THE INSTITUTE OF MARINE ENGINEERS
76 Mark Lane, London EC3R 7JN Telephone: 01-481 8493 Telex: 886841

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STOPPING AND ANCHORING LARGE SHIPS — A FEASIBILITY STUDY

A. J. Morton, B. H. Baines and K. Ridgway

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Stopping and Anchoring Large Ships — A Feasibility Study

A. J. Morton*, MSc, CEng, FIMarE, FIMechE, FInstE, B. H. Baines*, BSc(Eng), PhD, CEng, MIMechE and K. Ridgway†, MSc, BSc, CEng, MIMarE, MIMechE
*University of Manchester and †North Staffordshire Polytechnic

SYNOPSIS

The design of the anchor cable handling equipment on board some large ships would be familiar to a Victorian engineer, using as it often does a steam engine and a band brake to control the anchor cable operations. When dropping anchor a large ship must be virtually stationary or there is a considerable risk of severe damage to the anchor, the cable and/or the windlass. A re-design of the windlass is proposed which will not only improve the security and convenience of normal anchoring operations but also enable a crippled ship to stop and moor in adverse weather conditions. Thus a ship on which the engine has broken down or the steering has failed is provided with an emergency brake which can be used at the speeds at which a crippled ship may drift in adverse weather conditions.

INTRODUCTION

The *Amoco Cadiz* disaster off the coast of Brittany in 1978 highlighted the need for a means of arresting a crippled oil tanker. The cost then was massive pollution but an incident involving a liquid petroleum gas carrier (LPGC) could be much more dangerous. Even before the *Amoco Cadiz* disaster it was recognised that the cable handling equipment on very large crude carriers (VLCCs) was inadequate.[1] A re-design of the cable handling equipment is proposed here which will also allow the anchor to be used to stop the vessel in the event of a main engine or rudder failure.

Conventional anchoring equipment, as almost universally fitted, relies upon a steam, hydraulic or electrically driven windlass to heave or veer the cable in slow time and upon a band or disc brake to control the free run out during an actual anchoring operation. Such equipment has been standard for a century or more and its limitations when applied to very large ships have become clear in recent years.

The friction brakes normally employed do not give positive and reliable control of either cable tension or speed of run out and are incapable of absorbing the energy of a large ship moving at more than a fraction of a knot over the ground. It is therefore necessary to reduce the speed to a very low value before letting go the anchor. In conditions of extreme emergency, as for instance in the case of a large ship drifting without engine power in adverse weather conditions, conventional equipment offers very little help since even if the anchor holds fast the energy of the moving ship will almost certainly destroy either the deck equipment or the cable.

On the *Amoco Cadiz* the port anchor was lowered and the crew attempted to control the cable run out using the band brake. When this was found to be inadequate the steam engine was used to assist the band brake. Eventually the steam pipe to the engine fractured and the two cylinders broke and came adrift from the winch (insufficient attention to water drainage may have contributed to this failure). When the anchor was recovered it was found that both flukes had broken off.[2]

To stop a crippled vessel, some means of absorbing its kinetic energy is required. Taking 2.5 knots as a likely drift speed, the kinetic energy of a VLCC with a mass of 300 000 tonne is about 250 MJ. This is equivalent to heating 4 tonne of oil by about 30 °C, and the energy conversion is achieved in the proposed re-design by throttling the output of a positive displacement hydraulic pump/motor set attached to and driven by the gypsy (chainwheel) of the windlass. The gypsy controls the anchor cable.

When the anchor is in use for an emergency stop the output from the pump/motor set is prevented by relief valves from exceeding say 300 bar, limiting the anchor cable tension to a safe value. If the limiting tension is reached, the anchor pulls out more cable until the cable tension decreases. To absorb the ship's kinetic energy the anchor cable would be pulled out by approximately 250 m with the limiting force set at 1 MN.

This rough calculation is not very representative of the response of a ship in adverse weather conditions whilst being acted upon by a constant anchor cable force. A computer program has been developed to include the effects of the wind

Professor Morton entered the University of Manchester in 1943 and from 1944 was an engineer officer in the Royal Navy. In 1948 he became a lecturer at University College, Swansea. From 1952 he was Deputy Chief Engineer (Trials) at PAMETRADA until he joined the CEGB in 1959 as Assistant Project Engineer at Fawley Power Station. In 1967 he returned to Manchester University as a Professor of Engineering. He retired in 1986. Professor Morton is a former member of Council and a Fellow of the Institution of Mechanical Engineers. His interests include steam power plant, heat recovery systems and automatic control.

Dr Baines served a student apprenticeship at the National Gas Turbine Establishment and in 1954 graduated in mechanical engineering from Imperial College. This was followed by research, sponsored by the Admiralty, on the thermal stresses in steam turbine rotors. In 1960 he joined the staff of Manchester University. He is a Member of the Institution of Mechanical Engineers. His teaching and research interests include stress analysis, metal forming and dynamics.

Keith Ridgway graduated with a BSc (Hons) from UMIST in 1973. After completing a Special Apprenticeship with Mather and Platt Ltd, he joined Kennedy and Donkin, Consulting Engineers, as a design engineer. In 1980 he joined the University of Manchester as a Research Assistant working on the anchoring project. At present he is lecturing in computer aided engineering at North Staffordshire Polytechnic.

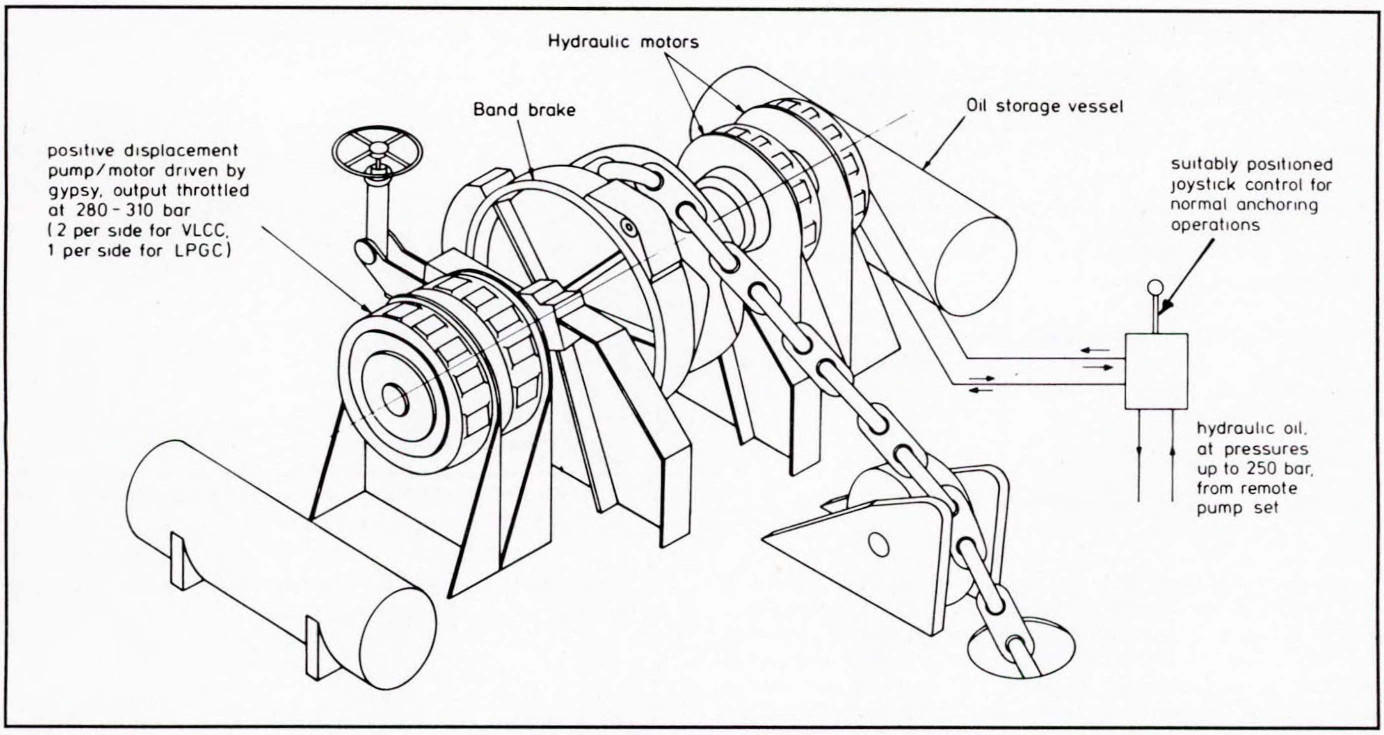

FIG. 1: Sketch of the deck machinery

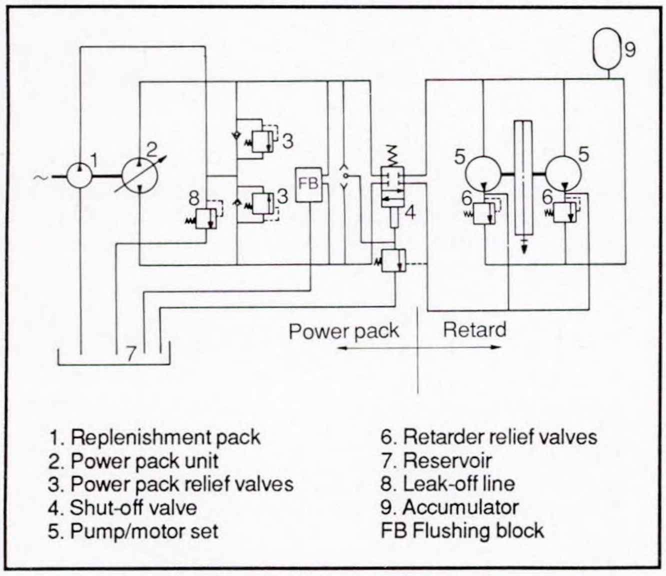

FIG. 2: Windlass hydraulic circuit

and water forces and the anchor cable force (including the catenary effects) on the dynamic behaviour of the ship.

REDESIGN OF THE ANCHOR WINDLASS

The proposed system consists essentially of one or more positive displacement pump/motor sets coupled to the windlass and arranged to pump fluid around a closed circuit through pressure relief valves. A sketch of a likely arrangement of the deck machinery is shown in Fig. 1. It is expected that the whole system will be mounted on a base plate and supplied as a complete unit.

A secondary hydraulic circuit includes a power pack to provide high-pressure oil with which to drive the positive displacement pump/motor sets for normal anchoring operations. A joystick valve in the high-pressure oil line enables the gypsy to be rotated in either direction. The two hydraulic circuits are shown in Fig. 2.

For normal anchor cable handling operations the power pack is used but if the cable attempts to drive the gypsy (as in a stopping operation or perhaps when lowering the anchor) then the shut-off valve (4) closes automatically and the anchor cable is then controlled by the retarder relief valves (6). Changes in the volume of the oil during heating and cooling are accommodated by the accumulator (9). The hydraulic circuits consist of standard components but there may be problems associated with this heavy-duty untested application.

There are several manufacturers of pump/motor sets suitable for this application (MacTaggart Scott, Flender etc.). The MacTaggart Scott radial hydraulic motors cover the required torque and speed ranges, although four of their larger motors would be required on each windlass of a 300 000 tonne VLCC, and for an LPGC of 56 000 tonne two MacTaggart Scott type 10/21/60 motors would be required on each windlass.

Operating at 280 bar these motors give a retarding torque of about 600 000 Nm. This will give a limiting force in the anchor cable of 1 MN (100 tonf), which the computer studies suggest is a satisfactory retarding force for this size of vessel. It is suggested that the vessel used to test the first prototype retarder should not exceed the LPGC size.

Pilot operated relief valves may be used initially as these are commercially available to handle the required pressure and flow rates. These valves do not require external air or electrical supplies and have a rapid response to pressure changes, opening or closing in milliseconds. It will, however, be necessary to study carefully the exact siting and setting of these valves to eliminate any risk from pressure waves or 'hammer' in the hydraulic circuit.

Further study may well lead to the conclusion that better operation and overall performance are attainable with valves operated by a microprocessor responding to a combination of input signals.

COMPUTER STUDY OF SHIP DYNAMICS

In the computer simulation the ship is assumed to be a rigid body with three degrees of freedom: surge, sway and yaw. The effects of roll, pitch and heave are neglected. The motion of the ship relative to an earthbound set of axis can be determined from estimates of the forces and moments acting on the ship, ie wind, current, hydrodynamic drag, wave, propulsion, rudder, anchor (including the chain catenary), Coriolis and inertia. The derivations of these forces and moments are discussed below.

Wind and current forces and moments

The wind and current forces acting on the ship in the longitudinal and transverse directions can be calculated from equations of the form

$$F = A\,C\,V^2$$

Similarly, the moments have equations of the form

$$N = A\,C\,L\,V^2$$

In these equations V is the velocity of the fluid relative to the ship, C is the appropriate drag coefficient, L is the ship's length and A is a constant depending on the relevant area and the fluid density. There are several sets of drag coefficients reported in the published literature but these are incomplete, being based essentially on static tests with the model held stationary whilst the water flows past. Data on the yawing of the ship (which is a feature of stopping the ship using an anchor cable force) have not been reported and the dynamic resistive yaw moments acting on the ship have had to be estimated from the static data.

Computer predictions based on drag coefficients measured on model ships must be used with caution. Also, the published data show a wide scatter for nominally similar ship models. Using drag coefficients from several sources, Brook and Byrne[1] compared the predicted current and wind forces and moments on a 123 350 dwt VLCC subject to a 45 knot wind and a 2 knot current assuming a water depth of about twice the draft. The values obtained differ significantly, in some cases by a ratio of 2 to 1.

The drag coefficients which were used in this study were based on the OCIMF publication 'Prediction of wind and current loads on VLCCs'.[3] Two ships were used in this study, a loaded 300 000 tonne VLCC and a loaded 56 000 tonne LPGC. A loaded LPGC rides higher in the water than a loaded VLCC, and to make some allowance for this the drag coefficients used for the LPGC were those given for a tanker in ballast.

The wind forces and moment thus calculated are consistent with the later OCIMF supplement 'Prediction of wind loads on liquified gas carriers'.[4] All the various drag coefficients are functions of the angle of attack θ between the ship's axis and the fluid stream (zero when the ship is bow into the stream), eg the plot of the transverse drag coefficients against θ is a good approximation to a sine wave but the plot of the static yaw moments against θ had to be represented by Fourier series.

The dynamic resistive yaw moment

When the ship yaws the velocities of the bow and stern can be quite different. This difference in velocity produces a dynamic resistive yaw moment which acts against the swinging motion of the ship, and also a modification to the transverse force on the ship.

Using the OCIMF data for a ship which is not yawing the transverse fluid force on the whole of a ship of length L can be written in the form (see Fig. 3)

$$F_T = K_T\,V^2 \sin\theta = K_T\,V\,V_T$$

where V is the relative velocity between the fluid stream and the ship. Assuming that the transverse force on an element of longitudinal length $d\lambda$ can be written as

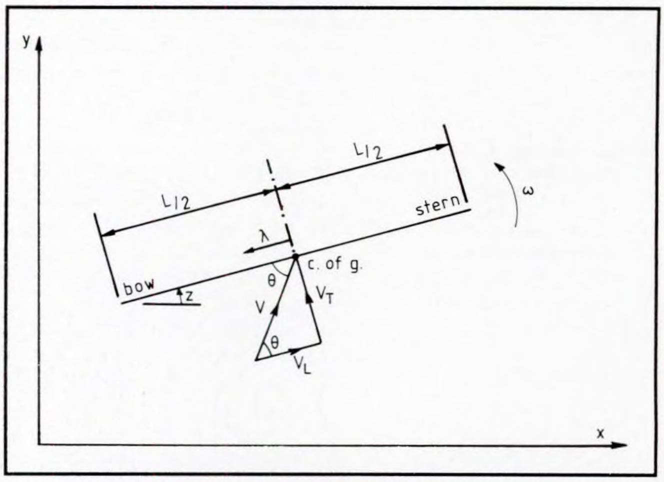

FIG. 3: Co-ordinate system used in the computer simulation

$$d\,F_T = \frac{K_T\,V\,V_T}{L}\,d\,\lambda$$

then this integrates to give a total transverse force on a ship with a yaw rate of ω as

$$F_T = \frac{K_T\left(V_F^3 - V_A^3\right)}{3\,L\,\omega} \qquad (1)$$

The resistive yaw moment due to the transverse force on an element a distance λ from the ship's centre of gravity is

$$d\,N_Y = \frac{K_T\,V\,V_T\,\lambda}{L}\,d\,\lambda$$

Taking the centre of gravity to be equidistant from bow and stern and considering the fluid flow along the ship's sides to give a secondary contribution to the resistive yaw moment, dN_Y integrates to give

$$N_Y = K_N\left\{\left[6U - 8V_T\right]V^3 - 3V_L^3\left[U\,V + V_L^2 \ln\left(U + V\right)\right]\right\}_A^F \qquad (2)$$

where

$K_N = K_T/(24L\omega^2)$,

$\omega = dz/dt$ = the rate of change of yaw angle,

F and A refer to bow and stern, respectively,

$U_F = V_T + 0.5L\omega$ $\qquad U_A = V_T - 0.5L\omega$

$V_F^2 = U_F^2 + V_L^2$ $\qquad V_A^2 = U_A^2 + V_L^2$

When the ship is yawing, V_T and V_L are the transverse and longitudinal components of the relative velocity V between the fluid stream and the ship's centre of gravity (see Fig. 3). V_F and V_A are taken as positive and N_Y acts to resist the ship's rotation. The effect of the ship's beam on V is largely self-cancelling and has been ignored.

When $\omega < 10^{-8}$ rad/s, then $F_T = K_T V V_T$ and $N_Y = 0$ were assumed.

Propulsion and rudder forces and moments

The propulsion and rudder forces and moments are included in the computer program but for this study of the disabled ship these forces and moments were set to zero.

Wave forces

The prediction of wave forces acting on an anchored vessel is difficult as there are many complex factors involved. In the

OCIMF publication 'The prediction of wind and current loads on VLCCs',[3] wave forces are not specifically considered as they are assumed to be negligible compared with the wind and current forces. This is probably true in sheltered moorings but in the case of a disabled ship anchoring in open sea the wave forces may be significant.

Lewison[5] has discussed the problem of predicting wave forces and their effects on a drifting ship and concluded that the effect of waves in addition to wind may be to make a disabled ship drift at a slightly higher speed and to lie in the trough of the waves. It would appear that it is not yet possible to model these effects with confidence and so any additional effects from the wave forces were ignored. The computer simulation must be viewed with this limitation in mind.

Anchor force and moment

The significance of the anchor chain catenary and force limiting device is well illustrated by the graph of cable tension against time given in Fig. 4. Here a sufficient length of anchor cable has been paid out to form a satisfactory catenary. As the ship moves away from the anchor the catenary tends to straighten. This causes the cable tension to rise at an increasing rate until a very small movement of the ship gives a very large increase in the cable tension.

Thus the catenary has a profound effect on the dynamics of an anchoring or moored ship and cannot be ignored. When the preset limiting force is reached more cable is pulled out until the cable force decreases below the limiting value. The dragging of an anchor and the dynamic behaviour of the cable handling system and of the chain itself, ie its inertia, hydrodynamic and friction (seabed and hawse pipe) forces, are all considered to be secondary effects well worthy of future study but necessarily neglected at this stage.

In the computer simulation the horizontal distance from the hawse pipe to the anchor is computed. Using this distance, the cable weight per metre and the well known catenary equations,[6] the tension in the cable can be calculated and hence the horizontal force and the consequent moment applied to the ship.

Coriolis force

The Coriolis force is given by $F = 2MV\gamma \sin \beta$ and always acts at right angles to the direction of motion of the vessel (to the right in the northern hemisphere and to the left in the southern hemisphere). In this equation M, V and β are the mass, velocity and latitude of the ship and γ is the angular velocity of the earth (72.7×10^{-6} rad/s).

For a 300 000 tonne vessel at latitude 55° N drifting at 2.5 knots the Coriolis force is 46 kN. This may sometimes be significant and so it has been included in the simulation, although for the solutions given this term was set to zero.

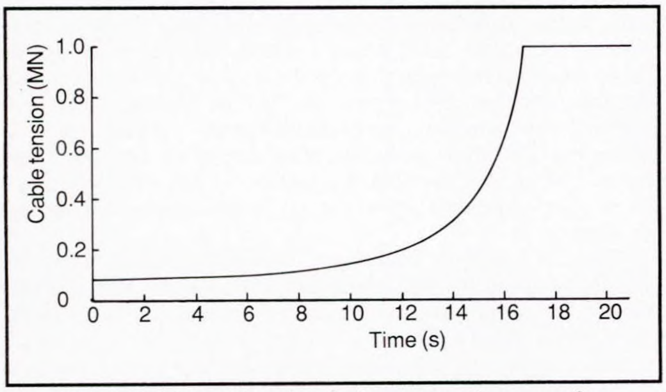

FIG. 4: Cable tension rise during a stopping operation for a 56 000 tonne LPGC initially drifting in a 2.5 knot current and a 50 knot wind, both acting in the same direction

The equations of motion

A force balance on the ship in the x direction and in the y direction and a moment balance leads to the following six first-order differential equations

$$Du = F_x/m \qquad Dv = F_y/m \qquad D\omega = N/I$$

$$Dx = u \qquad Dy = v \qquad Dz = \omega$$

where x and y are the co-ordinates of the ship's centre of gravity and z is its angular position (see Fig. 3), F_x and F_y are the summations of the x and y components of the forces discussed above, N is the summation of the applied moments and D is the operator d/dt.

These six differential equations were solved using a fourth-order Runge-Kutta procedure.[7] No special difficulties were encountered, and on the whole the procedure is stable for time increments of 12 s or less. For most solutions a time increment of 3 s was used, with occasionally a 1 s solution to give an accuracy check.

FREE DRIFT STUDIES

A study has been made of the behaviour of a freely drifting ship under the action of various combinations of wind and current speeds and directions. In the steady-state condition the ship sets itself within a small angle of one of the two possible broadside-to-the-wind positions (where the wind and current yaw moments are in equilibrium) and drifts with a velocity and direction which depend on the relative speeds of the current and wind.

The small angle is generally less than 15° and shows the effect of the wind force on the aft accommodation area. The ship has a small oscillation about this position with an amplitude of about 5°. At lower wind velocities, less than about 20 knots, the ship's drift angle becomes less definite. At very low wind speeds the vessel is accelerated beam-on to the current but may drift at any angle once it has reached the current velocity.

The computed drift velocity is very dependent on the drag coefficients, the relative areas exposed to the effects of wind and water and the angle between the wind and water directions. In still water the computed drift velocity was in good agreement with that calculated by equating the transverse wind and current forces, ie (wind velocity)/73 for the VLCC and (wind velocity)/33 for the LPGC.

These velocities are very much at the lower end of the very wide scatter in the replies received by Holder et al.[8] in response to their questionnaire on drift behaviour. They state that their summary covered geographical positions where the 'water depth was invariably at least 7 times the ship's draft so no adverse shallow water effects were likely'. Anchoring would not be feasible in such a depth and the 20 m underkeel clearance chosen for the present computations would significantly increase the current drag force relative to the wind force and hence give a lower drift speed.

Taking the OCIMF data for a loaded tanker in still water of depth 6 times the draft gives a computed drift speed of (wind velocity)/44.5 but this is still not as high as the average from the survey. Brook's and Byrne's[1] comparison of wind forces suggests that the transverse wind force obtained by several other reputable organisations is up to 50% higher than the OCIMF figure. Allowing this gives a drift speed of (wind velocity)/36, which agrees well with the ratio of the transverse drag coefficients given by Lewison[5] for a loaded tanker. The average drift speed from the survey is about (wind speed)/32.

This argument gives some confidence in the relative values of the wind and current drag data — at least the computed figures are seen to be sensible. It is arguable to what extent a casual survey is reliable (lower drift speeds not being reported as they are 'uninteresting' is one obvious example), and partly for this reason we have used the unadulterated OCIMF

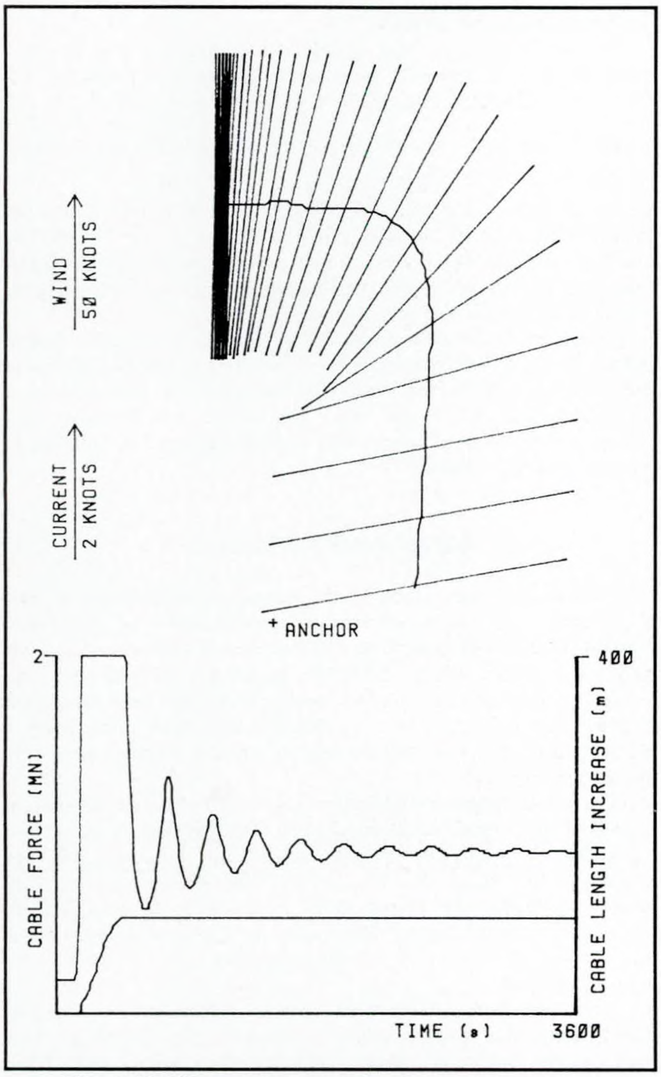

FIG. 5: VLCC stopping operation for a 2 knot current and a 50 knot wind in the same direction. Upper diagram shows ship position at 1 min intervals

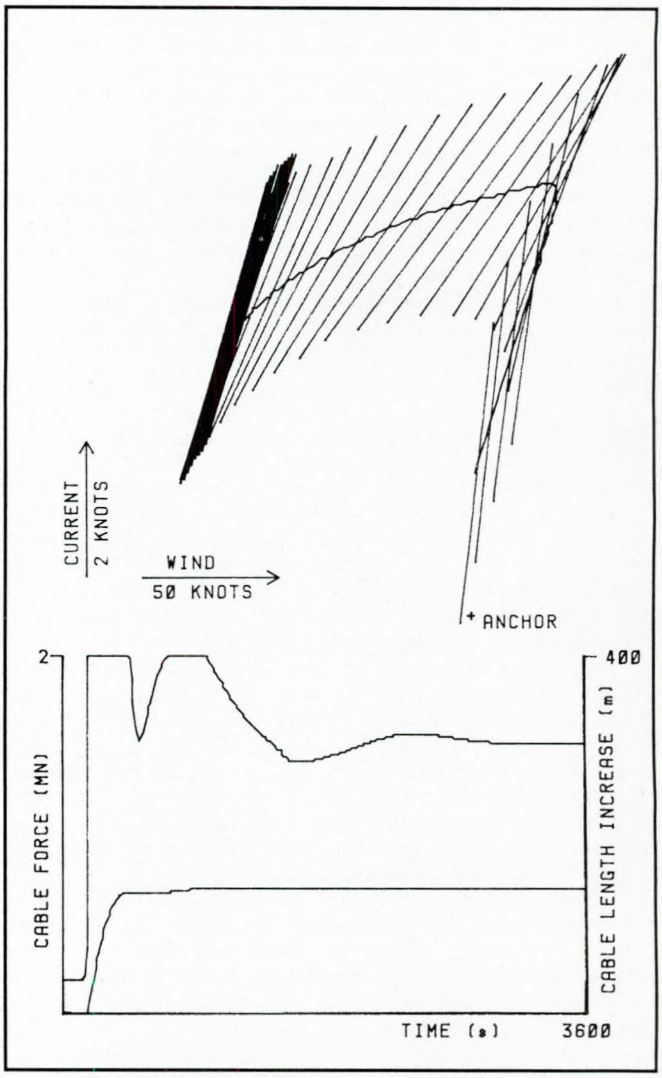

FIG. 6: VLCC stopping operation for a 2 knot current and a 50 knot wind at right angles. Upper diagram shows ship position at 1 min intervals

figures in the computations. Nevertheless, the above argument might be used to stretch the wind speed scale by 20%, ie the dynamic behaviour computed for a wind speed of 30 knots might be possible at 25 knots.

STOPPING AND ANCHORING A DRIFTING SHIP

The maximum cable tension at the hawse pipe was taken as 2 MN (200 tonf) and 1 MN (100 tonf), respectively, for the VLCC and LPGC.

A 20 m underkeel clearance was chosen and the cable catenary lengths are then 214 and 199 m, respectively. Only one anchor is deployed. The actual behaviour of the anchor is not considered, the anchor being assumed to bite where it is dropped. The anchor forces assumed are within the capabilities of anchors currently fitted to these ships.

A typical computation starts with the initial conditions obtained from a previous free drift study for the set wind and current velocities and directions. The anchor lies on the seabed at the start and the drifting ship then 'walks out' the cable until the length to form the catenary is out plus 5 m to hold the anchor stock on the seabed. As the ship drifts away from the anchor the cable catenary begins to form and the cable tension begins to rise at an increasing rate. If the cable tension reaches the preset limiting force the pressure relief valves on the arrester are assumed to operate, holding the cable tension at this limiting force whilst more cable is pulled out.

The cable continues to be dragged out by the ship's motion until there is sufficient cable out for the catenary force to drop below the limiting force. The pressure relief valves then close and the ship is held by the cable catenary force unless subsequently either (a) that force drops to zero because the ship's bow moves towards the anchor or (b) the ship's bow moves further from the anchor causing the cable catenary force to rise once again to the limiting force thus making the arrester operational again. Whilst this is happening the ship is being acted upon by the wind and current forces and the resulting motion of the ship was found to be very dependent upon the wind and current speeds and directions.

There are several identifiable modes of behaviour as the ship is stopped by the anchoring force. Both ships exhibit similar behaviour patterns within the context of the differing physical parameters. Compared to the loaded VLCC, the LPGC is much lighter and also rides higher in the water so that it is affected more by the wind and less by the current, and also has a relatively high anchoring force. Thus to produce the same behaviour pattern as the VLCC the LPGC generally requires a higher current speed and a lower wind speed.

Wind and current in same direction

The simplest anchoring operation is when the wind and current velocities are in the same direction. The ship then drifts more-or-less broadside to the wind until the cable force rises, and this force and the weather forces together create a moment which turns the ship head to the weather.

If the weather is severe the arrester will be operated whilst the ship is turned but even conventional equipment will hold the ship once it is in the head to weather position. When there the ship tends to perform a small figure-of-eight type motion (which may eventually die out) with the consequent snatching on the cable. These effects are shown in Fig. 5 for the VLCC in a 2 knot current and 50 knot wind.

Wind and current at an angle

When the wind direction is at an angle to the current direction the dynamic behaviour of the ship becomes much less predictable. Figure 6 shows the VLCC in a 2 knot current at right angles to a 50 knot wind. It can be beneficial to have the wind and current at an angle; in the extreme case it is possible to have wind and current directly opposed and have the ship drifting at almost zero speed. Conventional anchoring equipment might then stop the ship, eg with 2 knots opposing 60 knots on the LPGC the transverse current and wind forces are equal and the drift speed is less than 0.3 knots.

However, an angle between the current and wind directions is likely to produce the phenomenon known as 'kiting'. In this state the moments on the ship (including the moment due to the anchor force) are in equilibrium but the ship is at an angle to the current. The forces on the ship (mainly, it would seem, the current force) then cause the ship to sail across the current, gradually slowing down under the action of the very high anchor force, until it comes to rest in a state of static equilibrium.

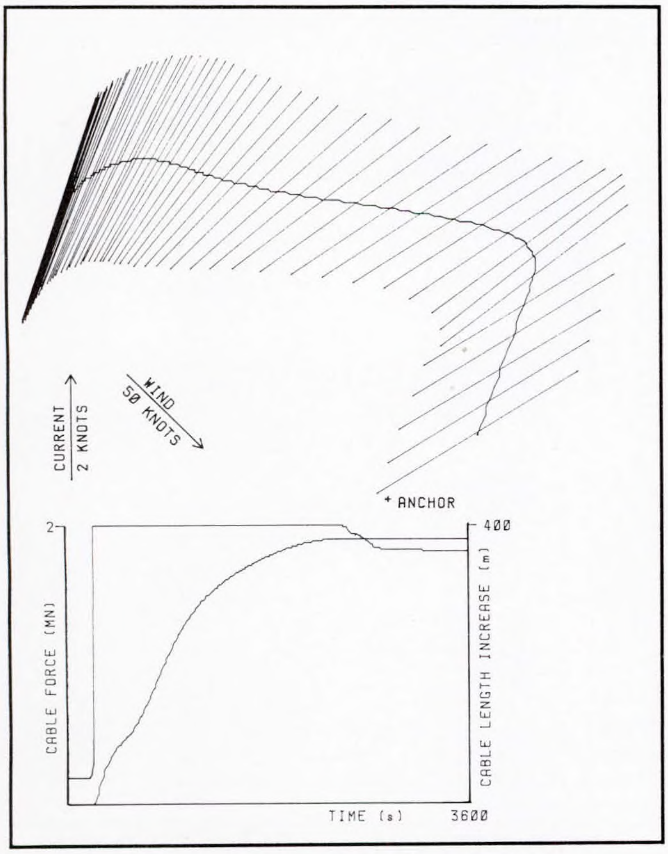

FIG. 7: VLCC stopping operation for a 2 knot current and a 50 knot wind at an angle of 135°. Upper diagram shows ship position at 1 min intervals

An example of this is shown in Fig. 7 for the VLCC in a 2 knot current at 135° to a 50 knot wind. In this case the wind is blowing the ship towards the anchoring point but the wind force is opposed by the current force acting on the inclined ship. There is a strong static equilibrium in this stationary position with steady-state wind and current conditions.

In the real situation where wind and current velocities are not constant this equilibrium will perhaps be broken. If that happens the likelihood is that the ship will turn under the action of the anchor force and then sail back across the current. At the end of this traverse the chance of the ship finding a static equilibrium position would be small and another sail back to the static equilibrium position shown in Fig. 7 seems very probable.

Wind and current directly opposed

Continuous sailing across the current from one extreme position to another is rather more likely when the wind and current are directly opposed. The ship then overrides any possible static equilibrium position at either end following a traverse across the current and performs a rather large and continuous figure-of-eight type of motion. At the end of each traverse the ship is virtually stationary and turning very slowly. The ship then has very low kinetic energy but the cable force is high.

As the ship completes its turn and starts the next cross traverse the anchor force draws the ship's bows towards the anchor point. The ship then sails to bring the bow nearer to the anchor point and the anchor cable may go slack. During

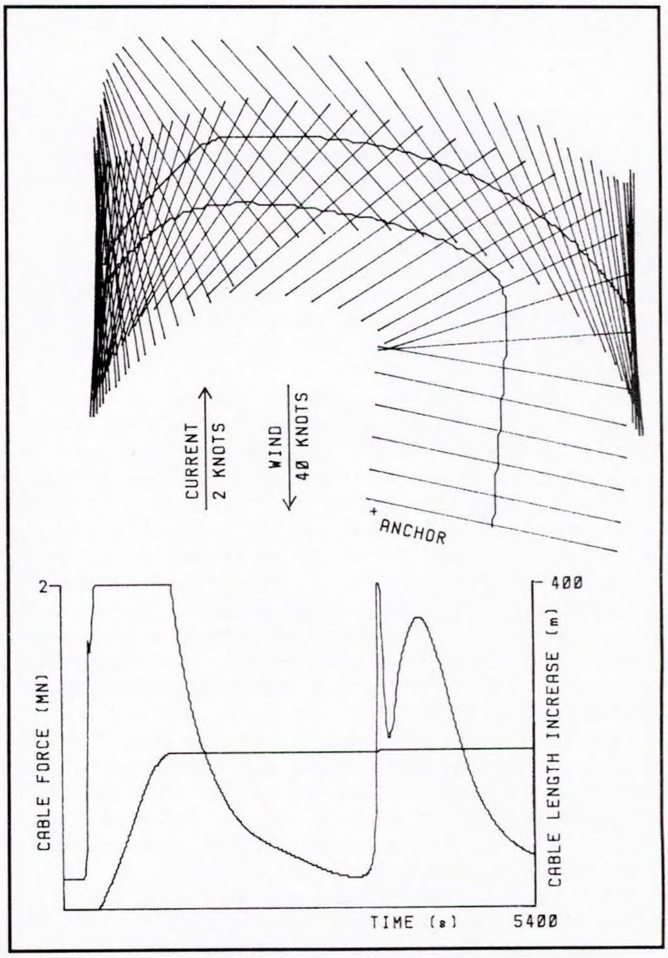

FIG. 8: VLCC stopping operation for a 2 knot current and a 40 knot wind directly opposed. Upper diagram shows ship position at 1min intervals

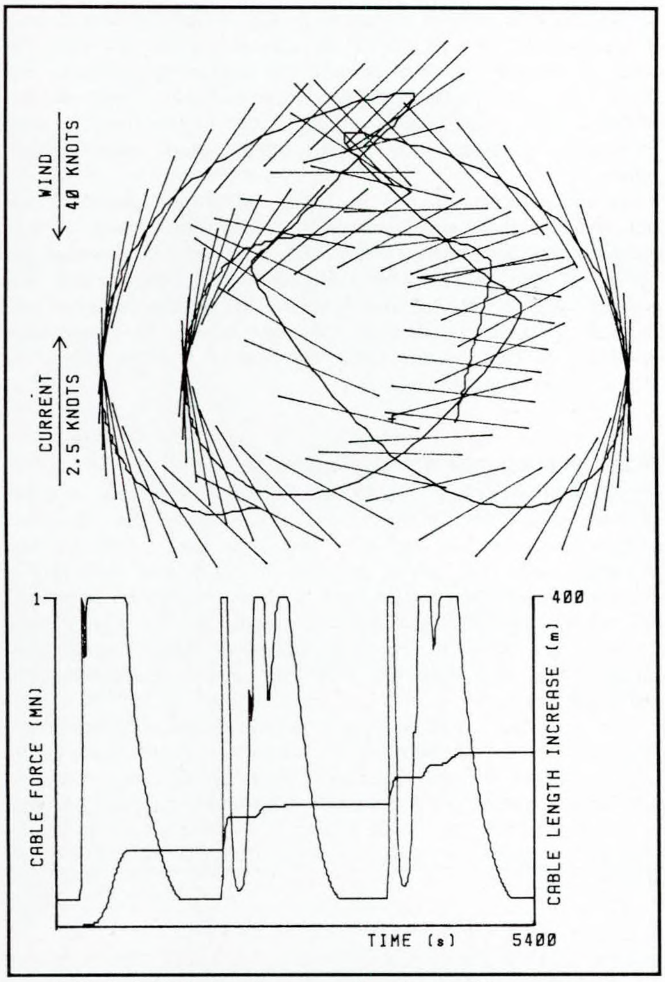

FIG. 9: LPGC stopping operation for a 2.5 knot current and a 40 knot wind directly opposed. Upper diagram shows ship position at 1 min intervals

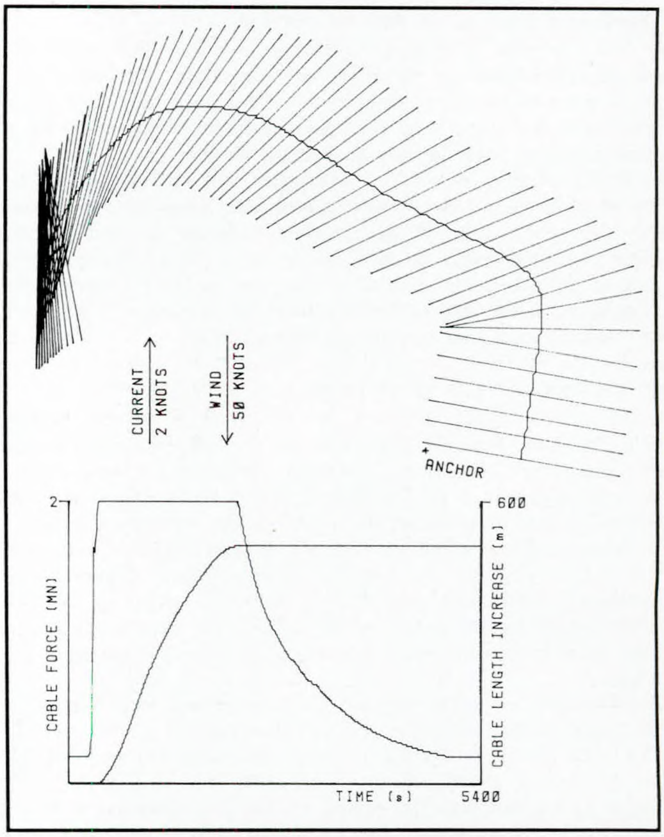

FIG. 10: VLCC stopping operation for a 2 knot current and a 50 knot wind directly opposed. Upper diagram shows ship position at 1 min intervals

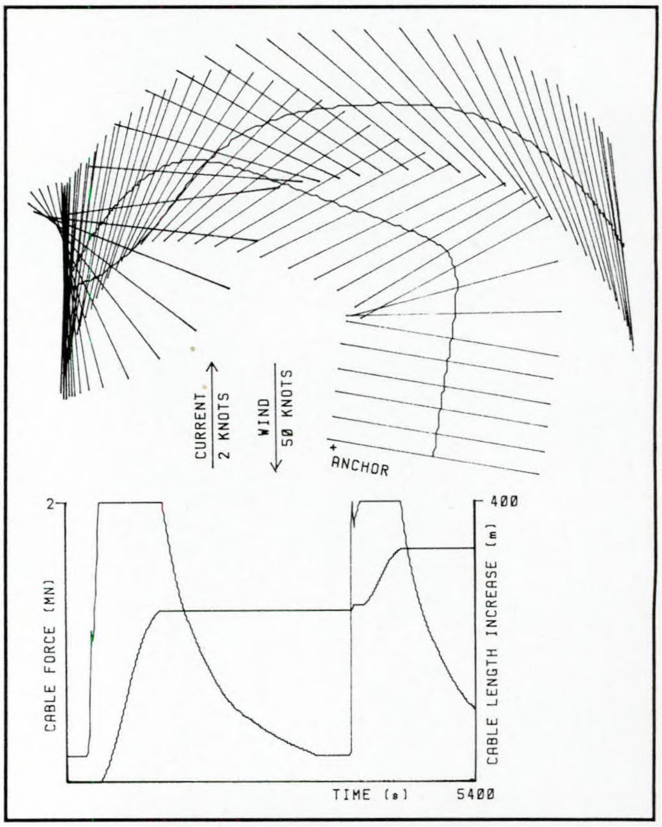

FIG. 11: VLCC stopping operation for a 2 knot current and a 50 knot wind directly opposed using half the dynamic resistive yaw moment given in Equation (2). Upper diagram shows ship position at 1 min intervals

the sailing the ship's velocity is increased by the wind and current forces and by the time the anchor cable catenary force again rises the ship's kinetic energy can be higher than in the original free drift state and the arrester may again be activated.

The timescale here is significant, and for the VLCC the arrester may be operational for perhaps 20 min and then holding for 45 min or more before being required again. The LPGC is rather more lively and these times are lowered to perhaps 7 and 18 min, respectively. This pattern of behaviour with current and wind directly opposed is found most strongly at 2 and 40 knots for the VLCC (see Fig. 8) and 2.5 and 40 knots for the LPGC (see Fig. 9). Above these conditions the ship may turn very slowly, or not at all, at the end of the first cross traverse. If the ship stays inclined to the current it may then be carried away and control is lost. This is shown in Fig. 10 for the VLCC in a 2 knot current directly opposed to a 50 knot wind.

The LPGC behaviour shown in Fig. 9 cannot be taken literally as the ship sailing round and over the anchor point will almost certainly dislodge the anchor. However, Fig. 9 illustrates graphically the way in which a moored ship can be buffetted by the weather. Under similar circumstances the deployment of a second anchor is clearly desirable.

The directly opposed current and wind state moves from being benign to potentially very dangerous with little apparent change in the weather, eg for the LPGC, 2 and 50 knots could be stopped with conventional equipment but 3 and 50 knots could not be stopped with a single anchor even with the arrester. This raises two points to be further examined: (a) how often and for how long is a strong state of current directly opposing the wind (say within 20°) likely to be met and (b) can a second anchor be deployed usefully?

Cable rewind

Should the cable go slack during the kiting, it could be wound in with benefit if auxiliary power is available to rewind at say 0.2 m/s. However, this is a rare case and power for cable rewind supplied specifically for a stopping operation is unlikely to be worthwhile.

The dynamic resistive yaw moment

The dynamic resistive yaw moment has a significant effect on the yaw rate at the end of each traverse. It would seem that the ship's kinetic energy in rotation may carry the ship over a possible static equilibrium position and hence allow the current the chance to start the ship kiting in the opposite direction.

The derivation of the resistive yaw moment given above assumes a transverse force distribution along the ship which is proportional to the square of the longitudinal distance from the ship's centre of gravity. This gives undue significance to the forces near the bow and stern of the ship, which must decrease to zero over a finite length of the ship.

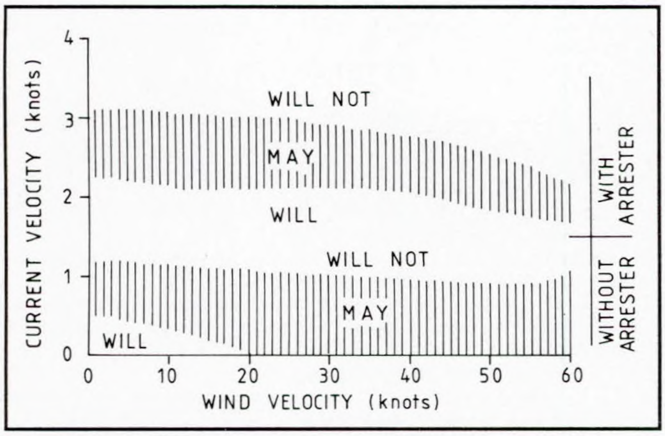

FIG. 12: Estimated stopping conditions for a VLCC. Maximum cable tension and length are 2 MN and 500 m, respectively

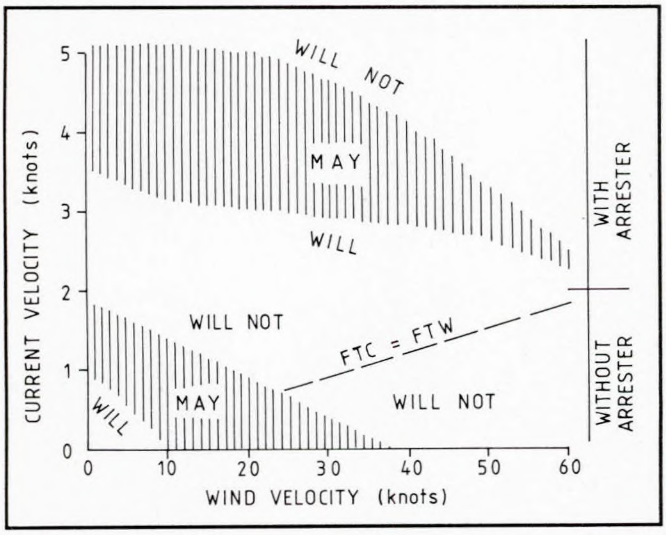

FIG. 13: Estimated stopping conditions for an LPGC. Maximum cable tension and length are 1 MN and 500 m, respectively

Some tentative experiments using a 1 m long model VLCC in still water did suggest that Equation (2) overestimates the dynamic resistive yaw moment. Even at model size there is an obvious lack of reliable experimental data on the dynamic behaviour of ships.

To examine the effect of this moment a few computations were performed using half the resistive yaw moment calculated from the above formula. The dynamic behaviour of the ship was then found to be rather more lively but modified with some benefit in several cases. For example, the conditions shown in Fig. 10 are for the VLCC in a 2 knot current directly opposed by a 50 knot wind. This was repeated using half the resistive yaw moment given by Equation (2), producing the results shown in Fig. 11.

Low wind speeds

With wind speed below about 20 knots the ships have a less definite drift setting and the possibility exists that they may drift at any angle. For any given weather conditions the ship drifting bow first is the most difficult to stop. After the ship is stopped by the anchor force it is turned and accelerated by the current force. When the ship is about broadside to the current the arrester is again needed to slow the ship and bring it bow into the current.

A survey of the computer results

The wind and current velocities for which a drifting VLCC or LPGC can be stopped are shown in Figs 12 and 13 for the survey with the arrester fitted (using one anchor only with not more than 500 m of anchor cable out). The ship can be stopped for conditions below the hatched area but not for conditions above the hatched area. Within the hatched area the ship may be stopped but much depends on the drifting ship's heading (for low wind speeds) and/or the current and wind directions.

For a wind velocity above about 20 knots the upper limit of the hatched area is obtained with wind and current in the same directions and the lower limit when wind and current are directly opposed. For a wind velocity below about 20 knots the upper limit is obtained when the ship is broadside to the current and the lower limit when the ship's heading is in the direction of the current.

The dotted line on the LPGC graph shows the state when wind and current are directly opposed and the transverse wind and current forces are approximately equal and opposite, ie *FTW* = *FTC*. In the unlikely event of this happening the ship could (in theory) be brought to rest using conventional anchoring equipment for conditions in the immediate vicinity of this line. The equivalent area on the VLCC graph is submerged in the lower hatched area because of the higher dominance of the current forces.

A similar survey, also shown on Figs 12 and 13, assumed that the arrester was not fitted and that the maximum safe operating force on conventional anchoring equipment is the same as the limiting force used in the arrester, ie 2 MN for the VLCC and 1 MN for the LPGC.

DESIGN OF ANCHORING EQUIPMENT: IMPLICATIONS OF THE COMPUTER SIMULATION STUDIES

When choosing a suitable pump/motor to limit the cable force the most significant parameters are the torque, the motor speed and the oil pressure. With gypsy diameters of 1.75 and 1.2 m and maximum cable forces of 2 and 1 MN, the torques on the VLCC and LPGC are 1.75 and 0.6 MNm, respectively, ignoring the hawse pipe efficiency. The maximum speed at which the cable is pulled out always occurs at the instant that the arrester becomes operational. The higher cable velocities occur when wind and current act in the same direction, ie the easiest conditions for stopping the ship, and have values generally less than the drift speed because the cable catenary force is already reducing the bow velocity. A 2.5 knot current together with a 50 knot wind gives maximum cable speeds of 1 and 1.8 m/s and gypsy speeds of about 11 and 29 rev/min.

Taking the MacTaggart Scott hydraulic motor range and operating at the upper design pressure of 310 bar, the VLCC will need four 12/25/85 motors (two piggy-back style on each side of the windlass) having a nominal maximum speed of 17.5 rev/min. The LPGC windlass could be designed around two 10/21/60 motors, one on each side: these motors will give the required torque at 310 bar pressure but have a nominal maximum speed of only 21 rev/min. It has been estimated that a working pressure of 310 bar and a speed up to 40 rev/min might reduce the expected life from 7500 to 5000 h, which is still far in excess of the expected usage in this application.

At the maximum cable speed the oil flow rate is estimated at about 60 kg/s for both ships and the local temperature rise in the throttle is then about 16 °C.

If the ship is to be stopped using 300 m of cable run out with a 2 and a 1 MN limiting force, and the oil temperature rise is not to exceed 30 °C, then the oil circuit must have a volume of at least 10 m^3 for the VLCC and 5 m^3 for the LPGC. The temperature of the oil may rise 30 °C in a period of time as short as 3 min when wind and current are together or as long as 25 min with current opposing the wind. In the latter case the arrester may be needed again 18 min or more later as the ship performs a figure-of-eight type motion. With the oil storage tanks exposed in the bow of the ship, the spray and wind may well be enough to cool the oil but this is a design detail for further consideration.

The very rapid increase in cable tension as the anchor cable catenary forms is a feature which must be considered. The most rapid tension rise occurs when wind and current are in the same direction and is illustrated in Fig. 4 for the LPGC in a 2.5 knot current and a 50 knot wind. This suggests that in the latter part of the build-up the cable tension may double in one second. Pilot operated relief valves have a fast actuation time and should be able to cope but it may be wise to consider a microprocessor system which can anticipate the necessary action. The hydraulic motors have four outlet ports and it is intended to supply each with a relief valve to minimise the mass of fluid in the pipework upstream of the relief valves.

Shock waves in the pipework is potentially a problem but it is felt that this can be overcome with careful design. However, it is not expected that the cable tension against time graph for a practical system will be as clean as that assumed in the computations.

CONCLUSIONS

There is no inherent reason why a large crippled tanker cannot be stopped using the existing anchor if the anchoring equipment incorporates a system to limit the anchor cable tension. The system proposed here, where the gypsy drives a hydraulic motor with a throttled output, could be supplied by a specialist builder on a single baseplate and simply bolted into place on the ship. Normal anchoring operations would be simpler and could even be controlled from the bridge since band and disc brakes, steam engines, draining steam pipes etc. can be eliminated.

ACKNOWLEDGEMENTS

This project was initiated and sponsored by Shell International Marine Ltd and the authors are grateful to Mr R. J. Clements and Mr D. G. M. Carpenter for their help and advice, and also to Rexroth plc, Sunderland Forge plc and MacTaggart Scott plc.

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Discussion

R. J. CLEMENTS (Shell Seatex): I should like to congratulate the authors on their paper, which describes the results of their study so clearly and contributes a great deal to the knowledge of the behaviour of an anchored ship.

Since some time has elapsed, I think it is worthwhile putting this particular study into the context in which it was first considered. As the size of ships increased during the late 1960s and early 1970s, the size of the anchoring equipment was increased accordingly. However, it would appear with hindsight to have been a case of selecting a suitable anchor and chain with sufficient holding power, purchasing the appropriate gypsy and then fitting a motor and brake which were adequate for the most likely depth of water in which the ship would anchor.

The dynamic behaviour of the equipment does not appear to have been considered, although this did not become important until the large anchors and chains required for VCLLs of 300 000 dwt and above were fitted. The result was that a number of anchors and chains were lost from ships when the operators lost control during lowering operations: once control had been lost, it was impossible to regain it before the whole chain ran out, causing considerable damage and ship delays.

We approached this problem in three ways. The first was to investigate alternatives already available on the market and we fitted a disc brake for evaluation. This was very successful, although expensive, and enabled an anchor chain to be stopped safely, even when running out at a frightening speed and when soaked with a fire hose.

The second approach was to commission a study at the University of Hannover with the purpose of understanding all aspects of the operation of the existing equipment. This work led to the testing of windlass brakes on which the drum had been coated with stainless steel and which gave a more consistent performance over a longer period. The apparent disadvantage of the lower heat transfer coefficient was outweighed by avoiding corrosion of the brake drum.

The third approach was to ask an academic body to consider the whole question, including that of stopping a drifting ship, without any pre-conceived ideas. The result is this paper, although from necessity it has only included a description of the equipment considered to be the most suitable for the purpose. As Mr Ridgway said in his presentation, there were a number of alternatives considered but rejected for practical reasons which looked very different from the conventional anchor handling equipment.

It was agreed initially to limit the scope of the study by making the assumption that the anchor was down and holding. Although the method of achieving this from a drifting ship is not obvious, it was considered to be beyond the scope of the project originally envisaged and would be a suitable area for further study.

The system described in the paper does resemble that proposed by Mitsubishi in the mid-1970s but has a much greater retarding power available. The Mitsubishi system was designed solely as a dynamic control system for paying out anchor cable and could not be considered for stopping a drifting ship. Consequently, the greater capability of the system described by the authors enabled it to be patented by Shell and we would be interested to discuss further development with manufacturers of this type of equipment.

Finally, I should like to conclude with two questions. The first is to ask the authors if they could give an estimate of the cost of retrofitting the system they have described. The second is to ask whether it would be possible to consider the effect of putting the ship's rudder hard over in one of the conditions described in Figs 5 to 11. It appears to me that it should be possible to improve the final position of the ship relative to the anchor or to damp out some of the wild oscillations shown.

B. RAPO (Lloyd's Register of Shipping): I should first like to compliment the authors for producing a very interesting paper on a very interesting subject.

In his paper 'Ten year review of defect and failure in large ships' anchoring and mooring equipment' published in 1979, A. Buckle suggested three possible solutions for stopping and mooring large drifting vessels. They were:

1. Substituting rope for chain.
2. Using sinkers, springs or buoys in the system.
3. Fitting the windlass with a quick response dynamic braking system.

Of these three suggestions, option 3 was thought to be most practical and worthy of development. The solution proposed by the authors of the present paper offers a promising variant to option 3.

On page 9 of the paper, when discussing Figs 12 and 13 the authors refer to anchor pay-out of no more than 500 m. However, for a quarter million tonne vessel the total Rule length of the chain cable on one side of the vessel is not greater than 385 m. It is also noted that in several of the examples presented by the authors, such as in Figs 9 and 10, the amount of cable veered is in excess of that which would normally be fitted to one anchor. One should also be aware that the end connection of the chain cable in the chain locker is neither designed or normally capable of accepting the full load required to stop the vessel. The authors' comments on this would be appreciated.

Also, if the limiting value of 200 t is applied to the windlass as suggested and in the event the anchor snags a reef, the anchor flukes, originally proof tested to 150 t under ideal conditions, would most probably be lost, as they were on the *Amoco Cadiz*. This would suggest that the windlass release load should be set at a lower value which would require even more chain cable to be carried. Alternatively stronger anchors would need to be fitted.

Persuading owners to have one or even more lengths of extra chain cable on board to enable emergency stopping of vessels is something which nowadays needs a degree of bravery beyond the call of duty. This should not, however, be interpreted as a suggestion that this avenue should not be explored further.

M. H. P. HEMBLING (Lloyd's Register of Shipping): I should also like to congratulate the authors on an interesting paper, and endorse the comments made by Mr Rapo concerning the length of cable considered in the calculations. I also question the problems of stowing greater lengths of cable within the forward ends of the ships concerned.[1]

Secondly, at what speed would the cable be run out when the pressure relief valves lift, ie when the cable tension reaches its predetermined maximum? It is a well known problem that when cable is run out of the chain locker at excessive speed it is very prone to tangling within the chain locker and jamming in the bell-mouth.

Finally, refering to page 9 of the paper, it would appear that the authors dismiss the possibility of cable rewind. However, Figs 8 to 11 show that the cable tension periodically drops to below about 15% of the predetermined maximum. I do not know the lifting capacity of the windlass considered but I feel sure that it is within the capabilities of the machinery concerned to rewind some cable under these conditions. This would then leave the ship with extra cable in hand which could be veered out if conditions should worsen.

Further reference to Figs 8 to 11 shows the development of an oscillatory motion, known to seafarers in its basic form as fishtailing, and under such conditions the peak loads will occur on a continuous cyclical basis. If weather conditions are such that these peak loads exceed the present limits of the windlass then, unless the rewind facility is used, it is only a matter of time before the ship runs out of cable. If by this time circumstances have not changed (eg improved weather conditions or availability of main engines), the ship would stand the same risk of losing its anchor as with the traditional windlass and chain-stopper. It would therefore appear to be of considerable advantage to incorporate the rewind facility into the system.

The system proposed by the authors could provide further benefits to the ship's Master. There is no doubt that any ship using traditional windlass equipment can be successfully anchored, but that success is critically dependent on the speed over the ground, a factor which is difficult to determine to the accuracy required for a large ship.

It has been shown[2] that the accuracy of Doppler logs is doubtful at low speeds, so the Master is usually dependent upon visual or radar fixes. In fact it has even been known for a hand-held lead-line to be used. The proposed system would give the Master an extra margin of safety in all anchoring situations and reduce his dependence on often inaccurate methods of speed measurement.

With reference to the authors' comments on page 10 regarding the very rapid increase in cable tension, it is felt that the problem could be overcome using a purely hydraulic system. The introduction of a microprocessor into the system, particularly considering their relatively poor record of reliability in the marine environment and the fact that if they are going to fail they do so without warning, would only reduce the reliability of the total system. As the critical factor would be the rate of increase of cable tension, it is thought that a system could be developed which could detect the pressure surge in the hydraulic pump/motor.

It is anticipated that this may work by careful design of the pipe layout but could, if necessary, be combined with extra pilot-operated relief valves located to detect the pressure surge and to open before the shut-off valve closes and the retarder relief valve opens (Fig. 2, items 4 and 6). Such valves would, of course, close again once the system reached steady-state conditions (ie cable being run out under maximum tension). Hopefully such extra complications may prove to be unnecessary but that is something for the hydraulics engineers to decide.

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Dr N. E. MIKELIS and G. D. W. LEWIS (Lloyd's Register of Shipping): The authors have addressed a serious problem and have used ingenuity in building up a mathematical model to demonstrate the value of the proposed arrester. They have presented findings in a most readable paper and the Institute is to be congratulated for encouraging the publication of such papers.

We are not sure, however, as to why the authors have developed the equations described in the paper when there are already agreed and proven equations based on a wealth of data from the maturing field of ship manoeuvrability [see for example Ref. (1) or the proceedings of the forthcoming RINA International Conference on Ship Manoeuvrability, April/May 1987].

We would suggest that if the authors were to replace their equations by a set of the well documented manoeuvring equations they would find that the resistive yaw moment, and added masses, are modelled correctly, as is also the effect of shallow water on all hydrodynamic reactions. The same equations will predict the effect of current on the ship's motion, without resorting to coefficients which are based only on static (pure drift) tests.

As a suggestion for a further improvement in simulation we would like to point to a paper by Gould[2] which provides a thorough treatment of wind forces and moments for a variety of ship forms.

Having made these comments regarding other work which could be used to generalise the authors' research, we wish to offer them our congratulations for bridging the usual gap between academia and the industry with their relevent and practical paper.

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D. B. FOY: The authors have failed to mention the greater holding power of the Bruce anchor and the greater strength of the Bruce-designed anchor cable.

The greatest contribution the authors have made to the problem in my view is their work on windlass design as three components are needed: anchor, cable and windlass. The loaded VLCC must not be moving over the ground when anchored and the Doppler log could have enabled the *Amoco Cadiz* to choose the right time to anchor. Her engines could then have relieved tension on the cables

G. M. ELSOM (Marine Representation): Concerning the Bruce anchor, it has the property of planing downwards, providing drag and then good holding.

T. LINDSAY (Lloyd's Register of Shipping): I should like to ask the authors if any thought was given during the study to the fitting of one anchor instead of two on board ships.

W. F. SPANNER (Spanner Marine Corporation): The authors of this interesting paper mention two contingencies: engine failure and failure of the steering gear. Both these contingencies are most likely to arise when a vessel is proceeding at or near full speed as most of the vessel's working life will be so spent. It is very desirable to work out beforehand the procedure to be followed in each case.

The authors have dealt mainly with the procedure to be followed to bring a VLCC to rest when its speed has dropped to 2.5 knots. To do this they propose to re-design the windlass so that more effective control by hydraulic braking can be excercised over the run out of the cable. This is good so far as it goes, but when all the cable has been run out the limitation on the tension in the cable will cease as the hydraulic braking is only effective when the cable is turning the gypsy on the windlass. The tension may then rise above the limit.

The authors have made no mention of the effects of a heavy swell which would be expected to cause pitching motions to a greater or lesser extent, nor of the effect of the vertical component of the cable tension acting at the entrance to the hawsepipe. This latter probably has only a small effect, but I imagine the effect of a heavy swell could be considerable.

I believe that for the exceptional and grave contingency of main engine or steering failure something more is needed than the authors have indicated for emergency stopping of the vessel and subsequent anchoring. Features to permit safe anchoring of the ship in such conditions (assuming depth of water permits) can and should be incorporated in the design of these ships.

Finally I should like to thank the authors for the work they have done on this subject.

Professor A. W. CROOK (Brunel University): I am a layman in the subject of the paper, but perhaps the authors would comment further upon Figs 5 and 6 which seem to present anomalies to a lay view.

With reference to Fig. 5, the symmetry of the situation would suggest that the vessel should come to rest downstream and downwind of the anchor point or possibly oscillate about that position, whereas the figure shows the vessel coming to rest to the left of the symmetrical position. If the wind is removed from Fig. 6, one might expect the same situation to obtain, and that the addition of the wind would move the rest position downwind. However, the figure shifts the rest position upwind.

Figures 5 and 6 depict intriguing situations which are unlikely to escape the notice of mariners, and perhaps the authors would also consider whether there is support for their predictions.

M. J. KENN (Consultant): The authors have commented that their calculations are based on uncertain values of drag coefficients stemming largely from simple model tests.

In this connection I should like to point out that, under equilibrium conditions, the combined force on the ship due to wind and current is equal and opposite to the horizontal component of the restraining force at the ship exerted by the chain catenary. This latter force could be ascertained at full scale merely by recording the water depth and by measuring the angle of the chain at the ship (relative either to the water surface or to a shielded vertical plumb line), provided that sufficient length of chain was in use to be certain that any pull at the anchor was horizontal and provided that the weight for unit length of chain was known.

If, additionally, the steady wind and current velocities were observed, together with the chain-angle measurements, certain drag coefficients could also be determined at full scale, at least for some of the wind and current conditions, and the correctness of any corresponding model data could be assessed.

Dimensionless graphs, showing the essential, and unique, relationships between the catenary geometries and the corresponding chain forces at the ship, the angles of the chain at the water surface, and the chain forces at the anchor have been published previously.[1,2]

In support of Mr Foy's comments on the merits of the Bruce anchor, it was because twin-fluked, stockless anchors were inherently ill conceived and basically unstable that they were adopted with such extreme reluctance (after sail had given way to steam) by both the Admiralty and private ship owners and, indeed, were later entirely rejected by the Royal Air Force for use in sea-planes and flying boats.[2,3]

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N. S. MILLER (YARD Ltd): The authors have attempted to deal with a subject of considerable practical importance. They rightly draw attention to the very poor quality of data which exist for determining the wind and more especially the current forces.

The current force data are all based on tests on models in test tanks or wind tunnels and recent papers by Palo[1,2] and Edwards[3] have indicated that many of these results have been affected by Reynold's number associated scaling effects.

This is also indicated by the very different shapes of the longitudinal force coefficient for tankers indicated by the OCIMF data and results from other model tests.[4,5] At heading angles around 40° off the bow the OCIMF data indicate considerable lift forces whereas other tests indicate large drags at those angles. This makes the type of analysis carried out by the authors open to some doubt as to the validity of their results.

I have been unable to understand the analysis given on page 4. To use the OCIMF drag coefficent data the velocity V does not need to be resolved into its longitudinal and tranverse components and thus the equation for F_T should either be:

$$F_T = 0.5\rho C_D A_T V^2$$

when C_D is given in the OCIMF data against heading angle or

$$F_T = 0.5\rho C_D A_T (V \sin \theta)^2$$

where the drag coefficient has been modified to take account of the resolved wind velocity.

The authors do not give the value of K_T but it must involve a partial modification of the OCIMF values and raises the question of the correctness of the subsequent integrations.

Although the expression for N_Y involves V_L, the longitudinal component of the velocity V, there appears to be no term which involves the drag coefficients associated with a current along the hull, nor the turning moments associated with the velocity V. It is only by considering the balance between all the forces and moments that one can determine the heading angle under some specified wind and current conditions.

I find it difficult to believe that a VLCC will drift almost broadside on to a 50 knot wind and 2 knot current, especially in view of the variation of the OCIMF longitudinal force coefficients with heading angle. I would have anticipated that the vessel would become much more nearly head to wind and current.

The authors have used a depth of water of 20 m under the keel of the vessel for the mooring studies and the associated OCIMF data on the increase of drag coefficients in shallow water. The data in Refs (1) and (3) suggest that these coefficients may be substantially too high and that the current drag force may have been overestimated.

Although the authors state there is little data on wave drift forces there have been a significant number of papers in recent years which give such data for tankers and a few other ships.[4-8] These forces may well dominate loaded tanker behaviour in a seaway but for ships in ballast the wind loading will usually provide the major forces and moments.

The paper does not make clear the treatment of the dynamical behaviour once the mooring cable begins to take effect. This is a very non-linear problem, especially if the winches are paying out more cable when the loads exceed preset limits and the dynamical behaviour of the complete system is very complex. It would be helpful if the authors could clarify their treatment of this dynamical problem and particularly how the difficult subject of damping was tackled. Wind gust effects and their associated spectrum may also play a significant part in ship/cable dynamics.

The techniques of mathematical simulation have greatly improved in recent years but they are still very dependent on the quality of data available to determine both the exciting and restoring forces on mooring systems. It is also necessary to improve the definition of the environmental situations to be used for design, eg current profile below the surface, time average to be used for the steady wind velocity and by implication the appropriate gust spectrum, the wind profile above the sea as a function of wind speed and averaging time, the type of wave spectra and the associated significant heights and zero-crossing periods.

It is hoped that this paper, having highlighted some of these deficiencies, will spur the industry on to reduce some of the uncertainties involved.

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Captain E. H. BEETHAM: While serving as Master of a half-loaded VLCC approaching Suez Canal from the south at the commencement of canal transit with the engine (motor ship) on full ahead (12 knots) and the ship moving at 9-10 knots, I wanted the engine stopped but (for various reasons) the engine would not stop. Within one mile radius were 15-16 ships, the nearest being a gas tanker less than half a mile away.

The objectives had to be first to avoid hitting other ships and secondly stop the ship somehow or even reduce speed as much as possible prior to grounding. Such circumstances do recognise some damage as acceptable.

The action taken was first to try and steer the ship to port (with the rudder) and at the same time let go the port anchor, check it at 6 shackles (each shackle 90 ft) and then let go to 12 shackles and screw up the brake. The starboard anchor was then let go to 6 shackles, checked and let go to 12 shackles before the brake was screwed up.

This checked, stopped and held the ship without grounding and without damage to anchors or cables. The engine stopped soon after as the fuel had been turned off.

The next day the cables, brake linings etc. were checked and all were found to be okay. Indeed 3 weeks later we rode out 50 knot winds at anchor for 36 hours. The ship was 11-12 years old and of about 140 000 gt (ship 35 000, cargo 105 000). That was an immediate decision and 3-4 years later I am still of the view that it was the correct decision.

Was it a miracle or are we trying to convince everybody (ourselves included) that VLCC anchors have so little chance of being successful that VLCC Masters are discouraged from trying them? In an emergency such anchors/windlasses will, in my opinion, serve the vessel to check her drift, help turn her into the wind etc.

When commanding a ship one has to make the best use of the equipment available and most of us were taught 'never go aground with both anchors in the pipe', which philosophy can still be constructurely applied in a VLCC.

I don't dispute the need for improved anchoring systems but while friction brakes/static loading windlass are fitted we should not lose sight of the need to make the best use of them until such time as alternative theories evolve into practical and available realities.

B. P. THOMAS (Lloyd's Register of Shipping): The authors are to be congratulated on a very clear and logically written paper.

It seems that under the main reoccuring load condition, most problems will be due to the peak loads. Therefore, the safety features could be improved by back up control/regulation systems and machinery mounting arrangements based on a means of load compensation. This would protect the anchor, chain and deck-supporting structure.

The machinery aspects present no major problem provided the load conditions are realistic and a reasonable degree of redundancy of operation is allowed for in the overall design.

D. CARPENTER (Shell Tankers (UK) Ltd): The authors should be congratulated not only for introducing a system that might possibly aid the reduction of marine casualties involving large vessels but also for highlighting the shortfall of data available on the dynamic behaviour of such vessels during an anchoring operation.

During the course of their investigations, and prior to arriving at the proposed system, the authors must have investigated the feasibility of other large ship retardation devices. An insight into the avenues pursued and later rejected would be appreciated bearing in mind the rather conservative approach to anchoring equipment displayed in the past.

As a result of recent studies into the dynamical behaviour of windlasses, the problems associated with the inadequacy of the braking system of windlasses of VLCCs should be reduced with the introduction of better brake drum material, improved brake drum design and enhanced control systems. Have the authors considered that the equipment they propose to arrest a ship in an emergency may in fact reduce any gains made in the reliability of the primary role of a VLCC's windlass, ie low-frequency normal anchoring?

Now that the feasibility studies are complete could land-based trials feature in the system's development to prove the equipment prior to translation to a ship-borne test bed?

Professor M. J. FRENCH (University of Lancaster): I found this paper to be very interesting, both in itself and in its practical implications. The mathematical model so lucidly set out by the authors seems to give a good representation of the kind of motion to be expected without adducing any but well understood dynamical effects. This is very satisfactory, when I for one would have expected that it would have proved more difficult to achieve a convincing 'phenomenology' in the results. The authors themselves stress the one additional element it would perhaps be most desirable to add, that of wave forces.

I wondered whether a hydrokinetic braking system should have been examined as an alternative to displacement machines. The step-up gear such a scheme would require would be likely in itself to cost about as much as the hydraulic pumps and control would not be so simple. On the other hand, there might be a gain in robustness and freedom from maintenance.

Some years ago there was a good deal of activity on the possibility of using a special form of cable for this purpose. This consisted of a resilient water-filled tube with a wall reinforced with unwoven filaments laid on a helix angle of about 45°.[1] When stretched such a tube develops high pressures in the interior, and these can be released through a relief valve to provide the same kind of braking effect. The Japanese have experimented at sea in a small way with such a device, and there has also been activity in this country and Sweden.

However, there have been difficulties with the end fittings and in any case the constructional difficulties are very great and the cost is likely to be high. Combined with the more limited extension under load which can be provided, this solution is interesting but not very practical.

It would be interesting to see a further study which examined the cost-effectiveness of such equipment, both for the operator and for the community at large. It would have the weakness that it would depend upon estimates of the likely numbers of disabled ships and the cost of such accidents, and it would involve such considerations as whether insurance rates might reflect the provision of such equipment. Few as would be the occasions on which it was used, the sums then saved would be very large, so that on balance I suspect it would pay for itself.

With the growing strength of the environmental lobby the pressure for such systems to be fitted is likely to grow, and it would be good to have some idea of their potential cost-effectiveness.

Reference

1. M. J. Platts and M. J. French, 'Improvements in or relating to a fluid displacement device'. UK Patent 2 054 756 A (1980).

G. VICTORY: The paper states that it was recognised 'even before the *Amoco Cadiz* disaster that the cable handling equipment on VLCCs was inadequate' and the need of a means to 'arrest a crippled oil tanker' was highlighted by that disaster. It might be mentioned that a number of 'near misses' of a similar potential hazard have happened since and should have emphasised this need. Now after almost ten years we have a proposal, but to describe what is shown in Fig. 1 as 'a major re-design of the cable handling equipment' is surely gilding the lily.

Just as the IMO resolution on steering gears concentrated on one aspect of the failure and ignored other potential faults which contributed to the disaster so the changes to what is in essence a similar arrangement to the windlass fitted on the *Amoco Cadiz* still perpetuate some of the weaknesses identified which could nullify the ability of 'the anchor being used to stop the vessel in the event of a main engine or rudder failure'.

The expression 'agricultural machinery' was used to describe this clanking clattering conglomeration of antique machinery considered suitable for installation on these most modern of marine giants. Doubtless it was the cheapest option but there were much more refined windlasses (smooth, electrically driven, and with enclosed gearing) on many ships in the 1920s.

Admittedly the rattling reciprocating steam machinery (driving through open straight cut gears and unprotected clutches), hammering around at the end of about a quarter of a mile of steam pipe designed to feed the cylinders with slugs of water rather than steam, has been replaced by hydraulic pumps and motors but some of the faults still exist.

On the *Amoco Cadiz* the Second Mate and his squad risked their lives by even being on the forecastle, and the ferocity of the sea was such that they could not use the starboard anchor as they could not remove the stoppers and slings by which it was secured in place. So much for 'immediate readiness'.

The only braking available was an antiquated arrangement of an external contracting brake band which might have come out of the ark and which could not be relied on to stop the anchor and chain running out, so that the chain had to be paid out using the steam machinery. In addition at times the anchor chain was, as it has been liable to do in other cases, jumping the gypsy and paying itself out, not a satisfactory arrangement.

I realise that Fig. 1 shows the system adapted to existing ships, but for new tankers and LPGCs when we 'redesign the cable handling equipment' we should arrange to eradicate the other faults and take note of:

1. The banning of steam machinery served by long pipes over the open deck.
2. The protection of the working area preferably by installing the windlass in a raised forecastle so arranged as to avoid concentration of flammable gas. (I wonder what the Health and Safety Executive would say about Management if these working conditions were found ashore?)
3. A redesign of the gypsy and the angles of contact of chain and gypsy to avoid the chain jumping. Idlers should be provided to ensure that the angle of contact is at least 180°.
4. The arrangement of stoppers should be improved to ensure that they can be rapidly released (say 10 to 20 s) and be so secure that they would not need the additional strops or slings which are so often fitted in practice.
5. The brake or brakes should be enclosed, unaffected by weather and be to an accepted automotive standard (preferably disc) with adequate cooling arrangements to cope with the worst possible loadings.

Dealing now with the proposed braking arrangments, I would suggest that, desirable as they are, they are open to question as to their effectiveness, as are the factors on which they are based. Much has been made of the use of OCIMFs 'Prediction of wind and current loads on VLCCs' (1977) in calculating forces likely to be encountered in stopping the ship, yet at the Official Enquiry in 1978 no less a person than Dr Ewen Corlett said that in his opinion, and as a result of work carried out on North Sea structures, the forces of waves on flat surfaces (in this case the rudder) were higher than had previously been anticipated. I wonder whether any allowance has been made for the fact that even OCIMF might be wrong?

It is also vital to realise that it is not what is happening at the windlass which is of prime importance but rather what is happening at the anchor end of the cable. One end of the cable may be yielding but if the anchor is snagged in rocks, then that end is unyielding and the forces on it are not those acting on the windlass. The dynamics of cable inertia, the snap reaction of the sudden changes from slackness to tightness in conjunction to the pitch or yaw of the ship may not be so amenable to analysis as the paper suggests.

The paper admits that maximum pay out speed occurs at the instant the arrester becomes operational, which is quite a shock to the system, and also that a 'very rapid increase in cable tension as the anchor catenary forms' which 'in the latter part of a build up may double the cable tension in one second'. This applies at the windlass end and I would have great doubts as to whether the change in catenary slope can be rapid enought to prevent this shock being passed on to the anchor.

Finally it is said that 'shock waves in the system' are a potential problem and that 'it is not expected that the cable tension against time graph will be as clean as that assumed in a practical system'. Might I just say that I am sure that it will not be.

It seems that a simple fixed pressure relief valve will not react rapidly enough to prevent a rise in pressures after it starts to open and, as the paper admits, shock waves in the pipe-work will be propagated by the opening and closing of the valve. Neither am I sure that 'a microprocesesor' could anticipate the neccessary action. Certainly it would be an unnecessary refinement unless the other failure possiblilities have been designed out.

I wonder has the possibility of duplicated valves opening at different pressures or a variable orifice valve giving a variable pressure with increasing flow rate been investigated? This would do much to ease the sudden effects shown on the 'cable tension against time graph' when the valve opens and closes.

Despite the above, the move to produce a solution which will permit Masters to use anchors to halt a ship in a drifting situation is very welcome and it is hoped that it will be refined and even more importantly fitted to all large tankers and liquefied gas carriers.

Authors' reply

We should first like to thank all the contributors to the discussion for their interest and for the good reception accorded the paper.

Mr Clements has performed a useful service by making clear, from the sponsor's point of view, the background to the study and the limitations deliberately imposed on it. We repeat that our object was to explore the feasibility of a new system of cable handling in large ships, assessing its general performance and revealing any serious problems.

On the question of cost, an accurate assessment of the cost of a retrofit system is not possible until the detail design has been finalised. Our best estimate, based largely on the cost of the main items, is about £125 000 per windlass for the smaller vessel. This can be expected to reduce if a market develops and a manufacturer produces motors specifically for this use.

In the situations considered, with the engine incapacitated, the water speed over the rudder is low and so the rudder forces, even if correctly used, are unlikely to have more than a marginal effect on stopping the ship. However, the longer-term application of the small rudder forces may be useful in subsequent mooring and this is another secondary effect that remains to be investigated.

Mr Rapo and Mr Hambling both express concern about the problems of carrying the extra cable required to cope with certain extreme weather conditions. The usual cable fitted to a 250 000 tonne vessel is 384 m long and has a breaking strength of 908 tonf.

One advantage of fitting a force limiting device in the cable handling equipment is that a lighter weight cable could be fitted. Thus it may be that the increased cable length could be accomodated in the existing cable locker. Different cable weights were considered in the final report to the sponsors[1] and were found to change the stopping distance by perhaps 20-30 m.

Mr Rapo also queries the strength and location of the ultimate restraint when the cable has run fully out, and the same point is raised by Mr Spanner. These matters were not covered in the study, but the load capacity of the restraint should match that of the cable and might be provided by either a bitter end attachment or a stopper on deck. Current practice in the design of the latter leaves room for improvement, as pointed out by Mr Victory, and much will depend on the final specification and arrangement of the winch and other equipment.

If a suitable control system was fitted then the working pressure (and hence the cable tension) could be increased as the cable runs out. This possibility remains to be investigated further.

It should be remembered that only in the extreme wind and current conditions would full run out of the cable occur, and conventional equipment would almost certainly have carried away at a much earlier stage.

The anchor itself, as already explained, was outside our terms of reference but again we would expect that it would be matched to the cable. There seems little logic in accepting an anchor that will fail at only a small fraction of the cable proof load as in the case of the *Amoco Cadiz* quoted by Mr Rapo.

The Bruce anchor, referred to by several contributors, appears to have substantial merits.

The speed of run out, queried by Mr Hembling, would inevitably be close to the ship's speed over the ground when the relief valves were lifting under designed maximum load. Under other conditions the possibility of rewind clearly exists and we do not discount it. However it requires power and one of our prime objectives was to examine the performance of the anchoring system with the ship completely dead.

A rewind speed of only 0.25 m/s would require auxiliary power of 0.5 MW for most of the situations shown in the VLCC diagrams. It should be noted that for Figs 8-11 the wind and current are directly opposed, and only in these conditions does the cable force for a single anchor drop to such a low value.

Related to this is the use of a second anchor, a point raised by Mr Lindsay. Although our studies to date have been confined to a single anchor we certainly would not recommend that only one anchor be carried.

The computer results suggest that, under extreme weather conditions, a single anchor may not be adequate to hold the ship. A study of the use of two anchors to stabilise the mooring of the ship is to be undertaken. It seems likely that a slow rewind may then be more practicable and useful.

Dr Mikelis and Mr Lewis question the form of the equations used and the validity of the basic data, as also does Mr Miller. The data are based largely on model tests which are hardly practicable at the appropriate Reynold's and Frounde numbers and so scaling effects are to be expected.

The OCIMF data were chosen knowing that they tend to be higher than other data available. The V^2 terms in the forces and moments given by the OCIMF data give no sense of direction to the computed forces. By resolving the velocities in the longitudinal and transverse directions it is possible to ensure that the directions of the forces are correct.

The drag coefficients are functions of the angle of attack θ and were represented by a single trigonometric term or a short Fourier series as necessary. Obviously all the forces and moments due to the wind and current velocities were included as indicated by the OCIMF data. Additionally the dynamic resistive yaw moment acts against the rotation of the ship and is largely responsible for damping out the ship's oscillation. Without this moment the ship's dynamic behaviour is much livelier, as can be judged by comparing Figs 10 and 11 in the paper.

Professor Crook and Mr Miller draw attention to the curious ship movements predicted in some circumstances. The combination of wind, current and anchor forces produces a complex balance which sometimes needs to be studied before it convinces. The crippled ship drifts more or less beam-on to the wind, an effect produced by the turning moment included in the drag data.

In Figs 6 and 7 this wind moment is opposed by the water and anchor moments which together hold the ship at an angle to the current. In this position the current forces (higher than the wind forces alone) have little difficulty causing the ship to sail against the wind to the final static equilibrium position where the wind and anchor forces together balance the current forces.

Mr Carpenter raises the question of other types of retardation device, and many of these were considered by the authors at an early stage in the study. Most, if not all, of them suffer from the drawback that they tend to reduce the speed of the vessel relative to the sea rather than to the seabed and therefore do not actually bring her to rest. It was felt more profitable to concentrate on controlling the cable properly and reliably so as to obtain the best possible service from a conventional or improved anchor.

Professor French has introduced an original note with his reference to the water-filled cable which generates its own fluid pressure when subjected to tension. It is difficult to comment on this without having figures for energy absorption but it seems doubtful, for the reasons he himself points out, whether anchoring is the best application for this device.

He also raises the question of cost/benefit analysis, which is central to the prospects for the adoption of any new system. The immediate cost of the equipment is plain to see; the benefits are more nebulous because there is always the hope that the accidents the equipment is designed to prevent will not happen anyway or will have to be paid for by someone else.

Environmental benefit such as the avoidance of oil spillage can only be assessed on a statistical basis involving contentious assumptions. However the benefit is not merely environmental; quite apart from the avoidance of occasional dramatic ship losses there can be little doubt that the cable handling system we have proposed would bring substantial direct savings to shipowners and insurers through the virtual elimination of anchor and cable losses during routine anchoring operations. This is by no means an infrequent occurence, as pointed out by Mr Clements in opening the discussion.

Captain Beetham's experience is interesting. He confirms our view that a good way to check the drifting ship is to let go one or both of the anchors and screw up the brake with the scope too small to allow the anchor to bite effectively. Provided it does not snag, the anchor will drag and absorb the kinetic energy of the ship whilst ploughing through the seabed. However, when the anchor snags, as on the *Amoco Cadiz*, damage to the deck equipment is almost inevitable unless a force limiting device is fitted.

Several speakers, including Mr Foy, Mr Hembling and Mr Thomas, refer to the possibly of using more sophisticated instumentation, either to provide information to the Master or to form part of a control loop to achieve improved performance.

This is a matter for serious consideration in any future study, and considerable improvement is undoubtedly possible by hydraulic or electronic means. The harshness of marine conditions necessitates great caution, but the prospect of enhanced safety at sea is of such importance as to justify a major design and development effort and a willingness to look beyond conventional solutions.

This raises a general issue which is clearly very much on Mr Victory's mind, as shown by his cautionary remarks. It is difficult to quarrel with most of his practical points, although his reference to the siting of friction brakes is irrelevant to the present proposal. His reservations as to the reliability of basic data and the adequacy of simplified calculations were uppermost in the our minds throughout the study, as should be clear from the paper itself and from our replies to other contributors.

The result may be only an approximation to reality, but it brings out some valuable points, confirming many initial ideas and highlighting both areas of uncertainty and opportunities for progress. Certainly it exposes the inadequacy of conventional equipment, with its limited energy absorbing capacity and uncertain cable control, when called upon to arrest a drifting ship.

The overall result, in our view, is to demonstrate beyond reasonable doubt that the system proposed in the paper, given competent detail design and prototype development, has every prospect of practical and commercial sucess, and we join with several contributors in trusting that means will be found to bring this about.

Reference

1. A. J. Morton, B. H. Baines and K. Ridgway, *The Anchoring of Large Ships* (Simon Engineering Laboratories, University of Manchester, April 1983).