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IMPACT OF BANKS ON SHIP SQUAT

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SUMMARY

In a restricted channel the hydrodynamic behaviour of a sailing vessel is affected by both the vertical and horizontal boundaries. The restricted space underneath and alongside a vessel has a noticeable influence on both the running sinkage and trim of a vessel, also known as squat. A different bank geometry will obviously change the available space around the vessel. To assess these influences an extensive model test program has been carried out in the *Towing Tank for Manoeuvres in Shallow Water* (cooperation Flanders Hydraulics Research – Ghent University) in Antwerp, Belgium. The tests were performed with 11 different ship models (both seagoing as inland vessels) along 25 different bank geometries and cross section areas. Systematic model tests were carried out along vertical quay walls, constant sloped banks (from full depth to free surface) and banks with a submerged sloped part and a horizontal submerged area (semi submerged banks). Also rectangular cross sections with a range of widths and water depths were tested in the towing tank. During the model tests the models were free to heave and trim and the running sinkage was measured at four discrete positions of the ship model (fore-aft / starboard-port side). In this article the executed model tests are described and the impact of different bank geometries on the squat of the vessel is discussed. The change in squat for different slopes of the bank as well as the bank type (quay wall/surface piercing) and cross section areas is shown.

NOMENCLATURE

| | |
|-------------|--|
| A_M | Midship section area (m ²) |
| B | Breadth of ship (m) |
| C_B | Block coefficient (-) |
| C_M | Midship coefficient $A_M/(BT)$ (-) |
| Fr_h | Froude number based on water depth h (-) |
| h | Water depth (m) |
| L_{PP} | Length between perpendiculars (m) |
| L_{OA} | Length over all (m) |
| m | Blockage ratio (-) |
| Re | Reynolds number (-) |
| T | Draft (m) |
| V | Velocity (m/s) |
| V_{ship} | Ship's velocity (m/s) |
| δV | Return flow (m/s) |
| W | Width of the cross section (m) |
| W_h | Width of the section at full water depth (m) |
| x | Longitudinal position from FPP (m) |
| y | Lateral position from the centre line (m) |
| y_{infl} | Influence width (m) |
| y_{small} | Closest distance between ship and bank (m) |
| z_{VA} | Running sinkage at the aft perpendicular (mm) |
| z_{VF} | Running sinkage at the fore perpendicular (mm) |
| z_{VM} | Running sinkage at the midship (mm) |
| δ | Thickness of the boundary layer (m) |
| λ | Scale factor (-) |
| θ_V | Running trim (m/m) |
| ξ | Coefficient of the mathematical model (-) |
| ν | Kinematic viscosity (m ² /s) |
| Ω | Canal cross section area (m ²) |

| | |
|------|------------------------------|
| APP | Aft perpendicular |
| FPP | Fore perpendicular |
| FHR | Flanders Hydraulics Research |
| LNG | Liquefied Natural Gas |
| RoRo | Roll on/Roll off |
| UKC | Under Keel Clearance |

Subscripts:

| | |
|-----|----------------|
| m | at model scale |
| M | at midship |
| s | at full scale |

1 INTRODUCTION

The dynamic under keel clearance of a sailing ship is affected by the changed pressure distribution around the ship's hull. As a consequence the ship will move vertically downwards and will mostly trim as well due to the asymmetry between the fore and aft part of the ship. This phenomenon is commonly referred to as squat. The magnitude of this squat is affected by different parameters, among which the ship's speed, the propulsion of the ship and the available water depth. These effects have already been the subject of numerous literature e.g. [1].

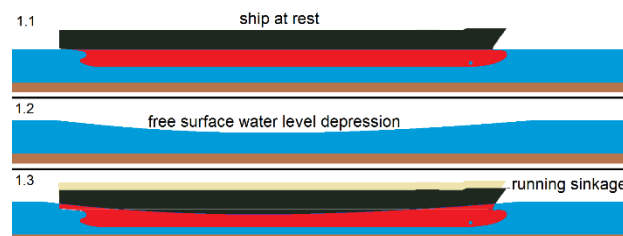


Figure 1. Ship and water at rest (1.1), return flow induced free surface water level depression (1.2), the sailing ship displacing the same volume of water as in 1.1 but having less under keel clearance (1.3).

In open and unrestricted conditions, the water displaced by a sailing vessel can travel almost without restriction underneath and along the ship's hull. In more shallow sailing conditions this water will rather be deviated around the hull, due to the limited space available between the ship's keel and the bottom of the waterway. This will result in higher velocities of the return flow travelling along the hull. This return flow will generate a pressure drop around the ship (Bernoulli's principle) and because of that, the free surface of the water will go down compared to the situation at rest. As such, the running sinkage or squat of a vessel should not be interpreted as an increase in the draft of the vessel but rather as a local decrease of the water depth around the ship.

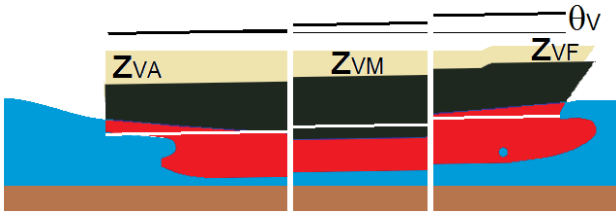


Figure 2. Two pairs of parameters to express the running sinkage: sinkages at the fore and at the aft perpendiculars (z_{VF} ; z_{VA}), or trim (θ_v) and mean sinkage z_{VM} .

In fact not only the water depth, but the entire cross section of the navigation channel affects the pressure distribution and the resulting squat. In general a more confined area will lead to larger squat for a given speed. In this article the attention will be put on the horizontal boundaries of a canal or fairway on the ship's squat. The effect of the bank geometry on the squat will be discussed based on an analysis of the database of captive model tests that have been carried out in the Towing Tank for Manoeuvres in Shallow Water at Flanders Hydraulics Research (FHR).

Banks with varying geometries are frequently built into this towing tank to investigate ship-bank interactions. The test results used for the present paper are acquired in the frame of different projects. As such a database is acquired with model tests with as much as 11 different ship models and about 25 different bank geometries. The database consists of more than 14 000 different test conditions (ship, draft, water depth, bank geometry, relative position, drift angle, speed, propeller action). A limited selection of these model tests has been made public as benchmark data in [2].

The running sinkage can be expressed as either the combination of the mean sinkage and the trim or as the sinkage at the fore and aft perpendiculars. The latter will be used in this article with a positive sign convention in case of a downwards motion.

2 MODEL TESTS

2.1 TOWING TANK

The model tests under consideration are a selection of the model tests performed over the last 10 years in the fully automated towing tank at FHR. A technical overview of this facility can be found in [3]; its main dimensions are listed in Table 1.

Table 1. Main dimensions of the towing tank at FHR

| | |
|---------------------|--------|
| Total length | 87.5 m |
| Useful length | 68.0 m |
| Width | 7.0 m |
| Maximum water depth | 0.50 m |

2.2 SHIP MODELS

Tests have been carried out with eleven different ship models of about 4m long (2 container carriers, 4 tankers, 3 RoRo-vessels, 1 inland vessel, 1 Wigley hull), some of them at different loading conditions. Detailed information on the ships' hulls is available in [4], the properties of the four ship models, used in this article, are shown in Table 2.

Table 2. Main properties of the ship models

| | | <i>KVLCC2</i> | <i>T0H</i> | <i>LNG</i> | <i>RoRo</i> |
|-----------------------|-----|---------------|------------|------------|-------------|
| λ | [] | 75 | 75 | 70 | 50 |
| Lpp | [m] | 4.267 | 2.210 | 3.809 | 3.800 |
| L_{OA} | [m] | 4.448 | 2.316 | 4.000 | 4.060 |
| B | [m] | 0.773 | 0.296 | 0.594 | 0.620 |
| T_M | [m] | 0.277 | 0.178 | 0.157 | 0.148 |
| C_B | [] | 0.81 | 0.85 | 0.77 | 0.62 |

2.3 BANK GEOMETRIES

In present article only a selection of tests in a steady state regime condition are considered to check the influence of the bank geometry on the running sinkage of the vessel. Therefore, the installed cross section did not change in geometry for a significant amount of ship lengths (at least six ship lengths) before the ship model decelerates or another bank geometry initiates. When two geometries are installed consecutively in the tank, the transition zone of one bank to another is constructed in such a way to create a smooth change in geometry, this is to avoid abrupt and long lasting transition effects.

In the past tests have been carried out with surface piercing banks (a constant slope from the bottom of the towing tank up to the highest water level tested, Figure 3) and semi submerged banks (a sloped under water part in

combination with a horizontal submerged part). In present examples, only surface piercing banks are considered. The slope of the surface piercing banks is expressed as the ratio between the rise and run (Figure 3) with a normalised rise (a run/rise ratio of zero is a vertical wall). In Table 3 the width at the bottom of the cross section (W_h) is added.

Table 3. Slopes and full width of different installed banks

| | | | | | | | |
|-----------|-------|-------|-------|-------|-------|-------|-------|
| run/rise | 0 | 0 | 0 | 0 | 0 | 0 | 0 |
| W_h (m) | 0.812 | 0.966 | 1.314 | 1.933 | 3.865 | 4.400 | 6.330 |
| run/rise | 0 | 1 | 3 | 3 | 4 | 5 | 8 |
| W_h (m) | 7.000 | 4.200 | 4.200 | 5.730 | 4.400 | 4.030 | 4.030 |

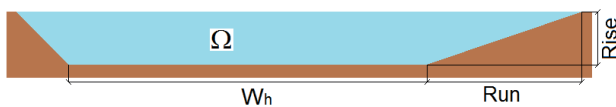


Figure 3. Definition of the dimensions of the cross section: run, rise, W_h and cross section area Ω .

2.4 WATER DEPTH

The tests are carried out at a range of shallow to very shallow water depths. For most ships this is a water depth to draft ratio of 2.00, 1.35 and 1.10 but for the ship model of a RoRo vessel as many as eleven different water depths were systematically tested.

2.5 POSITION, VELOCITY AND PROPELLER

The relative lateral distance between a ship model and a bank can be defined in different ways. The most straightforward method is by referring to the earth bound coordinate system of the towing tank itself. In most combinations, tests are carried out at about 5 different lateral positions to be able to understand the influence of the lateral position and bank effects.

Forward speeds are tested systematically from 8 to 12 knots (full scale). In very shallow water tests at a lower (less than 8 knots full scale) forward speed are added while in more deep water higher speeds (more than 12 knots full scale) were added to the test program. All the tests under consideration were captive motion tests. As such any propeller rate can be imposed on the ship model.

2.6 MEASUREMENTS

During captive manoeuvring tests, the ship model is forced to follow a predetermined trajectory applied by the towing carriage. The ship model is rigidly connected to the planar motion mechanism except for the vertical motions (heave and pitch). The running sinkages are measured at four positions on the vessel (bow-stern/port-starboard).

The forces acting on the ship model, the rudder and the propeller are measured as well as the propeller rate and the rudder angle. In some cases wave gauges were mounted at different locations in the towing tank to capture the water level variations.

3 RESULTS AND DISCUSSION

3.1 CANAL WIDTH AND CROSS SECTION AREA

In [5] results are shown of a tanker (KVLCC2) sailing in a rectangular cross section with a wide range of width and water depth combinations (Figure 4).

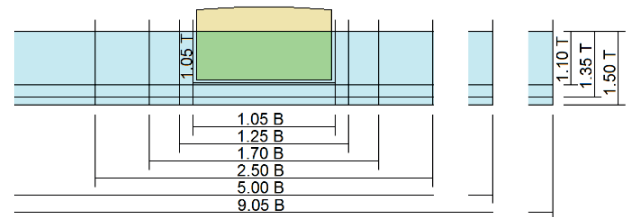


Figure 4. Rectangular cross sections in which the KVLCC2 was tested [5].

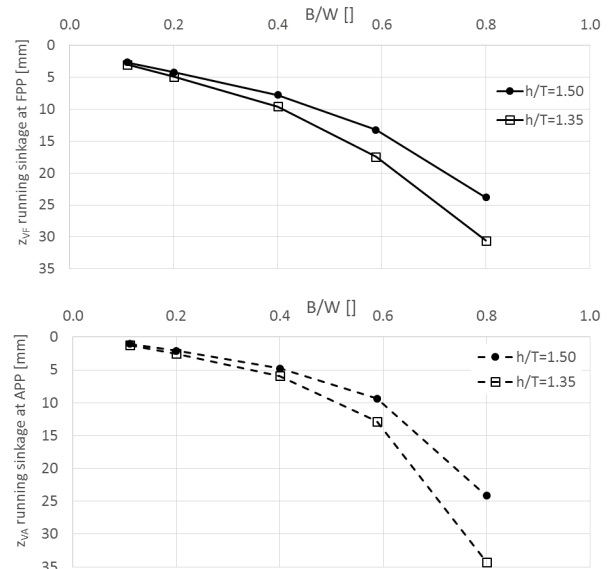


Figure 5. The running sinkage of the KVLCC2 at 6 knots (full scale) in a range of rectangular cross sections and two water depths.

In Figure 5 the running sinkage at the fore and aft perpendicular is plotted for this ship model sailing at a speed according to 6 knots full scale in rectangular cross sections with a width of 125, 170, 250, 500 and 905% of the ship's beam and an initial water depth of 135 and 150% of the draft. All captive tests in this figure are performed with the ship on the centre line of the cross section and with a propeller rate according to the self-propulsion propeller rate at 6 knots in open water. Because of the increased sinkage in very shallow waters, the tests at $h/T=1.10$ are not included because these were not possi-

ble in the narrow sections at 6 knots and hence no systematic comparisons could be made.

The difference in sinkage is significant when the most open test section (towing tank walls) is compared with a test section only 25% wider than the ship's breadth. In the first case the maximum sinkage is (scaled to full scale) 0.20 m and as much as 1.80 m in the narrow section for the ship sailing under the same conditions (6 knots, centre line, $h=1.50T$).

In Figure 6 the sinkage is plotted to the blockage ratio (ratio between the midship section area A_M and the cross section area of the canal Ω), an exponential correlation can be observed. Since the tests in both water depths are on the same correlation, the impact of shallower water depth seems to be the same as a more narrow canal (when the change results in the same cross section area). The dashed and full line in Figure 6 are both an exponential function and only added for the interpretation of the figure. In [6] the equivalent blockage m_{eq} was introduced, in a future publication the correlation between the running sinkage and this m_{eq} will be covered in detail.

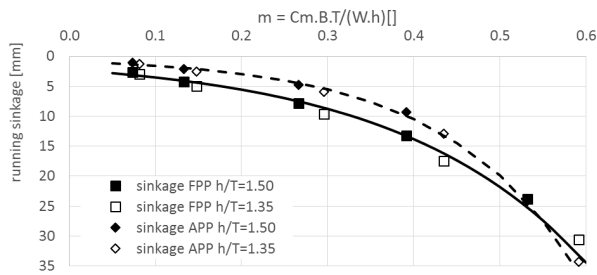


Figure 6. The running sinkages plotted to the blockage ratio m for the KVLCC2 at 6 knots in the different tested rectangular cross sections.

3.2 LATERAL POSITION

In Figure 6 all plotted test results were carried out with the ship model sailing on the centre line of the cross section, hence the distance between ship and port side bank equals the distance between ship and starboard bank. In Figure 7 only tests in the rectangular cross section with a width of $5.0B$ at a water depth of $1.5T$ are plotted. This means that all six tests in this figure are carried out at the same blockage ratio but an increase of running sinkage can be observed when sailing more eccentric (or closer to one bank) in the section. The value y is the lateral distance from the symmetry line of the section while y_{small} is the closest distance between ship and bank wall. At the side closest to the bank the return flow must be higher to evacuate the water in the smaller space available. As a consequence the pressure drop increases along with the running sinkages.

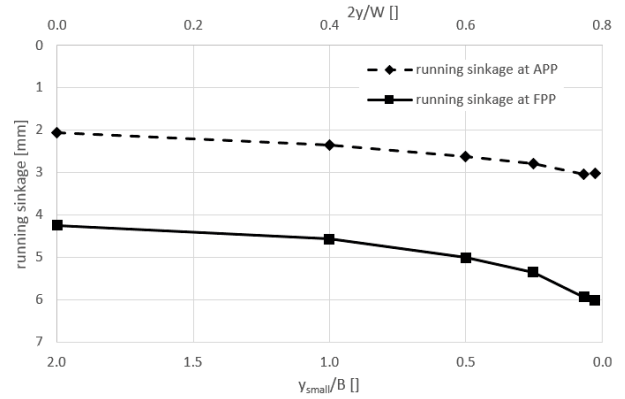


Figure 7. KVLCC2 towed according to 6 knots full scale in a rectangular cross section $5B \times 1.5T$ at six different lateral positions.

A bank will only affect the pressure distribution on the hull if the distance between ship and bank is sufficiently small. As a result, a value for the ship-bank distance can be defined that can be considered as the boundary between open and restricted water. If the ship-bank distance exceeds this value, no (significant) influence of the bank on the ship's sinkage (and more global, her manoeuvrability) will be observed. Therefore systematic model tests were carried out with a modest sized ship model of a tanker which is about half as long as a common ship model at FHR (Table 2).

A systematic database was constructed with this ship model being towed in the towing tank (without extra banks installed) at a range of forward speeds, water depths and lateral positions. In Figure 8 the tests at one forward speed (1.00 m/s) and water depth ($1.35T$) are plotted for 11 lateral positions.

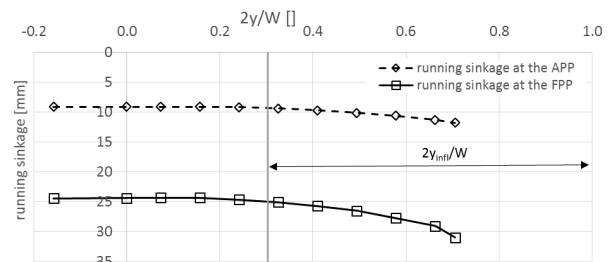


Figure 8. T0H in the towing tank at a water depth $1.35T$ and towed at 1.00m/s.

For the first five lateral positions the difference in sinkage is less than 1%. Therefore it is assumed that these tests are not influenced by the presence of the tank walls. When sailing closer to the port side tank wall the sinkage increases and this is ascribed to the influence of this wall on the hydrodynamic pressure distribution on the model's hull. The bank has thus a lateral reach which depends on the water depth and forward speed [7]:

$$y_{infl} = B(5Fr_h + 5) \quad (1)$$

In other words, a ship sailing at a distance larger than y_{infl} from the closest bank will not be affected by this bank and thus the ship will act as sailing in open (but possibly shallow) water.

3.3 RUN/RISE RATIO OF THE BANK

A common manmade canal often has an isosceles trapezoid as cross section geometry. The deepest section as the navigable area (fairway) and two equally sloped banks at both sides. In Figure 10 a ship model of an LNG carrier is towed with the ship's flat of side above the toe of three different sloped banks. Again, the slope of the bank is expressed as the run to rise ratio with a normalised rise, for a rise according to one unit these banks need 3, 5 and 8 horizontal units (Figure 9).

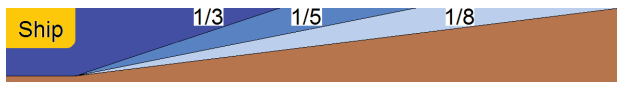


Figure 9. Ship's side above the toe of the three different sloped banks

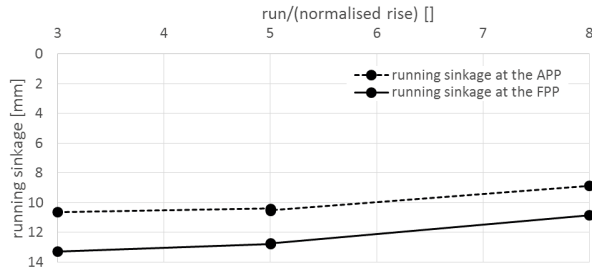


Figure 10. Model of an LNG carrier towed according to 12 knots full scale towed with the port side above the toe of the port side bank for three different bank slopes.

The steeper the bank the more the ship squats, however in these examples it is not easy to separate the influence of the bank slope because a gentler sloped bank will (in this data set) also result in a lower blockage ratio and, as mentioned before, a lower blockage ration will also result in less squat.

3.4 WATER DEPTH

For the RoRo ship model more tests than at three water depths were carried out. These series consists out of tests carried out in the towing tank without extra banks installed, at a range of forward speeds (6, 8, 10 and 12 knots according to full scale) and at three lateral positions ($y = 0.00, 2.06$ and 2.50 m). In Figure 11 the tests at 8 knots (according to full scale) and at lateral position $y = 2.500$ m (in the 7.0 m wide towing tank) are plotted.

In this figure the increase in running sinkage from deep to shallow water can be observed. However, the running sinkage at the aft perpendicular decreases again for under keel clearances of less than 20% (or $T/(h-T)$ greater than 5). This unexpected behaviour is ascribed to the

influence of the boundary layer which is developed along the ship's hull and keel.

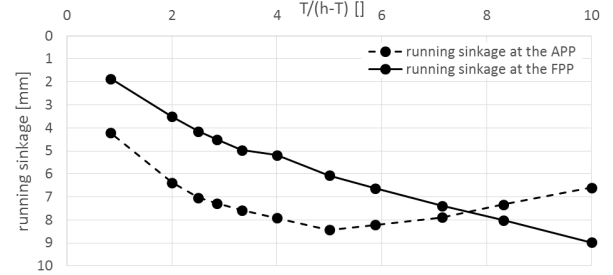


Figure 11. Running sinkage at a wide range of water depths for a RoRo ship model at 8 knots (full scale) and lateral position 2.50 m in the towing tank.

3.5 INFLUENCE OF THE BOUNDARY LAYER

When model tests are scaled according to Froude's law then the boundary layer in the model test will always be relatively thicker than at full scale. This is because fluids of more or less the same viscosity are used on both model scale and full scale (water). The velocity and ship length are both smaller on model scale than on full scale and therefore the Reynolds number will always be (much) smaller on model scale than on full scale. A major consequence of this is the thicker boundary layer on model scale compared to full scale (and relative to the ship length).

$$Re_m < Re_s \quad (2)$$

$$\frac{V_m L_{ppm}}{v_m} < \frac{V_s L_{pps}}{v_s} \quad (3)$$

For reasons of simplicity the viscosities of fresh water and seawater are assumed to be equal, so

$$V_m L_{ppm} < V_s L_{pps} \quad (4)$$

$$\frac{V_s L_{pps}}{\sqrt{\lambda} \lambda} < V_s L_{pps} \quad (5)$$

or

$$\lambda^{1.5} Re_m = Re_s \quad (6)$$

The thickness of the boundary layer at the aft body of the vessel can be calculated according to [8] and Prandtl & Von Karman's momentum law. The boundary layer thickness is defined as the locus of points where the velocity parallel to the plate reaches 99 per cent of the external velocity and is calculated with the skin friction law (assuming a turbulent flow along the plate):

$$\delta = 0.16 \frac{x}{\sqrt{Re}} \quad (7)$$

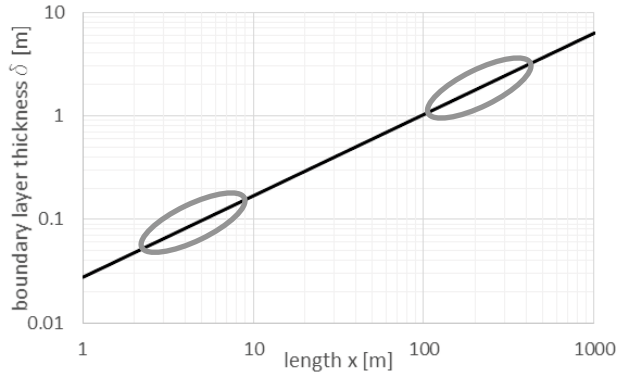


Figure 12. The boundary layer thickness for a wide range of lengths x at a constant Froude number $Fr(x)=0.08$

This boundary layer develops along the hull with the longitudinal position x (with origin at the forward perpendicular and the axis directed towards the stern). The boundary layer is thus thinner at the first half of the vessel compared to her second half, hence the influence being much more significant on the sinkage at the aft compared to the sinkage at the fore perpendicular. The influence of the boundary layer extends the boundaries of the layer itself. In Figure 12 the boundary layer thickness is plotted to the length x and with a constant (x -dependent) Froude number ($V/\sqrt{g \cdot x} = 0.08$). For a ship model the boundary layer (at the aft) is about 0.10m while the boundary layer for the same but full scale ship (at the same Froude number) is about 2.0m thick.

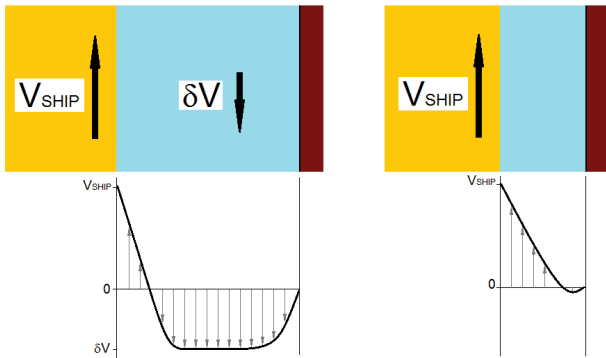


Figure 13. Velocity profile between the ship and bottom or bank.

The viscosity of the water results in water being stuck to the ship, bank and bottom (hence the presence of a boundary layer). This means that the water close to the ship (or bank or bottom) will not be able to evacuate and generate a return flow (Figure 13). Therefore the space available for the return flow is smaller and the influence of the boundary layer will extend its own thickness (by definition reaching 99% of the outer speed). Similar conclusions can be drawn from the other tests (at other lateral positions or forward speeds) as plotted in Figure 11.

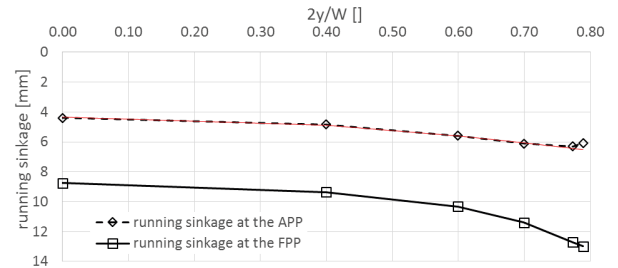


Figure 14. Running sinkage at different lateral positions for the KVLCC2 ship model at 8 knots (full scale) and water depth 1.35T.

Another example of the boundary layer influence is shown in Figure 14, where the running sinkage is plotted for the KVLCC2 towed in the 5.00B wide cross section and a water depth of 1.35T. The two most eccentric model tests were carried out at a distance between the vertical wall and ship as small as 0.050 m and 0.020 m. At the forward perpendicular an increase in sinkage is observed when the ship model is towed closer to the quay wall, but at the aft perpendicular the pace of increase decreases (the plotted red line is proportional to $(\frac{2y}{W})^2$) and the running sinkage z_{VA} even decreases being towed at a distance of 0.020 m compared to being towed under the same conditions but at 0.050 m from the installed vertical wall. Now looking back at Figure 7 the same can be observed for the running sinkage at the aft for the position closest to the bank.

It should be noted that the boundary layer influence will only occur at full scale at even more extreme conditions since the boundary layer thickness is relatively smaller at full scale compared to model scale. For example, the ratio of the boundary layer thickness for a 4 m long ship model ($\delta_4=0.08m$) to a 400 m long full scale ship ($\delta_{400}=3.06m$) (as in Figure 12) is 37 and thus smaller than the scale ($\lambda=100$). In [4] a mathematical model is introduced which takes into account the influence thickness of the boundary layer on the hydrodynamics on both the ship model as well as the full scale vessel.

4 CONCLUSIONS

A sailing displacement ship will generate a deformation of the free surface in the vicinity of the vessel (and generate waves). Close to the ship the (average) free surface level will drop and as a consequence, the ship will have a vertical displacement (often in combination with a trim), commonly known as ship squat.

Within the boundaries of the reach of the banks, defined by the influence width y_{infl} , the entire bathymetry of the cross section will influence the running sinkage of the ship. Both the horizontal boundary (bank) as the vertical boundary (bottom) has a similar effect on the squat. A steeper bank will also result in a higher magnitude of the running sinkages compared to a less steep bank slope.

The viscosity of the water will influence the squatting behaviour of the ship when sailing very close to the bottom or bank. Because of the scale effects, this influence is more pronounced on model scale than at the full scale vessel.

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6 AUTHORS' BIOGRAPHIES

Evert Lataire is currently assistant at the division of Maritime Technology at Ghent University. He has made a PhD on the topic of bank effects mainly based upon model tests carried out in the shallow water towing tank of FHR. His ten year experience includes research on ship manoeuvring in shallow and confined water such as ship-ship interaction, ship-bottom interaction and ship-bank interaction

Guillaume Delefortrie is expert nautical research at Flanders Hydraulics Research. He is in charge of the research in the Towing Tank for Manoeuvres in Shallow Water (cooperation Flanders Hydraulics Research – Ghent University) and is secretary of the 27th and 28th ITTC Manoeuvring Committee.

Marc Vantorre, naval architect, is full senior professor of marine hydrodynamics and head of the Maritime Technology Division at Ghent University, Belgium. His research focuses on ship behaviour in shallow and confined waters, mainly in close co-operation with Flanders Hydraulics Research in Antwerp. He is member of PI-ANC Working Groups and former member of the ITTC Manoeuvring Committee.