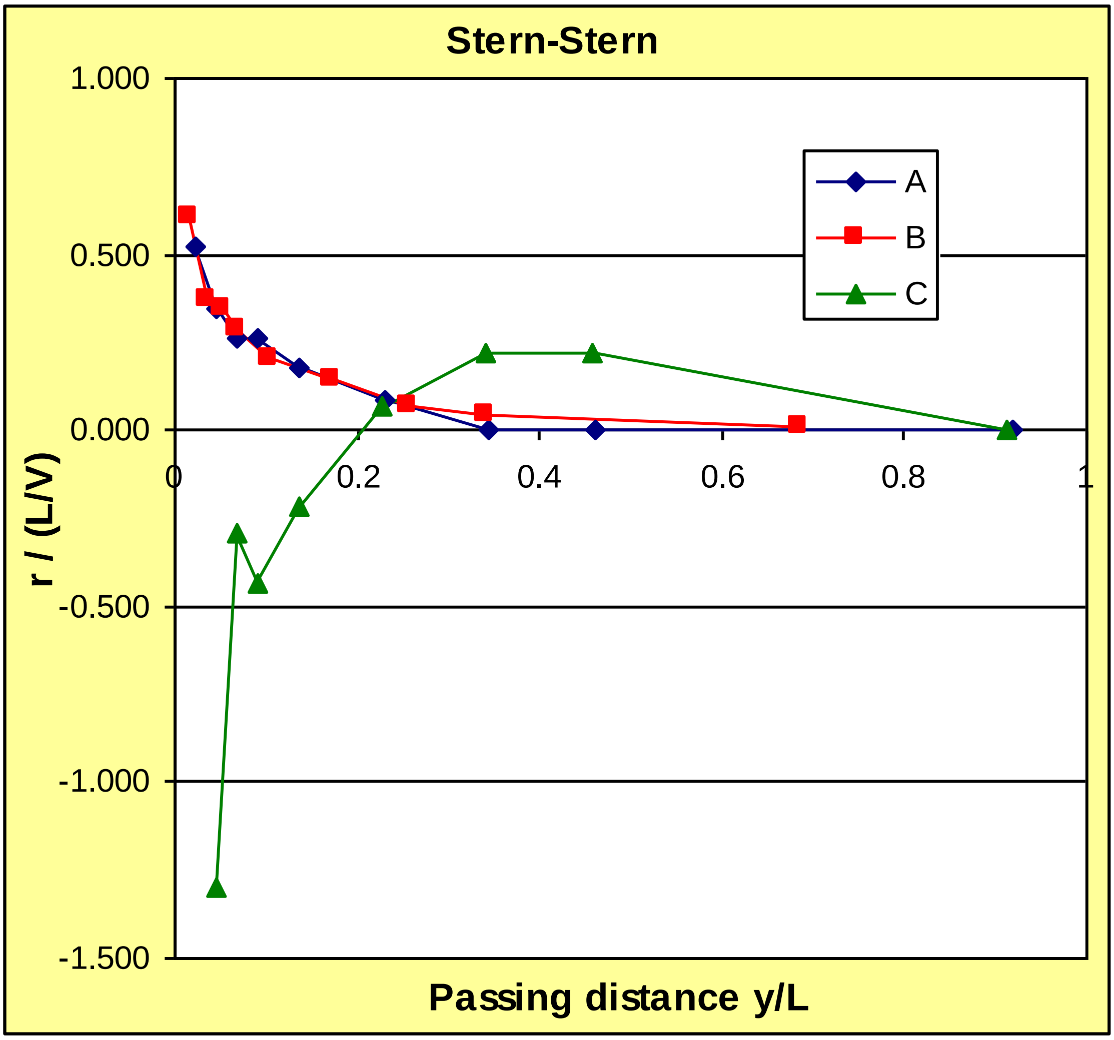



**Fig 57. Yaw rate at end of the encountering manoeuvre**

Compared with Fig 16 this result is quite astonishing. Now A and B show nearly the same result while at the beginning of the manoeuvre the differences were quite big. Most astonishing is the behaviour of the yaw rate at C. It is not to be expected that a sign change of the yaw rate occurs when the passing distance varies.

What is to be expected is a graph like 76 with the difference, that all signs are reversed because now we have (from theory) an inturning instead of an outturning moment.

Because of these unusual facts it is very difficult to judge, which simulator is correct and which predicts wrong results. Again due to lack of enough data no final judgement can be made.

.

Conclusions

- All special hydrodynamic effects are covered from the simulators investigated.
- The magnitude of the effects is sometimes very different.
- The expectations from theory are satisfied mostly.
- The development of the shallow water effect with decreasing water depth is not always modelled correct.
- The magnitude of the bank effect is very different on the two simulators investigated.
- The ship-ship-interaction effect shows reasonable development with the passing distance but some doubtful results during the time of the manoeuvre.

→It is recommended to repeat this part of the investigation and gather more data as for example the time histories of motion parameters position, course and yaw rate

## 2.2. Manned models simulators (MMS)

In the training centres using manned models (MMS), models perform manoeuvres on training areas where situations simulating influencing different factors that affect ships discussed in the Part 1 of this report are physically created. This includes shallow water, bank and canal effect, interaction effects between two or more ships, escorting operations towing operation and tugs work in general.

Both Port Revel and Ilawa training centres that provided information claim that they are capable of simulating the majority of such situations for training using models of conventional ships and in particular for training on pod driven ship models that are available in both centres. Basic data on models of pod driven ships available in both centres are provided in the report on Task 2.1 and are not repeated here.

In order to simulate shallow water, bank and canal effect in the training areas there are arranged areas where water depth is small, canals are dredged and bank effect routes are arranged. With manned models available there is no difficulty to arrange meeting or overtaking situations with two or more ships, also to arrange ship-to-ship and towing manoeuvres. As shown in the report on Task 2.1 at least in Ilawa training centre it is possible to arrange escort operations using one or two ASD or tractor tugs driven with azimuting propulsors.

The capabilities of training centres using manned models with respect to ability to simulate the typical environmental factors discussed in this report are shown by the example of training areas and arrangements of Ilawa training centre [29].

In Ilawa Ship Handling Centre there is a number of artificially prepared training areas that, apart of the standard model routes marked by leading marks, leading lights (at night) and buoys, comprise also routes particularly suitable for training ship handling in canals and shallow and restricted areas. They include:

A. restricted cross-section canal of the length 140m (corresponding to 3.3 km in reality), called Pilot’s Canal. Plan of the canal and its cros-section is shown with two ships at meeting situation is shown in Fig. 58.,

B. wide (corresponding to about 360m width in reality) shallow water canal of the length corresponding to about 1.5 km, where current could be generated from both sides, called Chief’s Canal (Fig.59);

C. long (corresponding to about 2.5 km in reality), narrow deep water waterway comprising several bends, marked by buoys, simulating some routes in fiords and similar areas called Captain’s Canal;

D. narrow fairway restricted from one side by the shore, called Bank Effect Route

E. narrow passages including narrow passage under the bridge;

F. river estuary area where several current generators installed create current.. Several mooring places are provided in the estuary, including sheltered dock. Current pattern and velocities could be adjusted by activating particular current generators, the maximum current velocity correspond to 4 knots in full scale.

G. two locks (one representing Antverp lock)

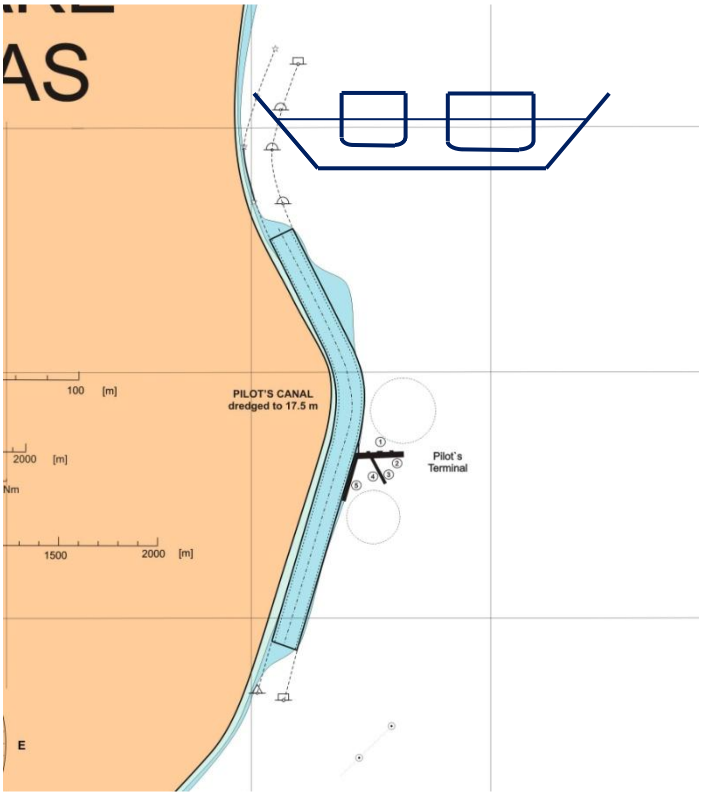



**Fig.58. Plan of the Pilot's Canal with its cros-section**

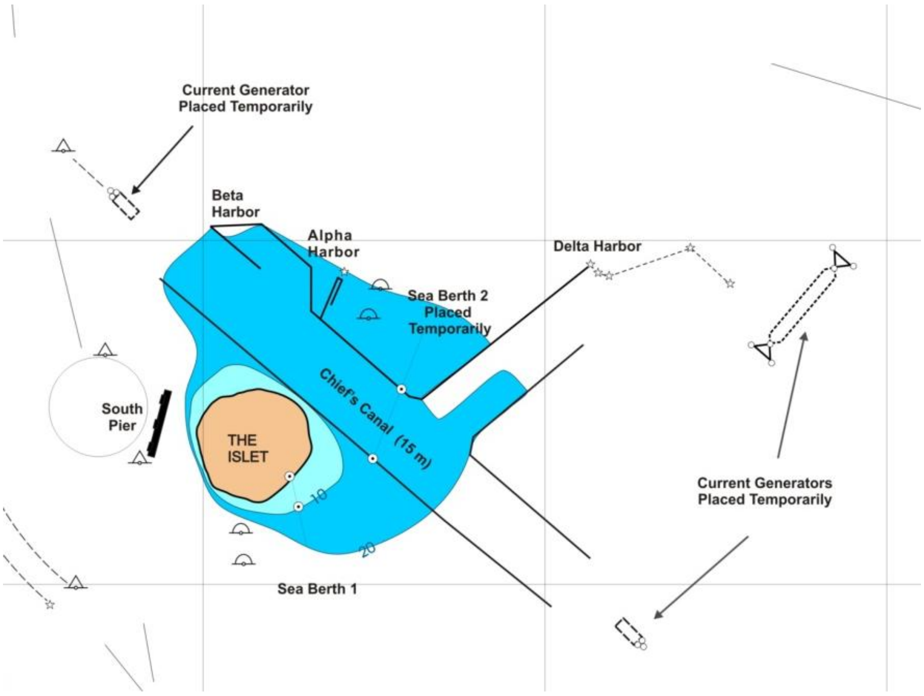



**Fig.59.Plan of the shallow water canal (Chief's Canal)**

Training areas arranged in a similar way are provided in Port Revel training centre as shown in the Report on Task 2.1.

The above arrangement of training areas provide ample opportunities to train ship masters and pilots to handle ships in difficult navigation situations, in particular in those, that may be present in inland waterways, harbour approaches etc, where strong interaction effects between ships and environment are present. Some of the above areas are shown in figures 52 and 53 as referred to above.

The safe operation of ships in restricted areas depends on the understanding by the operator the hydrodynamic interaction effects between the ship, shoreline, other ships and objects, taking into account current and wind forces and also peculiar manoeuvring characteristics of the own ship. Therefore the main purpose of training on manned models is to understand influence of manoeuvring characteristics of ships on safe operation and to recognize and understand the different hydrodynamic interaction forces and factors that may be present, in particular in close proximity situations. Ship masters and pilots when performing different manoeuvres are supposed to realize of how these forces act, how large they might be and how to counter or use them in order to achieve safe passage.

When using manned models there is no need to develop mathematical algorithms representing effect of environmental factors or hydrodynamic force components representing interaction effect. Those are simulated automatically because they depend mainly on pressure distribution around the ship body that, in turn, is simulated properly if Froude's law of similitude is used, which is the case in model work. The proper simulation depends therefore on the exact scaling down external limitations, i.e. harbour basins, canals, shallow water, banks etc.

These facilities in the training areas are arranged according to linear scale the same as chosen for the models used, including canal profile, depth of the water, piers and jetties, either on piles or consisting of solid wall.. They represent typical situations that could be met in different parts of the world in different ports and harbours, river estuaries, passages, docks etc.

There is, however, little possibility to arrange for individual trainees according to their wishes in the training areas mock-up of real port facilities or mock-up of specific canals and waterways built to simulate real canals, port approaches and similar situations. They may require considerable hydrotechnical and dredging work bearing in mind large scale models used and it would be impractical to prepare such costly facilities for limited number of trainees.

According to experience of Ilawa training centre such facilities are sometimes arranged if the number of interested trainees is greater, but to make artificial canals dredged to specified depth and profile appears to be a very costly enterprise. In particular to simulate muddy bottom and arrange layer of mud on the bottom of the fairway seems to be totally impractical.

On the other hand to arrange exercises such as ship-to-ship, ship-to-FPSO, towing, anchoring, meeting or overtaking other ships, escorting in the direct or indirect mode is rather easy and such exercised are arranged quite often.

Below are shown few examples of manoeuvres where interaction effects are simulated in the model work. Those are typical situations that may be met in different places around the world.

<u>Example No.1</u>. Feeling bank effect and suction force. The ship is sailing in the narrow fairway parallel to bank of the canal or river. (Fig. 60). Suction force causes that the body of the model is sucked towards the bank and the bow of the model due to compression of the bow cushion is rejected from the bank having the tendency to swing. The trainee is instructed to counter this effect using rudder towards the bank.

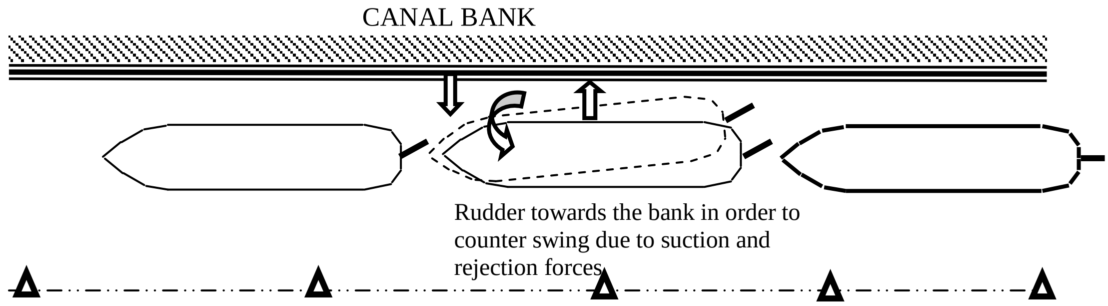



**Fig.60 Example exercise – negotiating Bank Effect Route**

Example No.2. Negotiating the left hand bend of the canal. Negotiating the left bend of the canal operator may use suction and rejection forces to help steering if it is sailing not in the centreline of the canal, but closer to the right bank. Trainees are instructed to use this effect in order to help turning. They are also instructed to do this in oppsite way, sailing closer to the left bank recognizing that it would be much harder to turn the ship. (Fig.61)

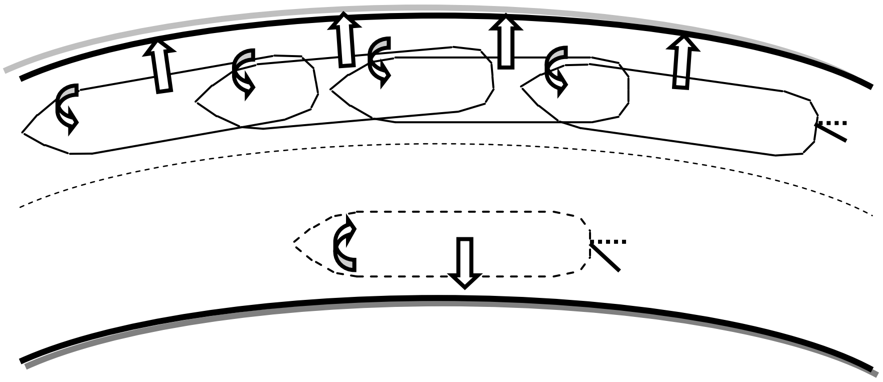


**Fig. 61. Example exercise – negotiating left hand bend in the narrow canal (Pilot's Canal)**

Example No.3. One ship overtakes the other in a narrow canal.One ship overtakes the slower one in the narrow canal (Pilot's Canal). Trainees are instructed to observe interaction effects, in particular they must counter effects of suction and rejection forces in order to achieve safe passage. The proper overtaking procedure is described in the briefing session (Fig.62)..

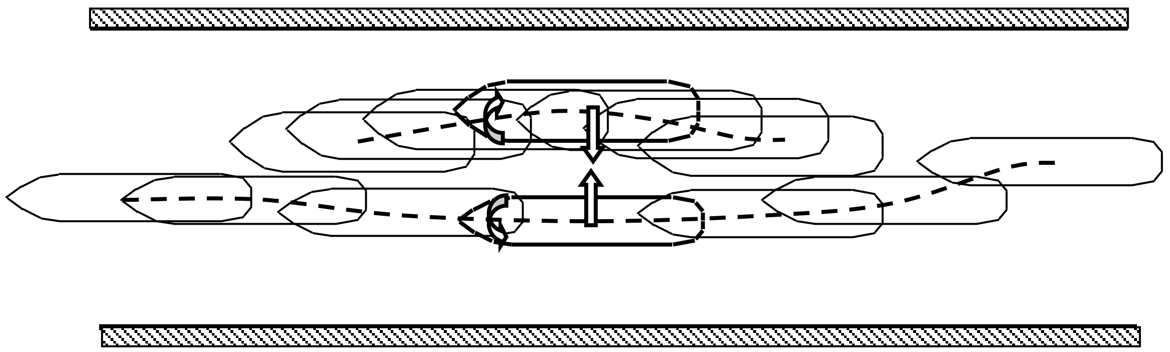


**Fig. 62. Example exercise – overtaking slower ship in the narrow canal (Pilot's Canal)**

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**APPENDIX 1.-**

**An Introduction to Reverse Tractor Z-Drive Towing Modes For Pilots**
**By Capt. Doug Pine**

When I'm handling a vessel, my greatest joy lies in the art rather than the science of how I cause that vessel to respond. I'm right-brained. The science puts me to sleep. However, as a prudent mariner, I learned the hard facts, and can acquire endless sources of information about the technical and scientific principles of escort/assist tug use. But, I don't stop with being able to automatically recall principles. Information meets art when it's filtered through the right side of the brain. Pure principles and the highly technical aspects of Direct and Indirect towing modes are transformed – from complexity to instant comprehension – without losing a drop of precision. It's a bit like seeing the finest political cartoonists convey a masterful message

In this article assume a good working relationship with the tug operator, which includes a high level of trust. A tug operator new to the job or unknown to the Pilot will initially require a higher level of management via more complex commands than one who is known to and trusted by the Pilot. Bottom line: The Pilot can achieve what he desires by issuing an order, in the simplest possible terms, and allowing the tug Master to work appropriately, given the dynamic environmental factors present as the job progresses. For the Pilot, this translates to less time spent on the radio managing his assist tug(s). For the ASD tug Master, this means using, at his discretion, the known capabilities of his tug to achieve the desired results.

**Important:** Given the 1,000,000 lb. breaking strength of today's modern towlines, the weak link in the operation is now found at the chocks and bitts on he ship being worked. Obtaining from the ship's Master the maximum rated tonnage force of the center chock towing bitt is a critical part of the Master/Pilot exchange.

**Pass this information to the tug Master.** If forgotten, he'll ask! He'll use the tug's tonnage gauges and (if so equipped) modern render/recover winch technology to dial in a tonnage limit below what will pull the bitt off the ship

**Reverse Tractor Z-Drive Towing Modes:**

In the normal course of operations, as the ship's water speed varies, the tug will operate through four different modes:

1. Transverse Arrest
2. Indirect
3. Powered Indirect
4. Direct pull (inline or out to either side)

**Commands:**

Here's the beauty of simplicity. Ask yourself: "What do I want my stern to do now?".

1. Slow me down
2. Stern to starboard
3. Stern to port
4. Check my swing
5. All stop or Run along (slack line/tight line)

Let's have a look each mode and the related artful commands.

**Transverse Arrest:** (Speed through the water >4 knots)

Tug thrust vectors are outboard at right angles to the tug's heading. Braking forces at speeds >8 knots can be greater than rated bollard pull. These forces drop rapidly as speed through the water **decreases**. Below 4 knots the tug operator will transition to Direct Pull Inline, with forces up to rated bollard pull.

Tip: Transverse arrest is useful when you want to keep turns on the ship for better steerage. For example, you can keep SAH turns on while making DSAH speeds through the water, taking advantage of greater water flow over the rudder. Especially handy in a following current.

**Commands**:

"Transverse arrest (easy, half, full)" **or** "Back (easy, half, full)" **or** "Slow me down (easy, half, full)" (Some Pilots issue orders using specific tonnage in lieu of engine orders e.g. "Transverse arrest x tons")

Some Pilots ask the tug to maintain a certain speed: "Hold me at x knots"

Once in Transverse Arrest, you adjust forces simply by giving the appropriate order (easy, half, full or x tons).

**Indirect Mode:** Ship speed through the water 4 to 10 knots. Generally speaking, 6 - 8 knots seems to be the "sweet spot" for use of the Indirect Mode. The "classic" Indirect Mode occurs when the tug positions the towline at 45° to the ship's transom. The angle actually varies depending on the ship's water speed and the capabilities of the tug. Indirect forces can be generated at any angle, but smaller angles induce more braking than turning forces along the towline vector. The tug is positioned at an angle to the towline that allows its hull and skeg to induce the hydrodynamic forces that generate up to or greater than rated bollard pull along the towline vector. Indirect Mode is used to steer the ship.

Adjust power settings to achieve desired rate of turn (or lack thereof). Water speed is critical if you wish to maintain high Indirect forces. The forces drop rapidly as water speed falls below 8 knots, to the point where Direct Pull Mode becomes the best choice. Properly used, Indirect forces can quickly overcome a hard over rudder failure on the ship, and give the Pilot the very nice option of continuing along the intended trackline. Remember, however, that once the rate of turn of the ship progresses beyond a certain point, no tug will be able to stop the swing. Prompt reaction to the steering casualty and quickly ordering the tug out is imperative.

**Command**:

"Indirect 45 ° to stbd/port (easy, half, full or x tons)" **or**

"Indirect to stbd/port (easy, half, full or x tons)" **or**

"Stern to stbd/port (easy, half, full or x tons)"

Can also be used to stop the rate of turn of the ship.

**Command: "**Check my swing"

**Steering failure command:** "Rudder failure. Take my stern to stbd/port" The tug Master recognizes the emergency, and goes out to Indirect Mode applying full power. As the swing is checked, the Pilot adjusts power orders or tonnage forces accordingly to fine tune the ship's rate of turn, stop it altogether, or to create a variable rate of turn in either direction.

**Powered Indirect Mode:** Ship speed through the water: +/- 4 to 8 knots (top speeds depend on the horsepower and hull form of the tug being used) The tug powers into the towline so that the towline is at right angles to the ship's transom. The tug is positioned at an angle to the towline, generating forces that can be up to or greater than rated bollard pull along the towline vector. Again, water speed is critical if you wish to maintain Powered Indirect forces. The forces drop rapidly at speeds below 8 knots. Powered Indirect Mode can be specifically ordered by the Pilot at slower speeds, at which he knows the tug can get out to 90 ° or, assuming the tug is already out in Indirect Mode, as the ship slows the tug will transition into Powered Indirect Mode and continue there as speeds slow to approximately 4 knots. At this point the tug will begin to overtake the ship,

and the tug Master must transition (jackknife) into the Direct Pull Mode. At zero knots through the water, direct pull generates rated bollard pull. With any way on, a percentage of the horsepower of the tug will be used to hold position and
bollard pull will decrease accordingly.

**Command**:
"Powered Indirect 90 ° to stbd/port (easy, half, full or x tons) **or**
"Powered Indirect to stbd/port (easy, half, full or x tons) **or**
"Stern to stbd/port (easy, half, full or x tons)

**Direct Modes:**
Ship speed through the water: <3 to 6 knots

**1. Direct pull inline:** Generates rated bollard pull at any speed, however the tug Master will run the risk of stalling one or both engines if ordered to use Direct Pull Inline at speeds over 4 - 6 knots

**Command:** "Back" **or**
"Direct Pull inline (easy, half, full, or x tons)". While in Direct Pull Mode, If the ship's water speed increases to above 4 knots, the tug Master will transition to Transverse Arrest Mode.

**2. Direct Pull to either side:** At zero knots through the water, direct pull generates rated bollard pull. With any way on, a percentage of the horsepower of the tug must be used to hold position and tonnage forces along the towline vector will decrease accordingly.

**Command:** "Stern to port/stbd" **or**
"Direct Pull to port/ stbd at **x** degrees (easy, half, full, or x tons)" If, while in Direct Pull Mode the ship's water speed increases to above 4 knots, the tug Master will transition to either Powered Indirect or Indirect Mode.

**Summary:**
Center lead aft escort/assist tug work is a brand new concept to many Pilots and tug operators, some who may have operated Z-drive tugs for years but have never been tied up to that center bitt on the stern. I've found that the best way to introduce Pilots and Tug Masters to these concepts is in simulation, training side by side. Giving them simple commands to use to convey advanced concepts is a key part of this training. Training together with the tug operators allows everyone involved to benefit from of the interactive nature of linked two-bridge simulation. A real bonus to this type of training is that the Pilots can walk from the ship to the tug bridge and observe from the tug's perspective, watching the tug Master put direct and indirect forces to use.

**Example: Slow the ship and make a 180° clockwise turn.**
All of the towing modes discussed above will be used. There is no wind or current. We'll assume that the Pilot is using the ship's engine and rudder in conjunction with the assist tug, and there is sufficient sea room to safely execute the maneuvers. On this ship, DSAH is 7.2 knots, SAH is 10 knots. Here's how it flows:

1. The tug is ordered to tie up center lead aft, with ship's speed through the water at 10 knots.
2. The Pilot decides to slow the ship to 8 knots, while keeping his engine at DSAH. The order is given to apply braking force to the ship:a. **"Back half"**
3. The ship slows and is held steady at 8 knots through the water, then enters the area where the 180 ° turn will be made.

4. The Pilot wishes to develop a rate of turn to starboard, while continuing to slow the ship. He orders all stop on the ship.a **"Stern to port half"**

5. The tug Master, given the water speed of 8 knots, works out to the **Indirect Mode** to port, and applies the appropriate power setting.

6. The Pilot issues power orders to the tug to fine-tune the desired rate of turn **(easy, half, full)**

7. As the ship's speed through the water falls, the tug will work out to the **Powered Indirect Mode**, and then jackknife to the **Direct Pull Mode** at 90° to port below 4-5 knots water speed.

8. At zero knots water speed, the tug, still in **Direct Pull Mode**, will be (if at full power) exerting rated bollard pull on the towline. In this state, the ship can develop a rate of turn above 30° per minute when using the tug and the ship's bow thruster.

9**.** At the appropriate time, the Pilot orders **"Stop"**, and then uses the tug tohelp check the swing of the ship.

**<u>Captain Doug Pine- Simulation Manager</u>**

Captain Pine is responsible for all simulation training and research activities at PMI. He also serves as a Project Manager and Instructor in research and training courses involving simulation and electronic navigation topics.

**Synopsis:**

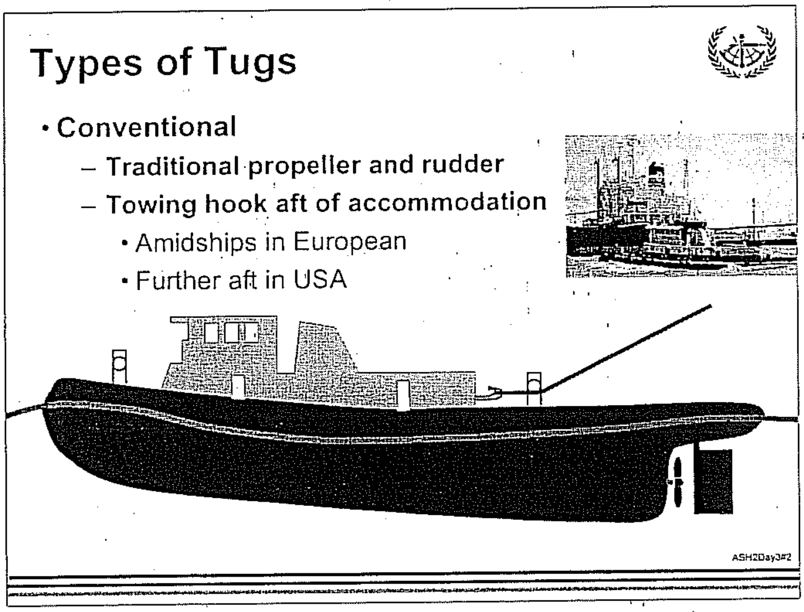
Captain Pine joined PMI in 2006 during his 25th year in the maritime industry. He has spent his career involved in an eclectic mix of industry segments, including passenger vessels both power and sail, linehaul and harbor tugs, Offshore Supply Vessels, ferries, and even a passenger submarine. He holds a Master of Oceans Power and Sail license, a Master of Towing license, and an increasingly rare AB Sail endorsement, and is a USCG approved Designated Examiner for Towing Licenses. He is primarily involved in the simulation and IT/computing areas of PMI's operation. From developing simulation exercises and urriculum, collaborating on research proposals and projects, and taking the lead on PMI's simulation and electronic navigation systems work with a growing number of United States and Canadian Pilot groups, Captain Pine is playing an key role in the success and growth of PMI.

**Maritime Experience**

- Washington State Ferries – Seattle, WA. Five years plying the waters of Puget Sound as a member of the deck department of the largest ferry system in North America.
- Hornbeck/Tidewater Marine – Galveston, TX. Operating Offshore Supply Vessels in he Oil and Gas industry in the Gulf of Mexico, during the hands-on pre-DP time frame. Heavy weather and close quarters boathandling in a dangerous and challenging environment was the daily routine.
- Sause Brothers Towing – Honolulu, HI. Towing container, petroleum, propane, cement, and flat deck barges in the interisland trade, and ship assist work in Honolulu and Kahului Harbors.
- Atlantis Submarines –Maui, HI. Operating passenger submersibles as a co-pilot to depths of 150 feet, and Master of the ocean going twin hull tug/submarine tender vessel "Ocean Twin".
- Passenger vessels – Maui, HI. Master and owner of a variety of vessels from the 190' dinner cruise vessel "Stardancer" to the 1920s era Alden schooner "Teragram", and everything in between.

## APPENDIX 2.

### Tug Operating Near the Stern of Pod-Driven Ships




**Conventional Tug**

Although a "Conventional Tug" may have twin controllable pitch propellers in shrouds to enhance power and twin Becker rudders to improve maneuverability, the defining elements remain.

•The propulsion system is at the stern.

•The towing point is forward of the propulsion.

•Thrust is only controllable while the propellers are turning ahead (twin screw will offer some astern controllability).

The use and therefore the design of conventional tugs in the USA differs from that in Europe.

In Europe, conventional tugs are generally used in the towing position, with a tow line led to the ship from the stern of the tug. To maintain maneuverability while towing, the tow hook and bollard must be positioned near amidships.

In the USA, conventional tugs are secured alongside the ship with two or more lines and can push or pull as required. The aft towing hook is positioned further aft than their European counterparts, because the tug's maneuverability is already restricted by the mooring lines. Having the tow hook further aft allows for a larger accommodation area.

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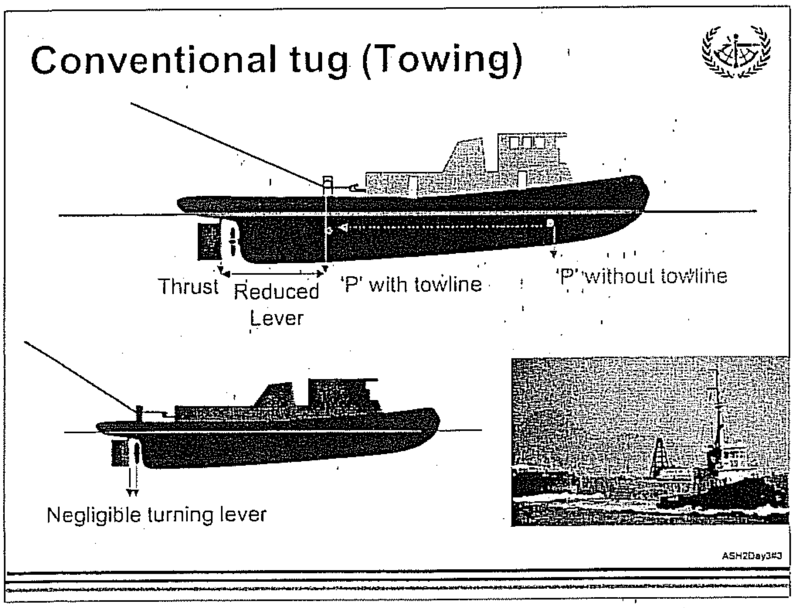


**Towing**

When a conventional tug is making way without a tow, its pivot point will be approximately a quarter of the boats length from the bow. This will provide a good turning lever in relation to the position of thrust. However, when the tug takes up the tow, the pivot point will shift to the position of the towing point. The turning lever is thus reduced and the tug's maneuverability is restricted. Tugs designed for towing, will have the towing point as near amidships as possible to maintain acceptable maneuverability.

As the towing point is moved aft, the turning lever reduces further. With the towing point over the rudder, there is no turning lever and the tug cannot be steered without slacking down on the towline. Sometimes the towing point is deliberately shifted aft with a "Gob rope". This keeps the stern of the tug inline with the towline to reduce the risk of "tripping" (to be discussed later).

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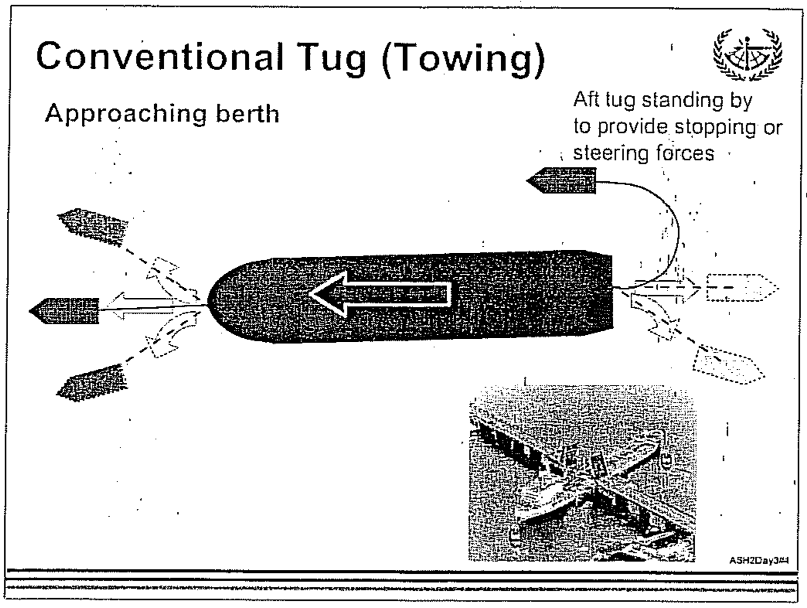


**Conventional Tug (European approach)**

The tugs are positioned fore and aft with a single towline from their stern. As the ship approaches the berth the forward tug controls the bow, while the after tug stands by to apply braking or steering forces as shown above. Conventional tugs can work either end of the ship provided that the ship's speed is kept low.

This method of tug use is ideal for narrow waters, especially when passing through narrow bridges and negotiating locks or entering dry-docks.

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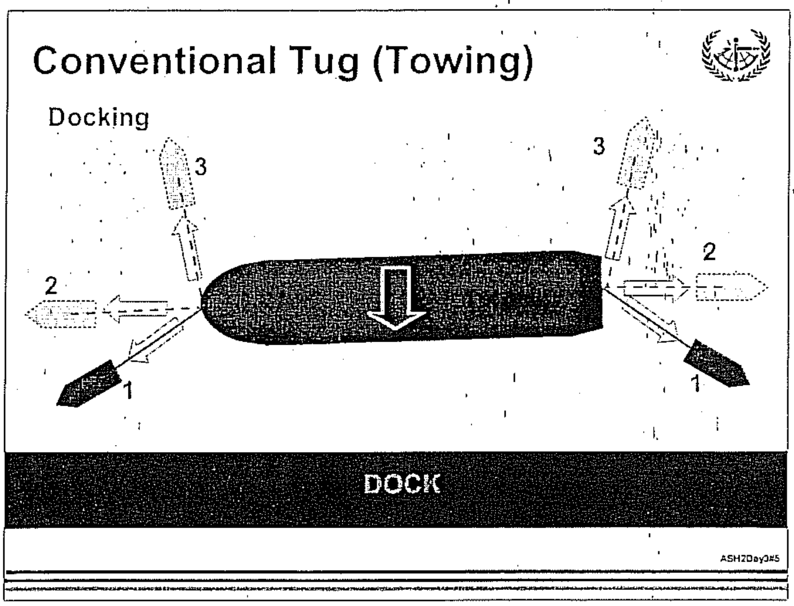


<u>Conventional Tug (European approach)</u>

The tugs are used to dock the ship as follows:

- Tugs pull the ship towards the dock, adjusting angle and thrust to ensure a parallel approach.
- Tugs pull fore and aft to position the ship.
- Tugs pull off the dock to reduce lateral motion.

Because the tugs are not positioned to push and hold the ship alongside, mooring boats are often used to take the lines ashore.

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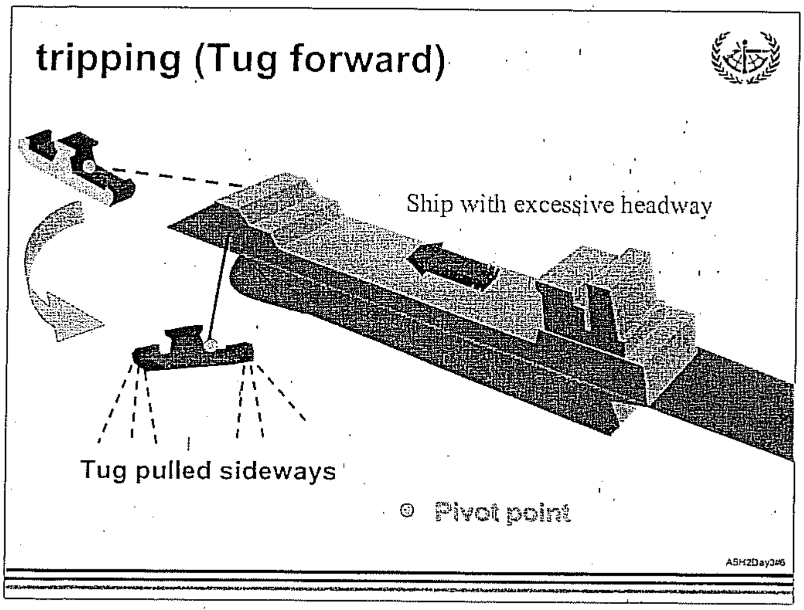


<u>tripping (Girding)</u>

tripping, is when a tug is pulled sideways by the ship. Conventional tugs, designed for towing with their towing hooks near amidships are vulnerable to "tripping" because the pivot point is near amidships. If the pivot point was forward or aft, the hull resistance would swing the tug around until it was being pulled ahead or astern.

**Tug Forward**

A tug working a line from the ship's bow is relatively safe provided it remains within a small arc of operation. However, if excessive ship speed or an alteration of course brings the tug broad onto the bow, it may not be able to keep up with the change in relative positions. If the tug cannot get its stern around and pull ahead of the ship, it may be dragged and pulled over sideways. Several tugs have been capsized before the towline broke or could be released. The rapid nature of this incident often results in loss of life.

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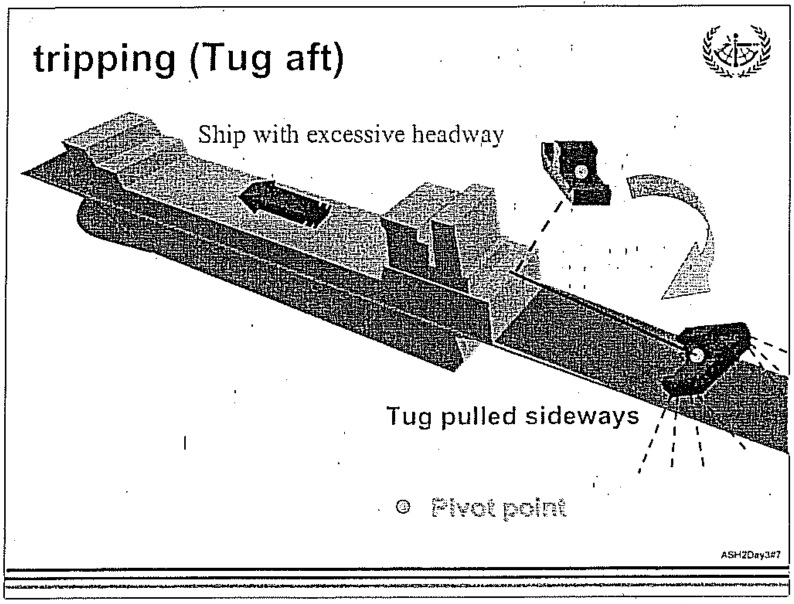


**tripping (Tug aft)**

Conventional tugs working aft are more susceptible to tripping than tugs working forward. While the ship is moving forward, the hydrodynamic forces on the after part of the hull in relation to the position of the pivot/towing point tend to force the tug sideways. A conventional towing tug should not be used aft unless the ship is at low speed or stopped.

Again, should the ship accelerate or turn to quickly, the tug will be caught with the towline leading on the beam and in danger of tripping the tug.

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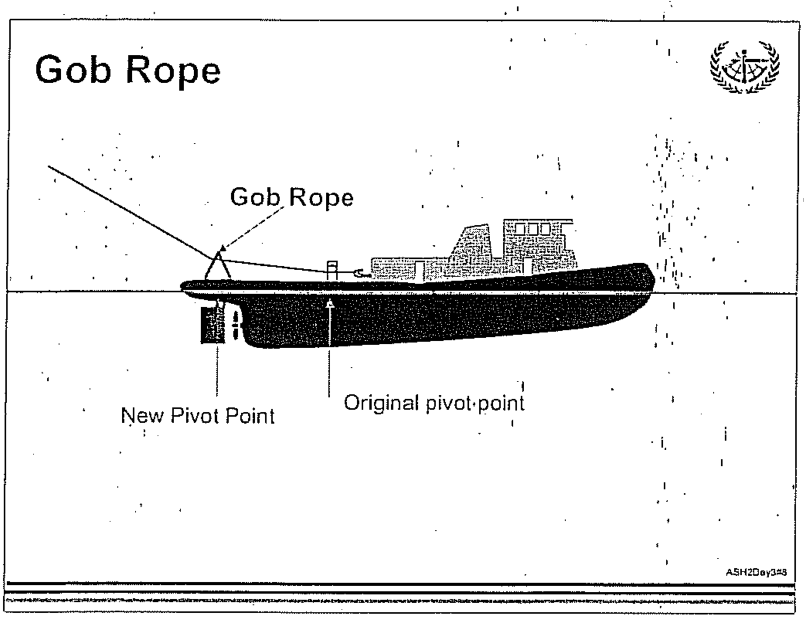


**Gob Rope**

A gob rope can be used to reduce the risk of tripping. A line is used to bowse down the towline aft of the towing hook. This has the affect of moving the pivot point to the location of the Gob Rope. The Gob Rope prevents the tug from being pulled sideways, but also restricts maneuverability because of the relative positions of the pivot point and thrust. The rope is adjustable and must be released when the tug needs to maneuver. Once in position the tug can reapply the Gob Rope and pull in the desired direction.

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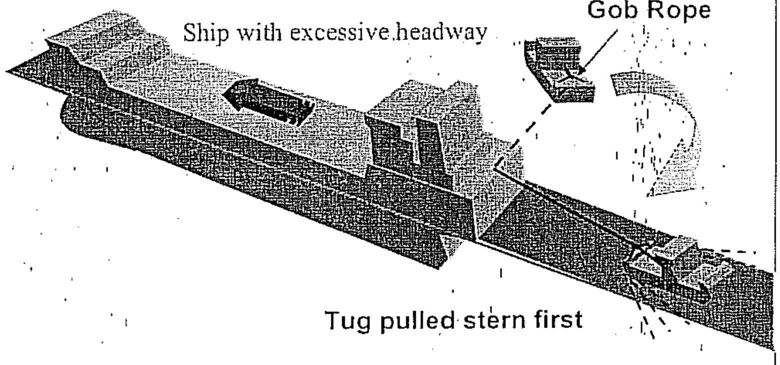
## Gob Rope (Tug aft)




**Gob Rope (Tug aft)**

Should the tug find itself at risk of tripping, the Gob Rope keeps the pivot point near the stern. As the towline takes control of the tug, the tug's stern comes in line with the direction of pull. Eventually the tug will be pulled stern first by the ship. This will give the tug master enough time to release the towing line if necessary. Most tugs have a quick release hook that can be activated from the pilot house.

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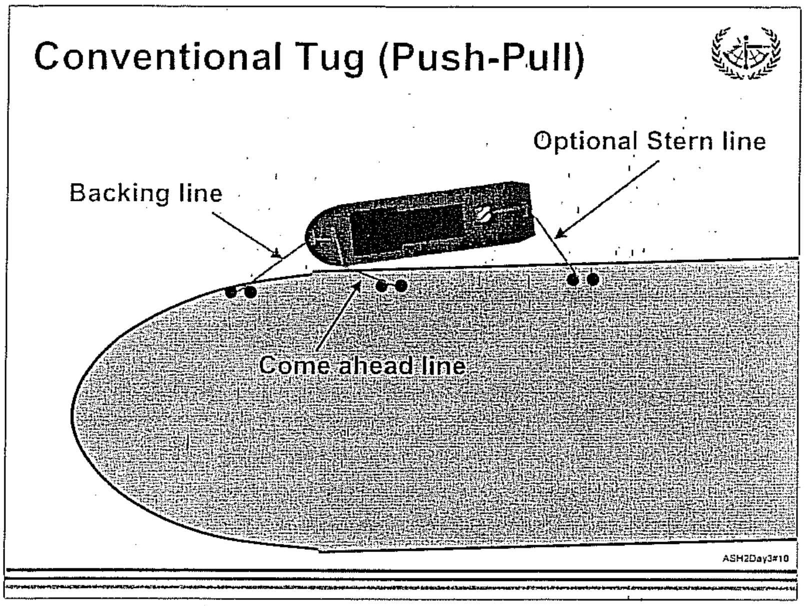


**Conventional Tug (U.S. approach)**

In U.S. ports the tug will normally be secure by two lines from the tugs bow. The first line will be led forward as the "backing line" and the second will be led aft as the "come ahead line".

The tug goes astern on the "backing line to slow the ship down and can pivot on the "come ahead line" to position itself to push.

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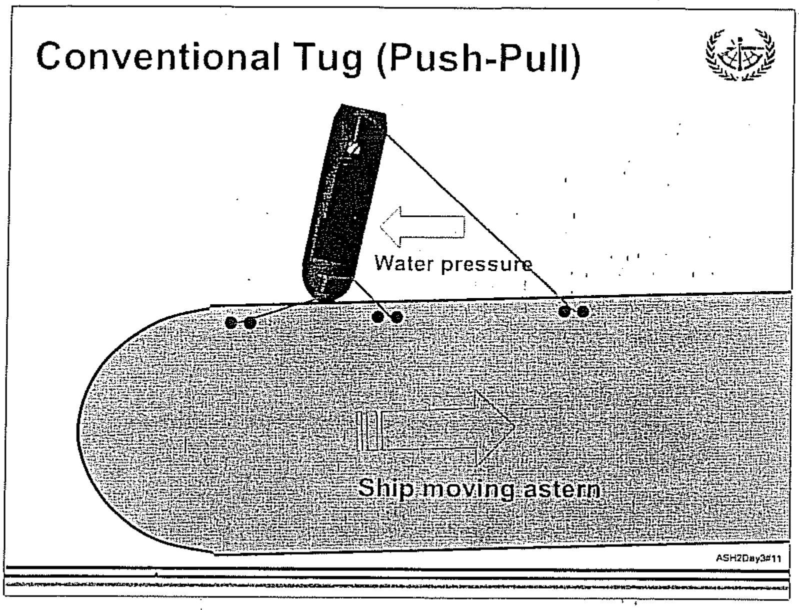


**Optional stern line**

The optional stern line is used when the ship must be backed to or from a slip. The stern line prevents the tug from being forced around by water pressure. A great deal of pressure will build up on the side of the tug if the ship's sternway is excessive. An obvious alternative to using the stern line is to rig the tug so the "backing line" becomes the "come ahead line" and vice versa. Twin screw tugs may use a backing line only because they can maneuver into position to push without the aid of the come ahead line.

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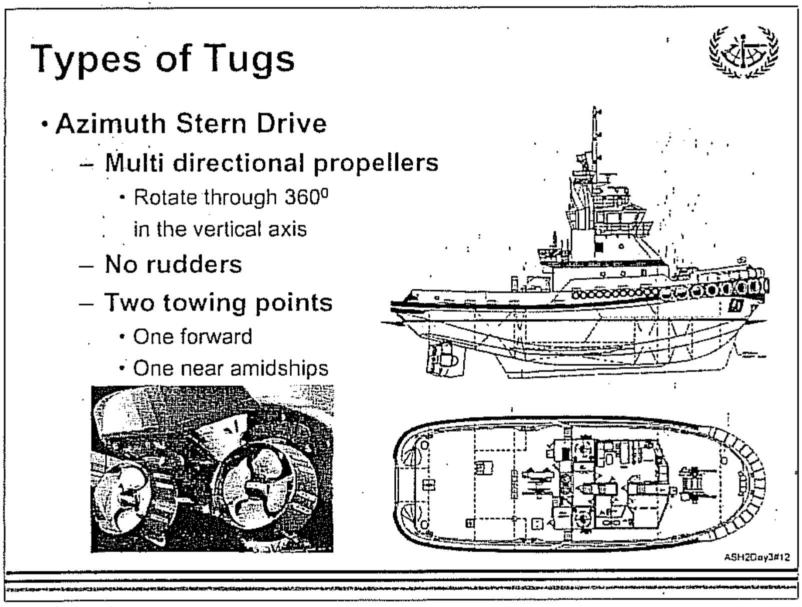


**Azimuth Stern Drive Tug**

This type of tug is propelled by two Azimuth Drives located aft. It has a towing position near amidships, like a conventional tug, and another on the bow. Directional thrust is provided by multi-directional propellers that can rotate through 360 degrees in the vertical axis. The direction of the propellers are coordinated automatically in response to commands relayed from a joystick control at the conning station.

An ASD tug can be used much the same as a conventional tug with the towline connected amidships, but it comes into its own when towing or pushing from the bow. In many ways the tug is now acting like a tractor tug (propulsion units forward) in reverse. The ASD tug retains the inherent directional stability of a conventional tug and benefits from tractor tug- like handling. This makes it very versatile and it is becoming common in ports around the world.

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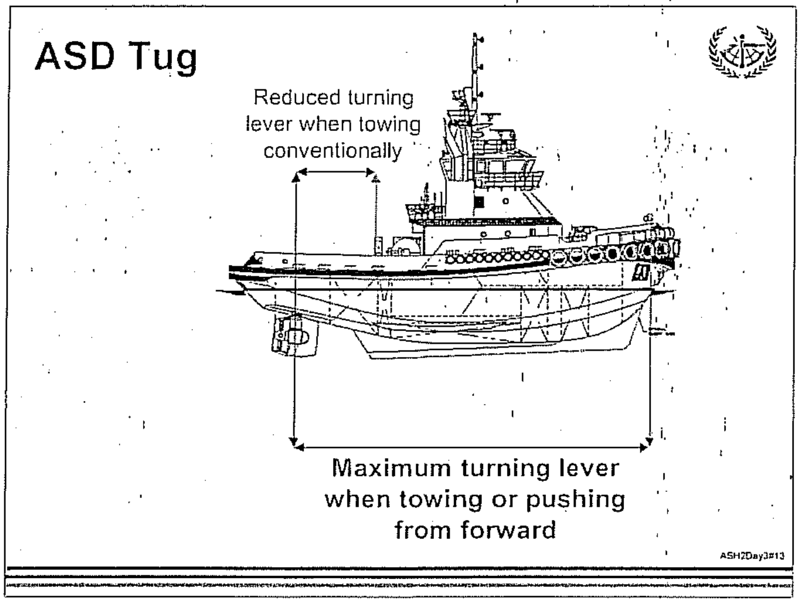


**ASD Tug**

The benefits of towing or pushing from the bow are illustrated in this slide. The distance between the forward towing point (Pivot Point) and the propulsion units is practically the full length of the tug. The long turning lever allows the tug to be re-positioned quickly and apply power at any angle of attack. The dangers of tripping remain while the tug is towing from amidships, but are eliminated when towing from the bow. Even without the risk of tripping, the power and maneuverability of these tugs can have them heeling over beyond safe limits.

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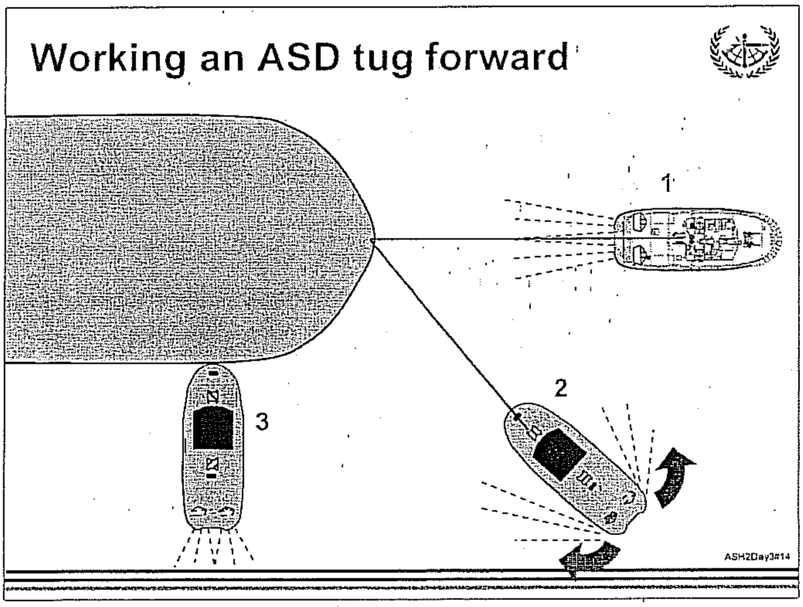


**Working an ASD Tug forward**

- The ASD tug is being used like a conventional tug with the towline from led from amidships. Even though the tug is highly maneuverable it remains at risk of tripping when used in this manner and its capabilities are somewhat limited.
- Here the tug is at its most versatile. With the towline led from the bow, the tug can side thrust around the radius of the towline without risk of tripping.
- Most ASD tugs can easily adjust the length of the towline and reposition against the ships side to push. A short headline can be used in this position so the tug can be used to push or pull, U.S. style.

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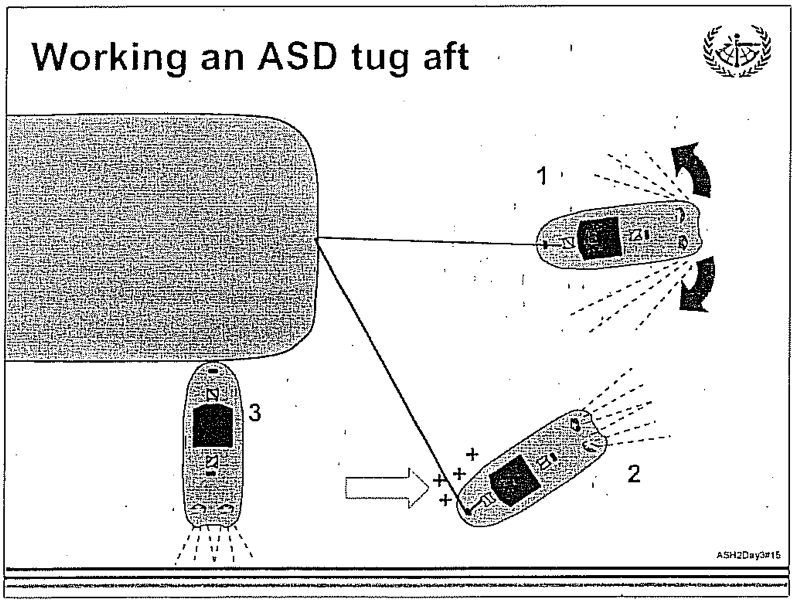


<u>Working an ASD Tug aft.</u>

- Here the towline is led from the bow. The tug can side thrust from this position to apply braking or steering forces. This is the preferred location for escort work.
- Here the tug is using the "Indirect" towing technique. The tug is positioned so that the water flow striking its hull exerts a force that is transferred to the towline.
- The tug can adjust the length of its towline and maneuver alongside to push. Again a short line from the bow will permit the tug to push or pull at short notice for maneuvering the ship alongside.

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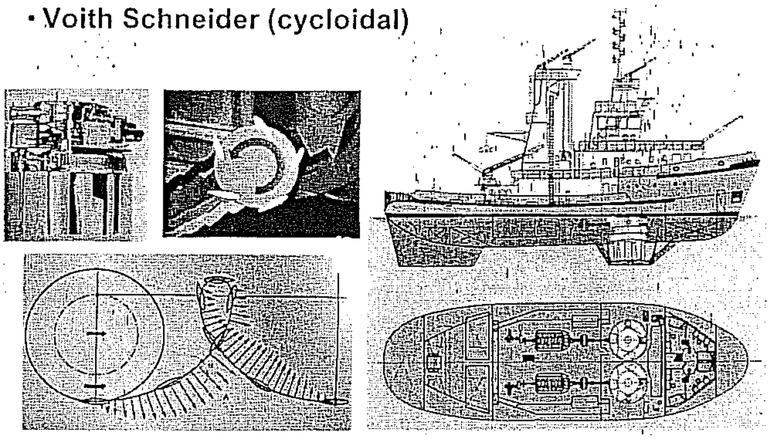
## Tractor Tugs

• Voith Schneider (cycloidal)



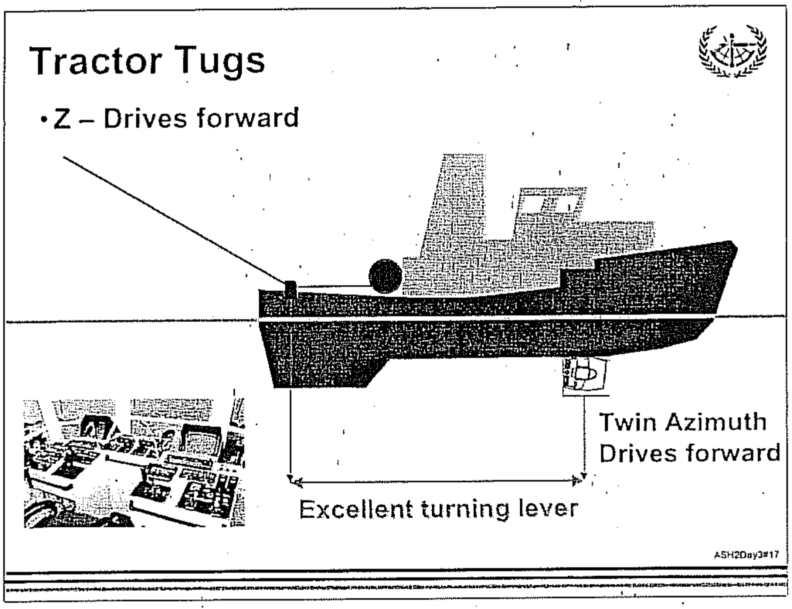
ASH2Day3#16

**Tractor Tugs**

The term "Tractor Tug" refers to tugs that have their propulsion units forward. They use two, multi-directional propulsion units and have a large skeg aft to provide directional stability. No rudders are needed as the propulsion is directional. The towing point is well aft to prevent tripping, yet excellent maneuverability is maintained because the propulsion units are well forward. The propulsion is provided in two ways; Voith Schneider (cycloidal) or Z-Drive.

**Voith Schneider propulsion system**

This comprises of a vertical axis propeller. A series of 5, rudder style blades, are mounted in a rotating disc. The angle of the blades are altered by a linkage system that is controlled by a joy stick at the conning station. The blades rotate at a constant speed, with the angle of the blades adjusted to provide the thrust in any direction. The large skeg is necessary to improve directional stability and towing performance (especially indirect towing).




**Z – Drive Tractor Tugs**

These tugs perform and operate on much the same way as the cycloidal type tractor tugs. The Z – Drive units are lighter than the VS units and therefore the tug will have a lighter displacement for a given power output. Like VS tugs, Z-Drive tractors tow and push from the stern.

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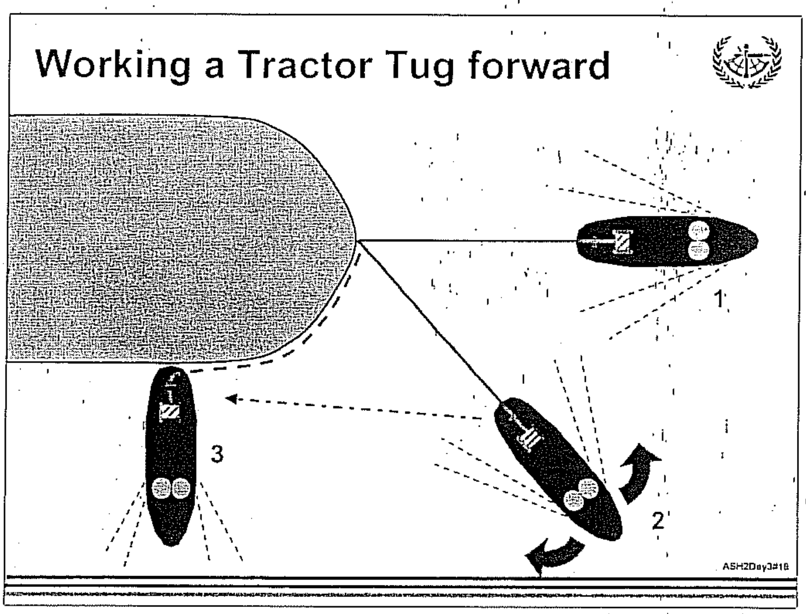


Working a Tractor Tug forward

1. & 2. The towline is led from the the stern of the tug. The tug operates ahead of the ship like a conventional tug, but can thrust sideways around the radius of the towline. Should the tug find itself at risk of tripping, the position of the aft towline will pull the tug around stern first.

3. The tug can also adjust the length of its towline and bring its stern alongside in position to push the ship laterally.

A tractor tug can be used alongside the ship in the same way as a conventional tug in the U.S.. However, maximum power cannot be achieved due to backwash and turbulence caused by the close proximity of the ship's hull. Also, care must be taken to avoid damage if the tug heels over into the ship's side when applying full side thrust.

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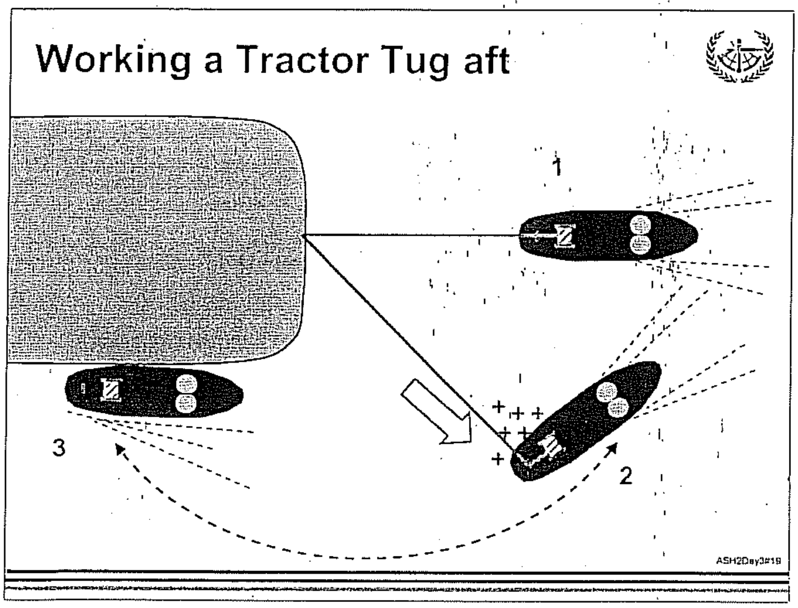


<u>Working a Tractor Tug aft</u>

- This has become the standard escort tug position. The tug follows the ship stern first and can quickly apply braking and turning forces to control a ship that has suffered a failure.
- Positioning the tug as illustrated above, takes advantage of hydrodynamic pressure on the hull (especially the skeg). The water flow striking the angled hull generates forces that can exceed that generated by the tug alone. This is known as indirect towing.
- This towing position is similar to the conventional U.S. method, but tractor tug maneuverability makes it even more effective. It is ideal for shifting a dead ship, but maximum bollard pull may be limited by turbulence between the tug and the ship's hull.

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## Tractor Tugs Compared to Conventional Tugs

- Advantages
  - Full thrust over $360^0$
  - Rapid power on response
  - Outstanding maneuverability
  - Low risk of tripping
- Disadvantages
  - High construction costs
  - Repair and maintenance expensive
  - Less bollard pull per HP
  - Open sea handling poor
  - Sophisticated underwater units prone to damage
  - Increased draft
  - Extensive training period

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## ASD Tugs
## Compared to Tractor Tugs

- Advantages
  - Better directional stability at speed
  - Better open sea performance
  - Improved bollard pull per HP
  - Less heel when thrusting sideways
  - Shallower average draft
- Disadvantages
  - Slightly less sideway maneuverability
  - Danger of swamping the aft deck when backing with full power
  - tripping possible when using after towing position
  - Complex control system

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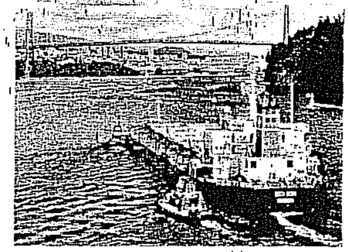
## Using Tugs

- Pivot Point
  - Changes position
  - Influences effectiveness of applied force.
  - Influence where and what tugs to use when.



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**Using Tugs**

Having discussed how to get the maximum power and flexibility from different types of tugs, let's now look at how to apply those forces to control the ship.

As you might expect, we have to return to the basics and revisit the Pivot Point.

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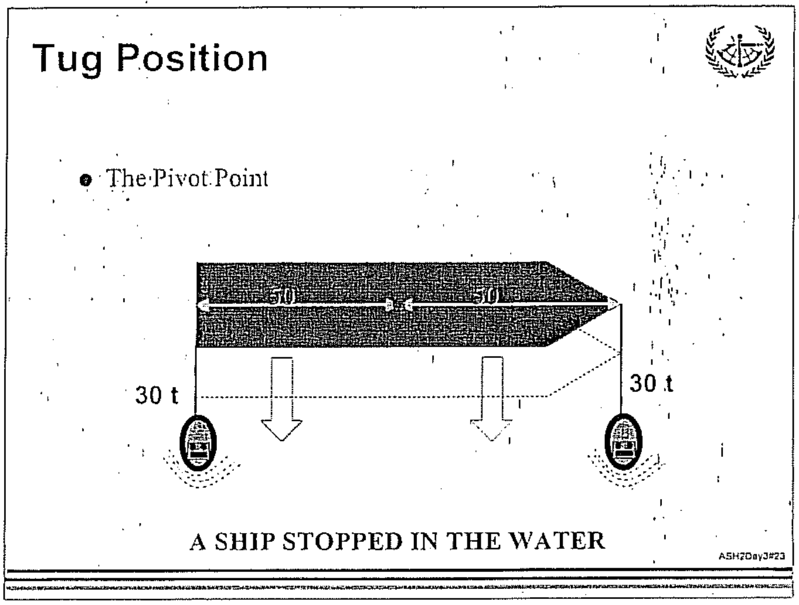
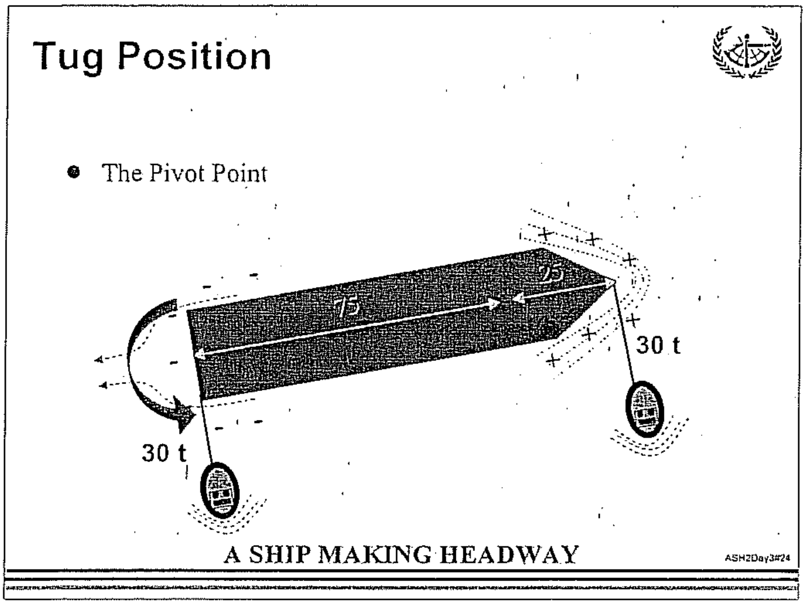



**Applied forces when stopped**

Equal forces applied to the ends of a ship stopped in the water, result in even lateral motion.




**Applied forces when making headway**

Turning forces are equal to the length of the turning lever multiplied by the force applied at right angles to that lever.

Equal forces applied to a ship making headway result in an imbalance of turning forces.

The tug at the stern finds it easier to turn the ship than the tug at the bow.

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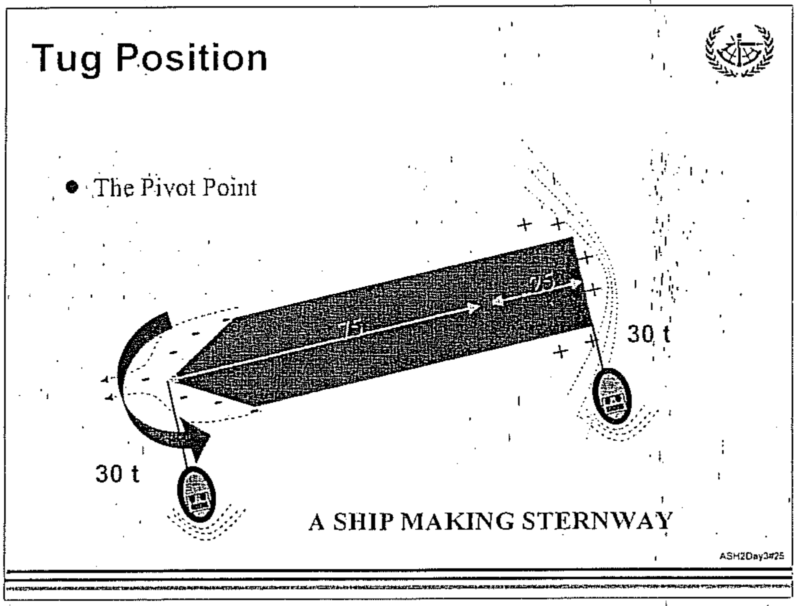


**Applied forces when making sternway**

It can be seen in the illustration above, that equal forces applied to each end of the ship, result in an imbalance due to the shift of the Pivot Point. In this case, the bow of a vessel making sternway is much easier to control than the stern.

This accounts for the improved effectiveness of a bow thruster when making sternway, compared to making headway.

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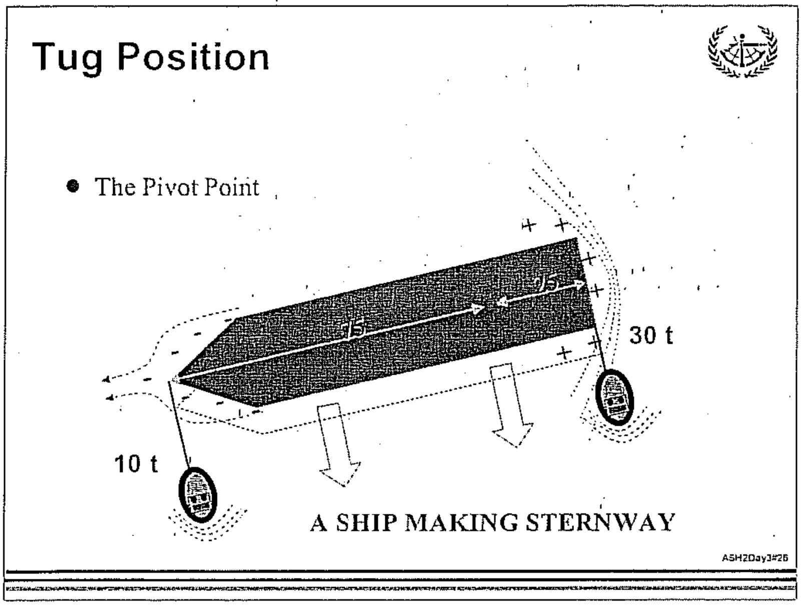


**Balancing tugs and pivot point**

Using a lower powered tug in a position furthest from the pivot point will balance its effectiveness with a more powerful tug closer to the pivot point. It is important to remember that when the ship stops or begins to make headway, a substantial imbalance will occur as the pivot point shifts.

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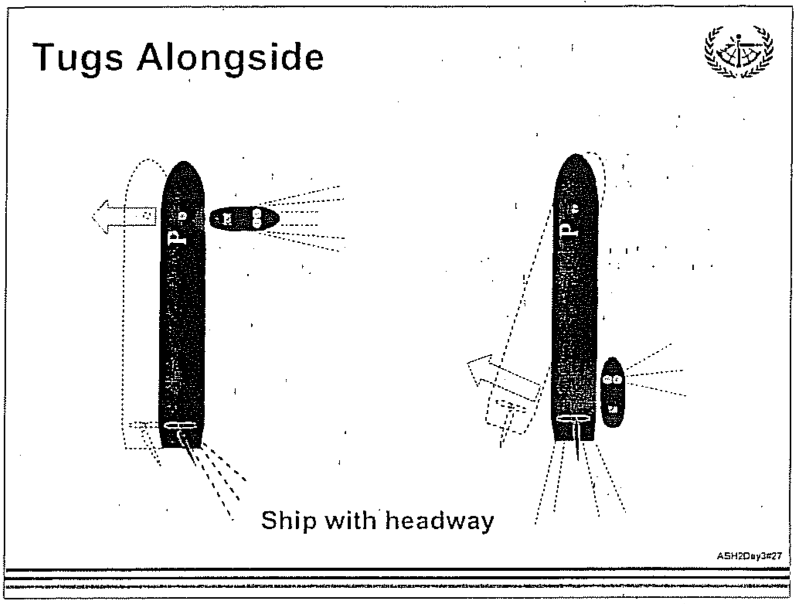


Tugs alongside with headway

When making headway, a tug at the bow can assist in generating lateral motion, but will be limited in its ability to swing the ship until the ship stops. On the other hand a tug located on the quarter or at the stern will be effective at swinging and steering the ship, but will be of minimal value for inducing lateral motion.

**Note: The tug becomes less effective at applying side thrust as the ship speed increases. This is because a portion of the tugs available power has to be used to maintain forward speed, thus limiting the thrust available for pushing or pulling sideways.**

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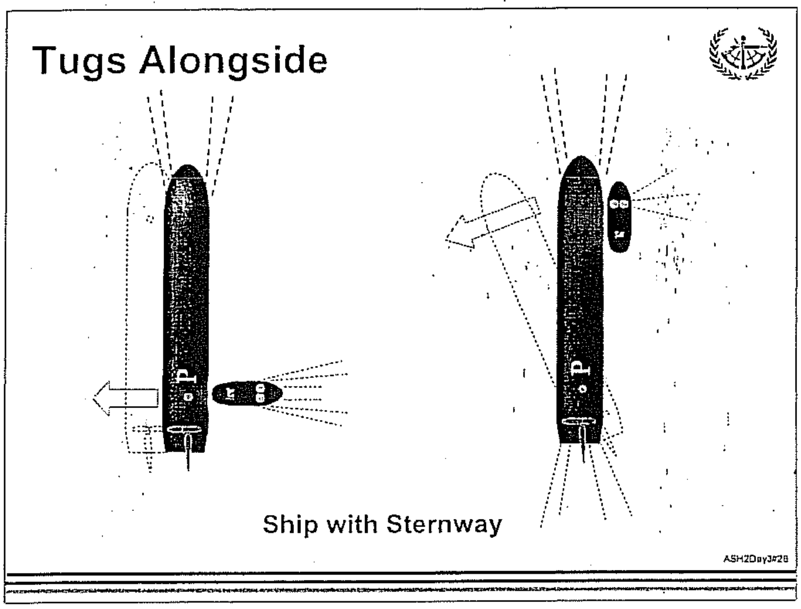


Tugs alongside with sternway

The opposite is true of tugs used when a ship has stern way, with the tug on the quarter providing lateral motion.

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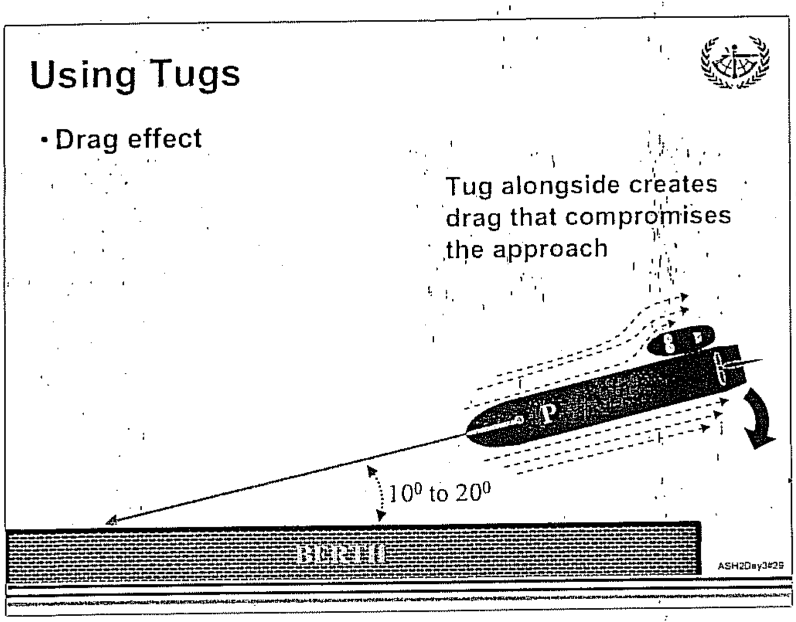


**Drag effect**

Having a tug secured alongside at the quarter will act like a rudder and cause the stern to swing away from the tug. The larger the angle of the tug to the ship, the more severe the effect. It is better to have the tug stand off the quarter until the ship is stopped and then have the tug come in and assist the ship alongside.

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