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THE BRAKING OF LARGE VESSELS, PART I + II

by

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THE BRAKING OF LARGE VESSELS

(Het remmen van grote schepen)

by

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Summary

In this report a method is indicated how to brake large ships. This method is tried out by means of tests in the towing tank of the Technological University at Delft.

As a result of these tests suggestions are given how to fit in installations for braking ships in new designs.

1. Introduction

The braking of large ships tends to become an insoluble problem. To stop supertankers of masses varying from 80.000 to 175.000 tons, an enormous distance of brake-way (stopway) has to be run through. This distance covers many miles and it will be clear, that something has to be done to remedy this state of things.

Even if the propellers are turning astern the difficulties remain. In this case, nearly all ships become totally ungovernable, as is proved by the tests made by JOURDAIN [1] in 1961 and 1962 for the French Research Centre for Shipbuilding (Institut de Recherches de la Construction Navale). That the results of these tests were disappointing can be understood when one considers the means by which the braking was performed.

Indeed, the only way to try to stop a ship consists of backing the propellers by means of turning astern the propulsive machinery. The backwards power thus obtained has about the same effect as when one tries to stop a heavy military tank with a 2 C.V. Citroën engine. In reality the disproportion with the kinetical energy of a 80.000 ton supertanker at 18 knots is even much greater.

The same problem exist in aeronautics. Here the kinetical energy $\frac{1}{2}$ mv^2 is influenced mostly by the enormous speed of jet-liners etc. Fortunately incase of the mammouth-tankers, it is only the mass "m" which has the preponderous influence. But the remedy in both cases must be the same: Try to enlarge the resistance to translation and thus to annihilate the kinetical energy.

2. Stopping-tests at sea

Referring to the French tests, [1], the following conclusions may be drawn:

In stopping-tests two periods must be discerned:

1st. The period of speed-slackening

2nd. The period of stopping.

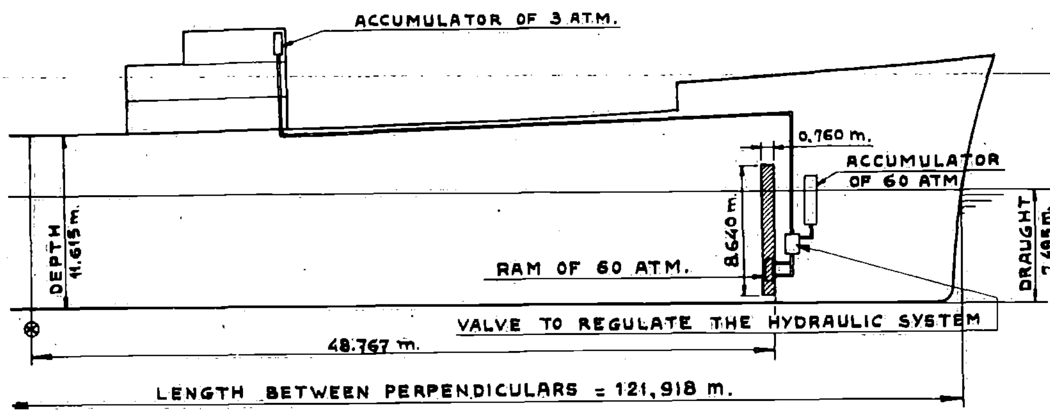
In the first period the ship loses about 75% of her kinetical energy and this loss is due only to the resistance of the hull. The change of direction of the propeller-turning may add to this resistance, but the effect is nearly negligible and there is no sense in brutalising the machinery in this stage of the proceedings. Generally speaking this period covers half the stopping-time and three quarters of the stopway. It is practically impossible to shorten this period.

In the second period, the hull resistance becomes very small and the back-turning propeller becomes important as a braking source. In this period it is interesting to augment the power to the screw and turn full speed astern. One may gain considerably on the last minutes and on the last part of the stopway when the vessel itself has a speed of a few knots only, but on the whole the total stopping-time and stopping-way are scarcely influenced.

As the stopway is very important, it is indicated that in circumstances where a collision is to be feared, the ship will try to manoeuvre out of the danger-zone. Now this cannot be done effectively with propellers turning astern. It can be done when the ship is going full speed ahead, but though instructions give the captain of a ship the liberty to do what seems best to him, he may fear, rightly, that when a collision takes place in spite of this going full speed ahead and manoeuvring, he has a big chance of being reproached for not turning astern.

Therefore it seems difficult to recommend this effective manoeuvre of trying to escape the danger of collision by manoeuvring away in the forward direction. So it is necessary to try other means of increasing the resistance to the forward movement of the vessel.

Tests have been carried out in the Delft towing tank to obtain this extra resistance by means of flaps in the forebody of the ship.



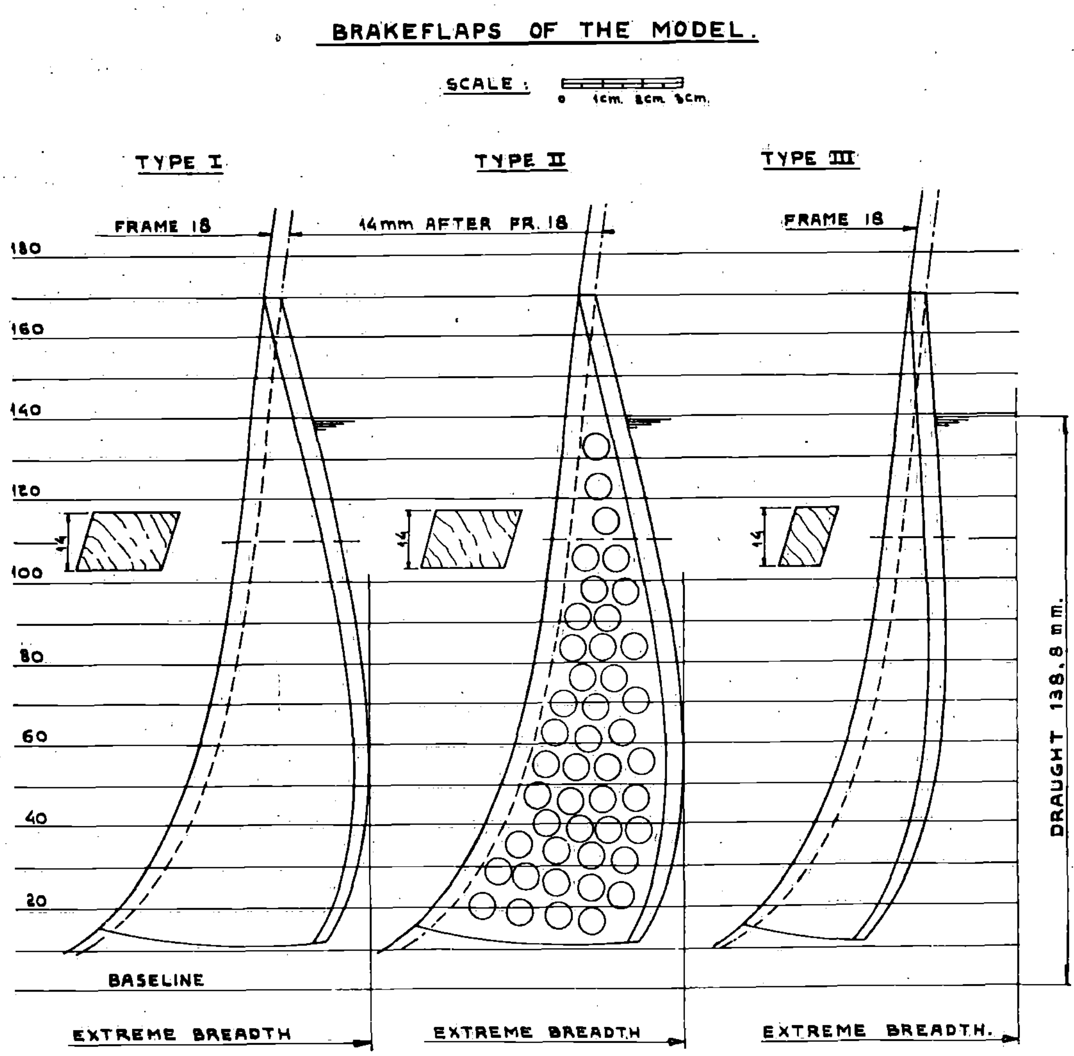


Figure 1. Brake-flaps of the model, situation and types tested.

3. The braking of ships by means of transversally moving side-flaps. Tests.

So far only model experiments were carried out. But it will be seen from the conclusions drawn from the results of these experiments, that a possible and acceptable solution of the problem of the braking of large ships has been found.

The characteristics of the model and the corresponding ship are given in table I.

The hydrodynamic brakes are supposed to consist of flaps moving laterally, as indicated in figure 1. The flaps have been placed at the ship's forebody in such a way that the contour remains within the contour of the midship-section. The placing of the flaps at the forward side of the ship was deemed more advantageous, because there they do not work in the turbulent flow of the wake. The wake itself would probably

TABLE I

Todd Series 60 $\delta = 0,80$	Model No. 42 Scale 1 : 54 and 1 : 50 Delft Towing Model Basin No. 4214 W - B4	
	Ship (Design)	Model (Design) Scale 1 : 54
L_{WL} waterline length	123,962 m.	2,2956 m.
L length between perpendicular	121,918 m.	2,2577 m.
B breadth	18,757 m.	0,3474 m.
T draught	7,495 m.	0,1388 m.
Δ displacement	13,737 m^3	87,091 dm^3
α_{pp} waterline coefficient	0,871	0,871
β midship section coefficient	0,994	0,994
δ_{pp} block coefficient)	0,800	0,800
φ_{pp} cylindrical coefficient	0,805	0,805
⊠ Surface of midship-section	139,200 m^2	4,79 dm^2
CG_L centre of gravity in length till midship section	+ 3,048 m.	+ 56,4 mm.
Ω wetted surface	3455,8 m^2	1,1851 m^2

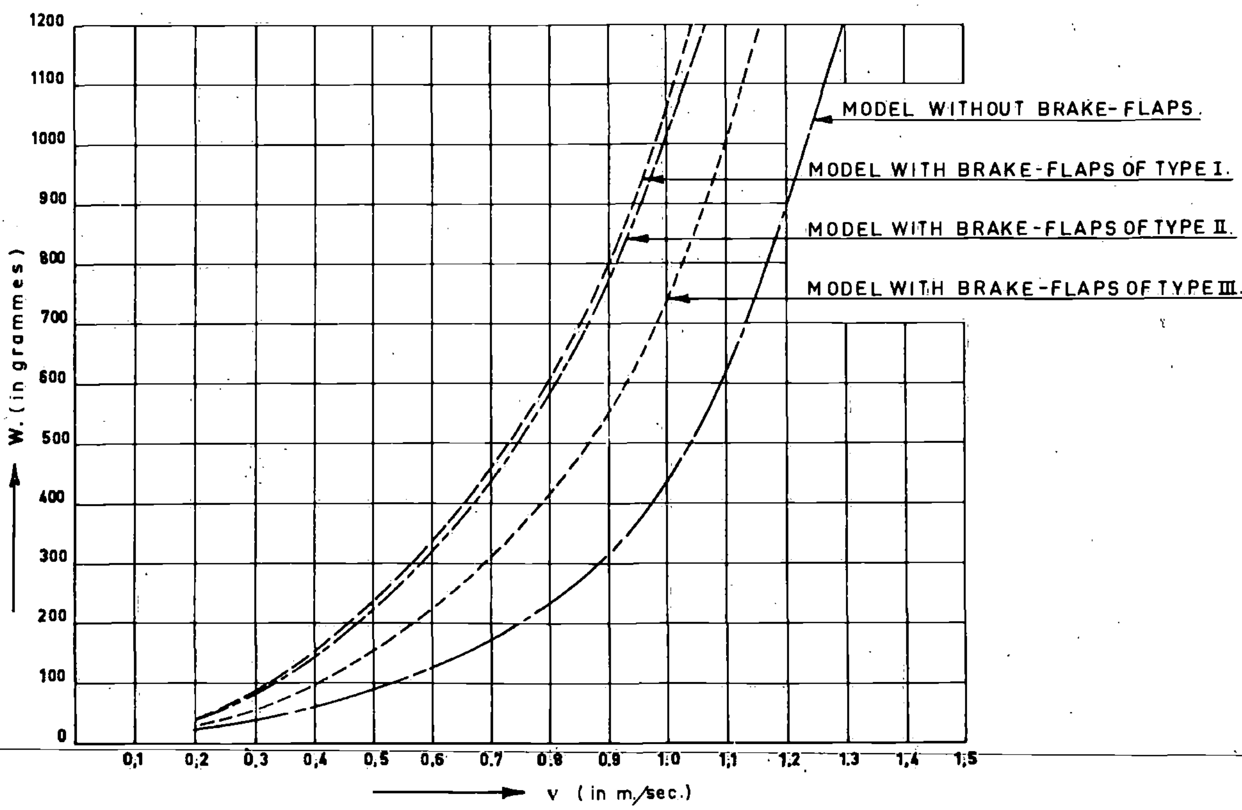


Figure 2. Resistance of the model with and without brake-flaps.

also diminish the pressure on the flaps. The question whether the flaps forward should influence the course-keeping stability was studied separately.

The resistance and stopway-tests were carried out with the flaps glued to the model. The following tests were carried out to determine the stopway:

a. With a model without flaps.

b. With a model with two flaps of type I having the greatest braking surface remaining within the midship-section and following the frame-lines of the forebody on the place indicated in figure 1. Type I has a braking area for the actual ship of 25,672 m² (two flaps) representing 18,45% of the midship-section.

c. With a model with two flaps of type II (figure 1) of the same form as type I but provided with 45 holes of 324 mm. diameter (actual ship) in each flap. The total surface becomes 18,256 m or 13,10% of the midship-section, which means a reduction of surface of 28,85% to type I.

d. With a model with two flaps of reduced area of type III (figure 1) having a total surface of 12,480 m² or 8,96% of the midship-section.

In reality the thickness of the flaps is supposed to be 760 mm. or a little bit less than one frame-spacing. The midship-section of the ship is 139,200 m² (see table I).

Different series of tests were carried out:

a. First series of tests.

The first series of tests was carried out to measure the resistance of the model in the cases a. to d. mentioned above (see figure 2) It is clear that the increase in resistance caused by the flaps is very great indeed. This increase is denominated by:

w = Resistance of ship with flaps - resistance of ship without flaps.

For the three types of flaps this becomes w_I, w_{II} and w_{III} These extra resistances are added to the resistance of the ship en route W' determined by model a. without flaps. The total resistance is expressed as a specific resistance following the practice of the International Towing Tank Conference

$$\xi = \frac{W}{\frac{1}{2}\rho . V^2 . \Omega} \qquad \text{where}$$

$$W = W' \; ; \; W' + w_I \; ; \; W' + w_{II} \quad \text{or} \quad W' + w_{III}$$

respectively.

V = speed in knots
Ω = wetted surface of the ship.

The results are given in figure 3.

During this first series of tests the propeller was kept stopped during the stop manoeuvre. In the results described in section 4 it is shown,

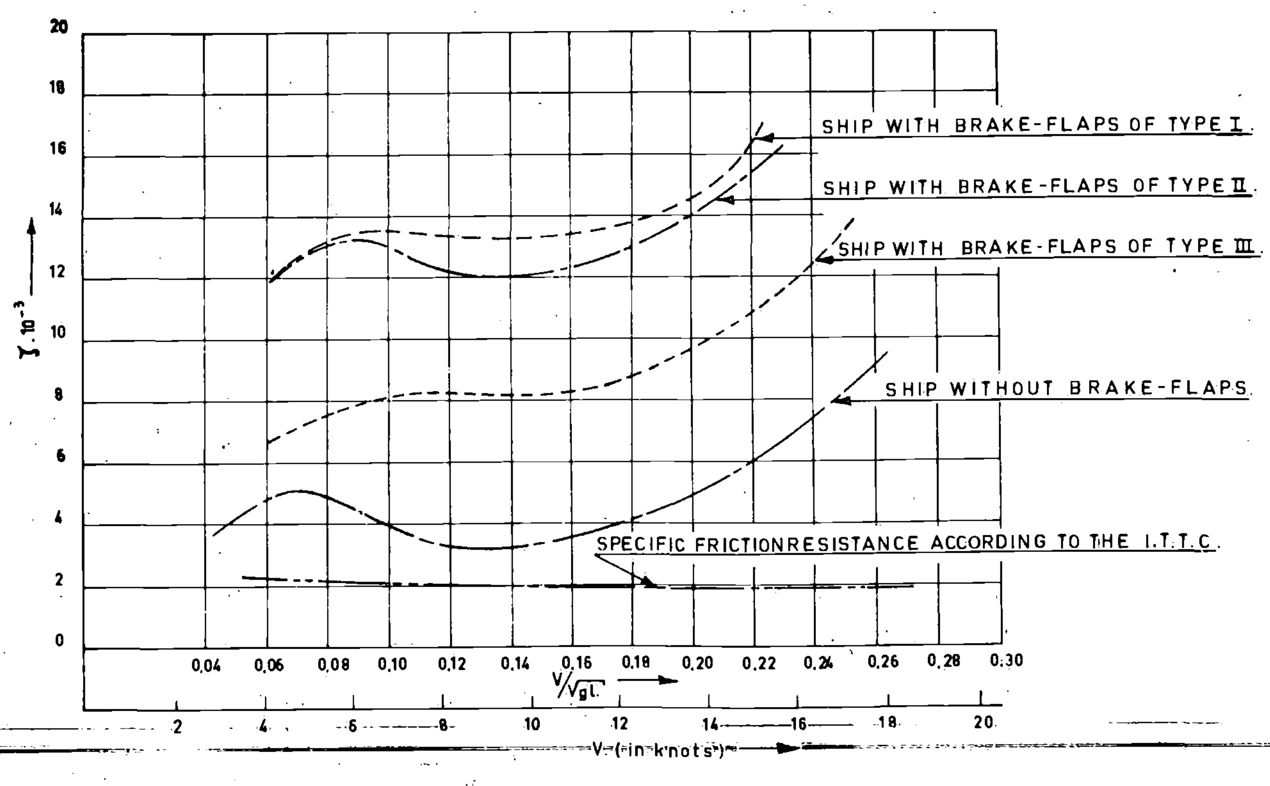


Figure 3. Specific shipresistance extrapolated according to the international towing tank conference.

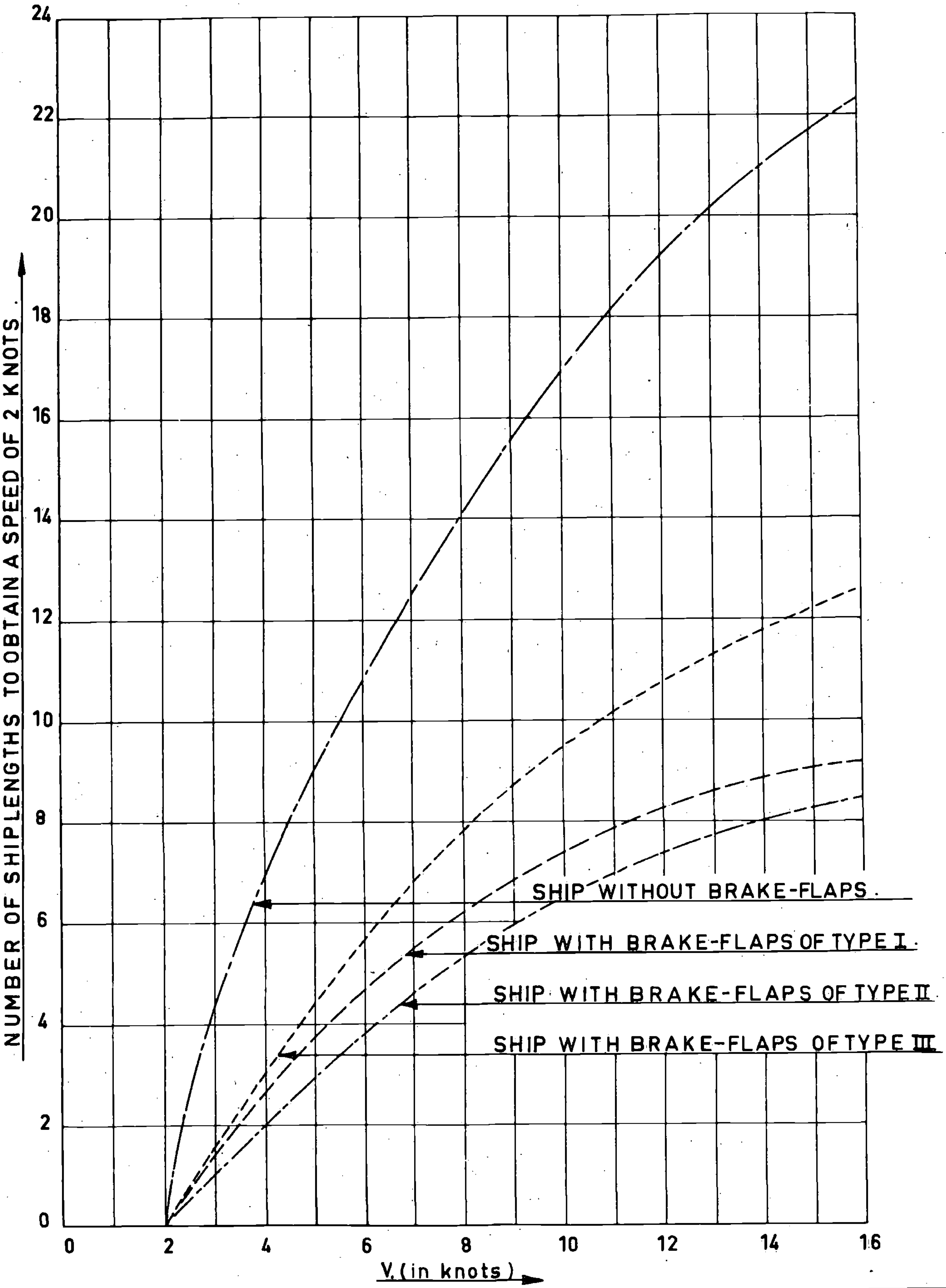


Figure 4. Stopway test-results.

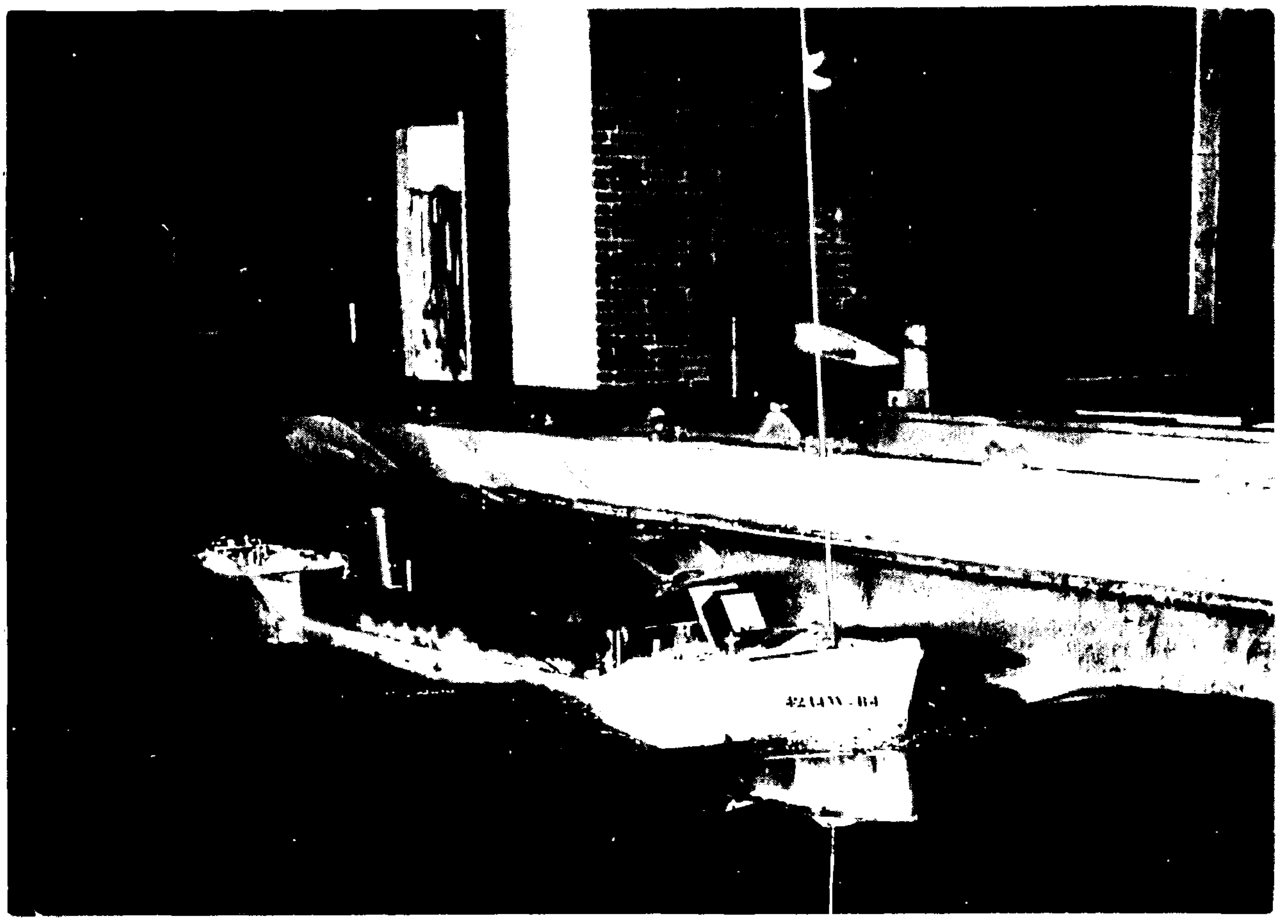


Figure 5.

that the propeller working astern does not assist in the braking of the ship.

One conclusion immediately became apparent: below a speed of 0,3 m/sec (or in reality 2 knots) these tests are no longer reliable. But this is not important as at that speed the ship can easily be stopped by the machine (see section 2). Therefore this bottom limit has been called "stop-speed".

b. Second series of tests.

In the second series of tests the stopway was measured. The model was towed at a certain fixed speed by means of a wire. At that fixed speed the wire was cut and the deceleration was measured as a function of the time.

In this way a deceleration curve was obtained. Integration of this curve gives the relation between time and stopway. From these two parameters the stopway can be calculated and compared with the measured one. If a scale-effect exists it will work out to the advantage of the actual ship, so the curves in figure 4, are a little too pessimistic. As stopway was considered the distance covered from the moment giving the stopping order to the moment that the speed of the vessel was reduced to two knots (see above), this speed being the "stop-speed". In this way the disturbing scale effect has been avoided and in reality this does not influence the practical results.

c. Third series of tests.

In the third series of tests, the influence of the flaps on the course-keeping qualities of the ship was measured. Therefore a self-propelling model was necessary and a big steeringbasin was required. Now this last item was not available and therefore the course-stability was considered sufficient as long as the model in the towing tank continued its straight course after operating the flaps. The least tendency of the model to deviate from the original course was considered as a criterium of non-stability which evidently was a manifest exaggeration for this criterium.

Figure 5 shows the self propelled model constructed to the scale of 1 : 50 (instead of 1 : 54 for the non-propelled model). The model is of the TODD sixty series with a block-coefficient of 0.80. It can navigate for several hours.

The regulation of the number of revolutions of the propeller, the rudderangle and the manoeuvring of the flaps is done by radio. The electric propulsion-motor is fed by a battery of accumulators of ultra-light type.

Measurement apparatus on board and at land

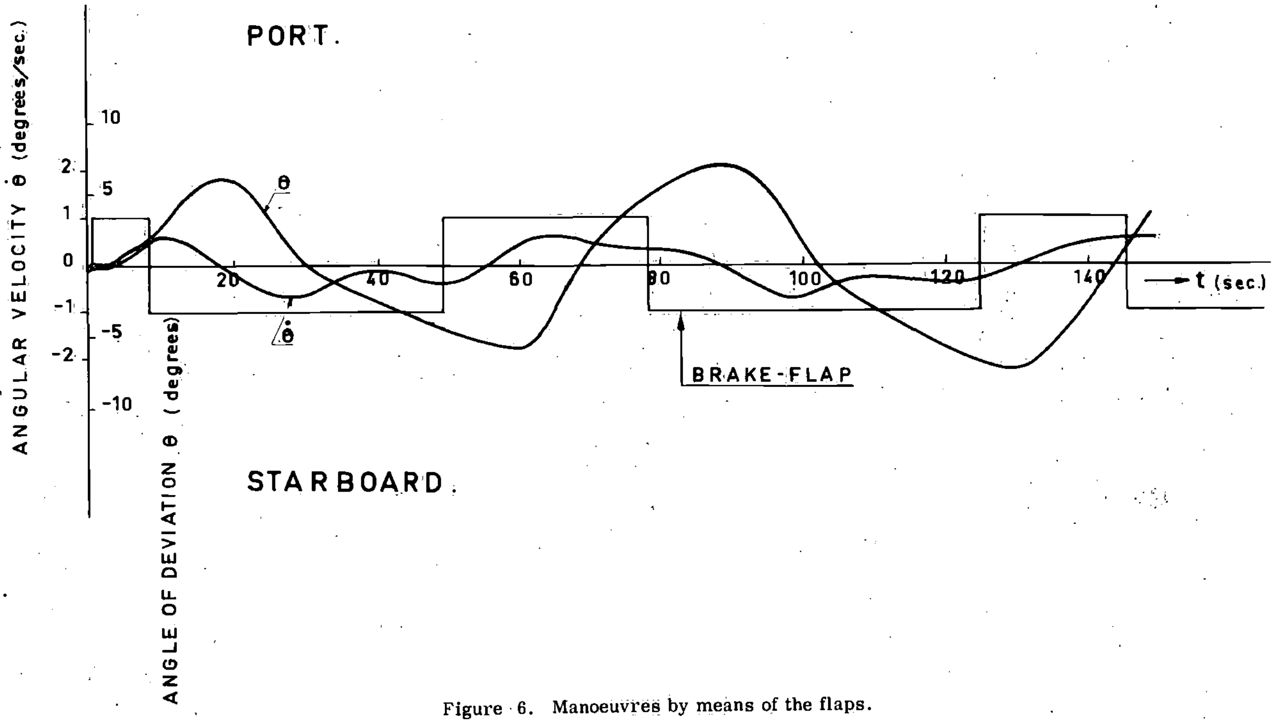


Figure 6. Manoeuvres by means of the flaps.

give all the registered information necessary concerning the number of revolutions and the rudder angle. Furthermore an independent manoeuvring of the braking-flaps is possible. A self-registering apparatus for measuring the angle of deviation and a photographical registrator are installed. These registrators give indications as to the rudderangle δ , the number of revolutions n, the angular speed in the horizontal plane $\dot{\theta}$ and the momentary position of the flaps.

The displacement of the 1/50 scale model is in accordance with the 1/54 scale model. All results given in the figures 2, 3 and 4 remain exact for this other model. As inertial radius $l/4$ is taken, "l" being the length of the model. The selfpropelled model is fitted out with the flap of type II.

During these tests it became clear, that the ship functioning with hydrodynamic brakes remained perfectly course-stable, even more stable than when the flaps were closed. Of course, no disturbing influences from wind, waves and current were measured, but these influences will be relatively small.

The stopway tests while navigating the model in a straight course and a speed of 0,5 m/sec (corresponding to 7 knots of the actual ship) were carried out with the rudder blocked at 0°. At a given moment the stopping manoeuvre began respectively

a. By stopping the propeller only
b. By stopping the propeller and opening the braking flaps.
c. By turning the propeller astern
d. By turning astern and opening the braking flaps.

The results were very conclusive and are given in section 4.

d. The fourth series of tests.

In this series of tests the manoeuvring qualities of the flaps were studied.

One flap, for instance on the port side, was opened and the deviation due to the unsymmetrical resistance of the model was registered. When the model approached the tankwall too closely, the other flap was opened and the first shut and so on, and so forth. The results of these tests are given in figure 6.

The steering effect of the flaps is small. This is comprehensible as, though the load on the flap is great, the momentum arm to the line through the centre of gravity of the model is small. Therefore the steering moment also remains small.

For the compensation of the flap-moment a counter rudder moment had to be given. The rudder angle necessary for this compensation was about 6°.

4. The results of the braking tests by means of transversally moving side-flaps.

The results obtained by the four series of tests described above, gave as a first indication, that the flaps of type II were most efficient. This is very fortunate, as it is clear that the presence of the holes, which considerable diminish the load on the flaps, has a salutary influence on the braking activity of these flaps. An explanation of this salutary influence is given by the extra turbulence-effect on the laminar flow along the ship's hull.

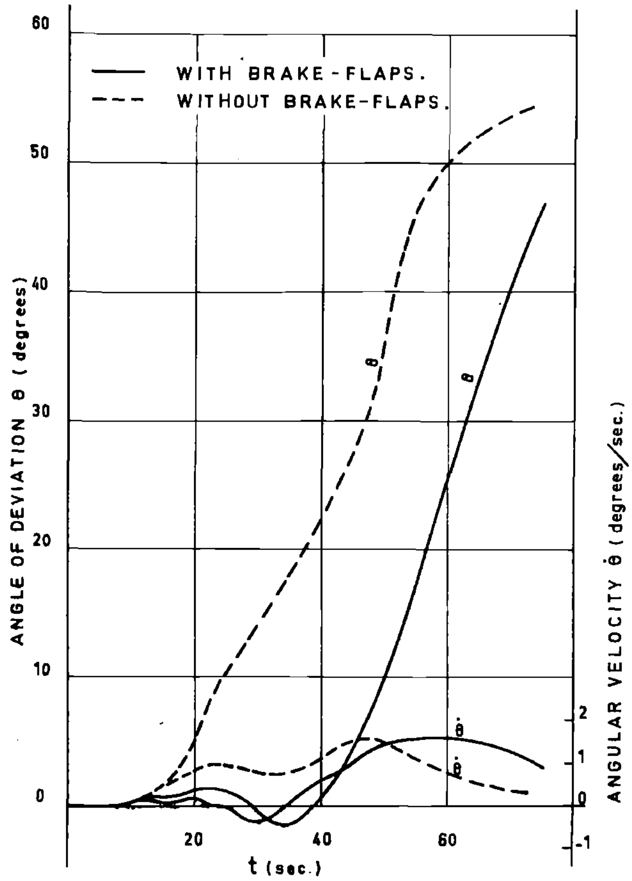


Figure 7. Course stability of the models with propeller turning astern.

Figures 3 and 4 indicate the greater specific resistance and the gain in stopway respectively. This gain in stopway is indeed very remarkable. With flaps of type II pushed out at 16 knots the gain in stopway is 62,4% compared with the non-braked ship. Even at 10 knots the stopway is 61,6% less when braked.

It must be kept in mind, that the "stop-speed" is considered to be a residuary speed of two knots. In reality astern power should be given at about half speed, if astern-power is thought advisable at all. In this respect the A.T.M.A.-paper [1] indicated clearly that a ship with astern-turning propeller loses all governability and the same applies when the flaps are put out as is shown in figure 7.

Therefore it can be said, that flap-braking alone gives the best results and that the moving of the flaps has no influence at all on the course-stability. This seems to apply even if the flaps do not open absolutely simultaneously. The figures 8 and 9 show the model in movement with flaps opened. The effect obtained is clearly visible.

From figure 4 can be seen, that between full speed of the ship at 16 knots and half speed at which one might consider putting the propeller astern, the stopway is already reduced to 60% and the stopping-time too. Generally speaking, it is supposed that the flaps reduce the stopway by at least 50%.

The conclusions as to the qualities of the ship fitted out with flaps may be said to be:

a. Flaps do not disturb the course-keeping qualities. On the contrary it is defendable to suppose even better course-keeping qualities during a stop-manoeuvre.
b. Reduction in the stopway is from about 50% to 60%.
c. Astern turning of the propellers gives bad manoeuvring of ships without flaps as well as of ships with flaps. Astern turning should be used only in the last stage of stopping when the ship has obtained the "stop-speed" (about 2 knots)
d. Flaps assist in the steering of ships, but only to a small extent.

5. The constructional conception of the braking flaps.

If this system of braking large ships by means of flaps is to be carried out, different constructions may be used. However, to obtain an idea about the cost and the weight involved, a type of conception is given here. This conception does not pretend to be a good construction as such.

In the first place, the flaps are supposed to be hydraulically operated. This permits the application of fast moving and large forces.

A schematical design of the flap is given in figure 10. A certain supporting surface must be available to transmit the braking force to the hull.

In order to have an idea of this force, the indications obtained from [1] are used. Maximum thrust of the large French tankers ranged from about 120 tons* to 150 tons. Figure 2 indicates that the resistance of the ship is increased by about 130% to 150%. This means a maximum load per flap of about

$$\frac{1,5 \times 150}{2} = 112 \text{ tons}$$

If this load is thought concentrated in point P (see figure 10) the supporting surface takes 112 tons of load and a moment with respect to the most inner point of support of 112 x 3,9 = 436 ton-metres. This calculation supposes that the spindle of the flap M only bears the weight of the construction, which is not exactly true and is therefore on the pessimistic side. In any case it may be said, that the pressures per cm^2 on the bearing surface remain within reasonable proportions.

The edge J glides between two supporting ways, well lubricated and with a small tolerance. The bearing edges are aft at the hull-side and forward at the innerside. As an approximation a linear distribution over the edge is admitted (see figure 10, sketch).

If this specific load per unit of length is called p (in KG/cm) at the end of the hull side, the load in every point of the edge is p'

* Tons are metric tons of 1000 KG = 0,985 British tons

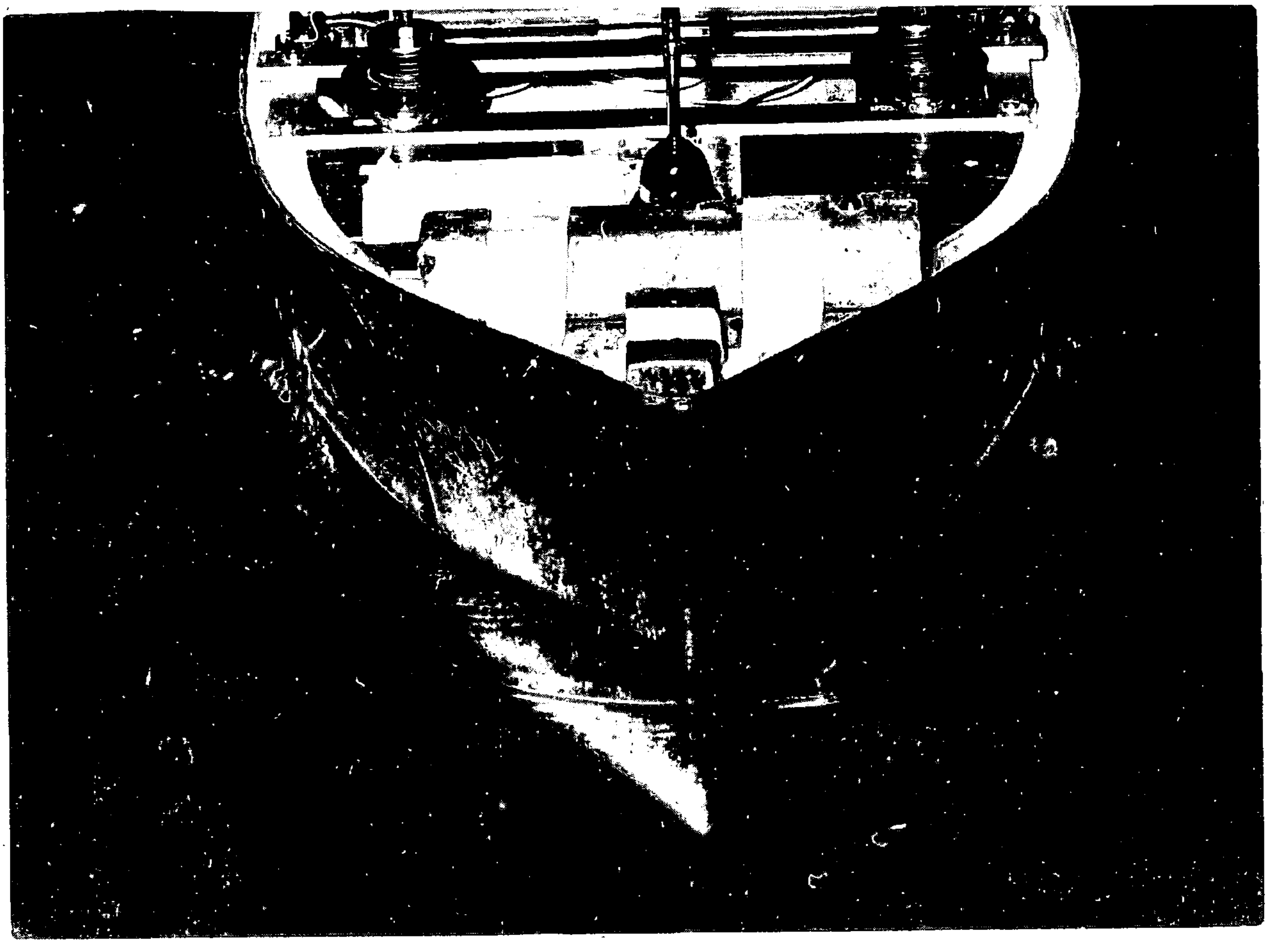


Figure 8.

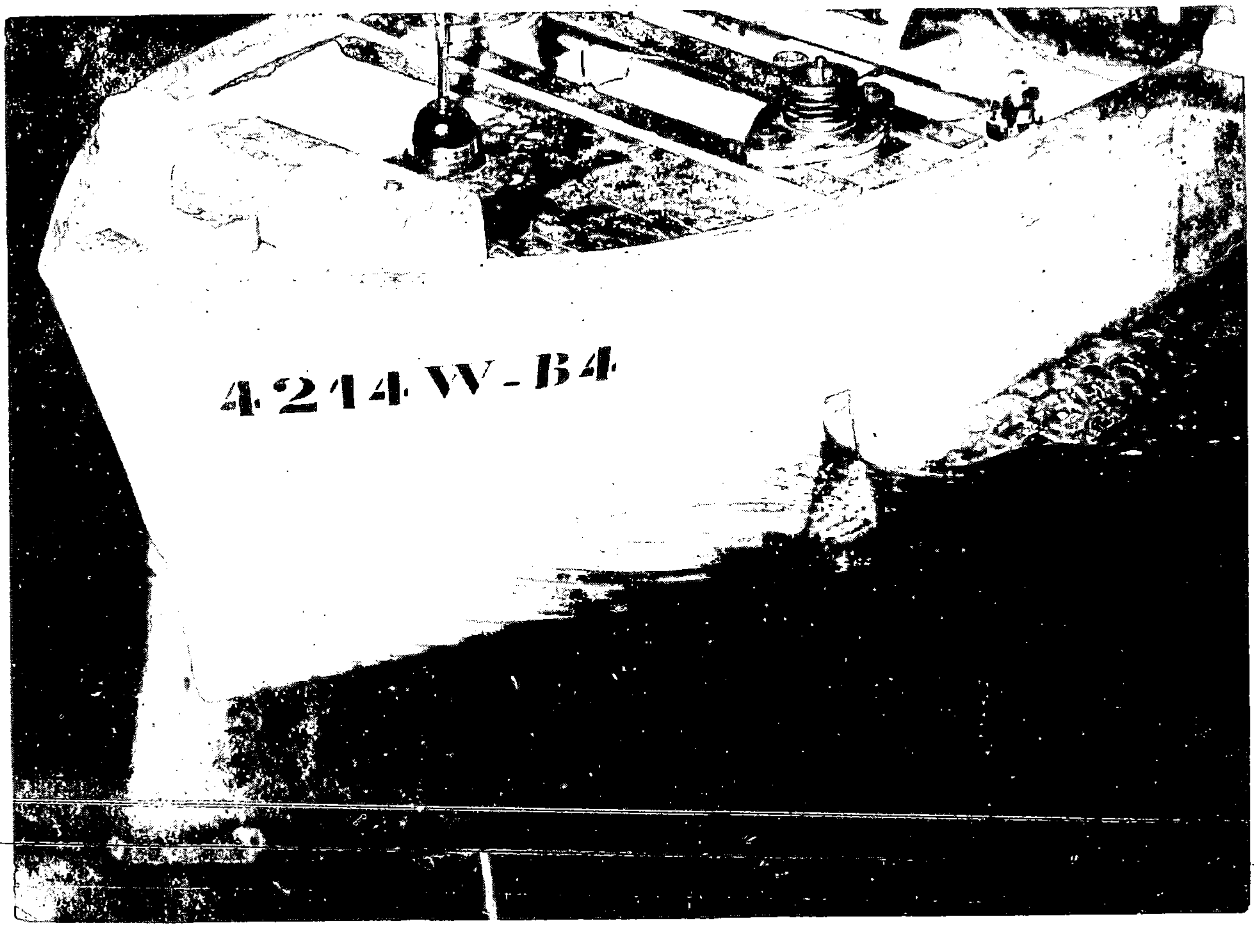


Figure 9.

$p' = p - k\,(146 + x)$ where

$$k = \left(p - \frac{112000}{292}\right) \cdot \frac{1}{146} = \frac{p - 384}{146}$$

The total moment of $\Sigma p'$ with regard to the most inner point of the flap becomes:

$$M_{cm} = 112000 \times 146 + (p - 384)\ 292 \times \frac{292}{3} =$$

$$164.10^5 + 14211\ (p - 384)\ \text{KG.cm}$$

This moment must be equal to 436.000 KG.m therefore

$$p - 384 = \frac{436.10^5 - 164.10^5}{14211} = 1920\ \text{KG/cm}$$

and thus $p = 2300$ KG/cm.

If the width of the supporting edge is 10 cm. the maximal pressure on the hull extremity will be only 230 KG/cm².

To overcome the friction when pushing out the flap, the maximum effort in the most unfavourable conditions will be about

1/3 x 230 x 292 x 10 = 225000 KG = 225 tons.

To this the weightcomponent of the flap must be added for shutting it and keeping it shut. Therefore a double working 300 tons hydraulic ram per flap will be amply sufficient for manipulation.

Supposing a hydraulic pressure of 60 atm. in this ram, a construction with a factor of safety of about ten may be easily obtained.

Each flap consists of two plate-surfaces about 760 mm. apart. The plates are of heavy construction and are cut in such a way that the circumference has the form of the hull-section (B), the inferior edge (J) and the interior edge (C) (see figure 10). As the 45 holes in the plates are connected to each other by tubelures, the whole will be of very robust construction. It is always possible to reinforce the part of the flap inside the ship in the same way.

When the flaps are retracted they enter a niche, which is part of the main hull structure (see figures 1 and 10). To ensure, that these niches do not harm the longitudinal strength of the ship, they must be of robust construction and have longitudinals in connection with the main structure, though the longitudinal bending moments in this part of the ship will be small. Care must be taken, that the notch effect due to these niches is neutralised and that shearing forces in the fore part of the ship are taken over by other scantlings.

These niches must be of absolutely watertight construction. Between them there will be a watertight compartment containing the rams, an oiltank, two hydraulic pumps having an oil-

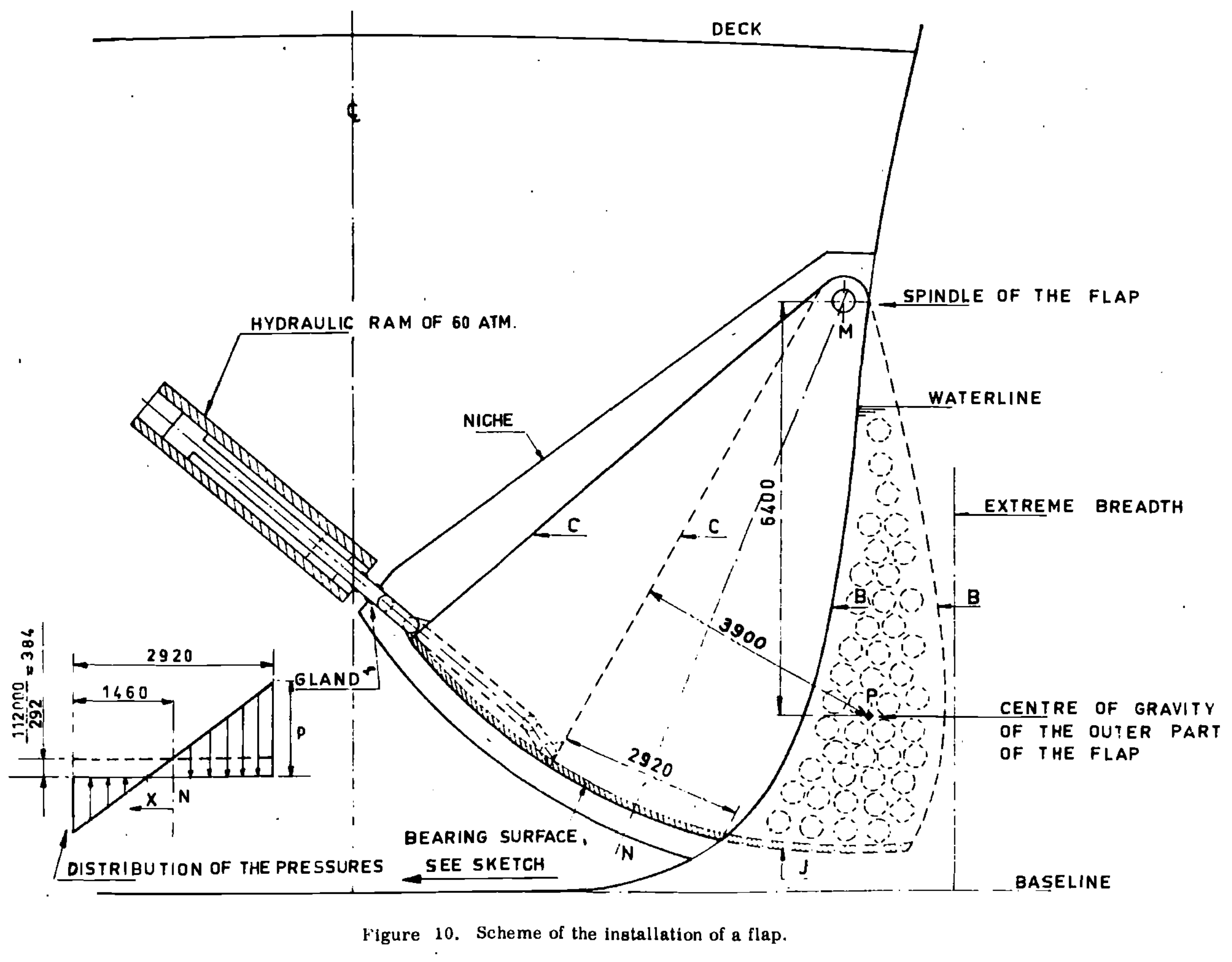


Figure 10. Scheme of the installation of a flap.

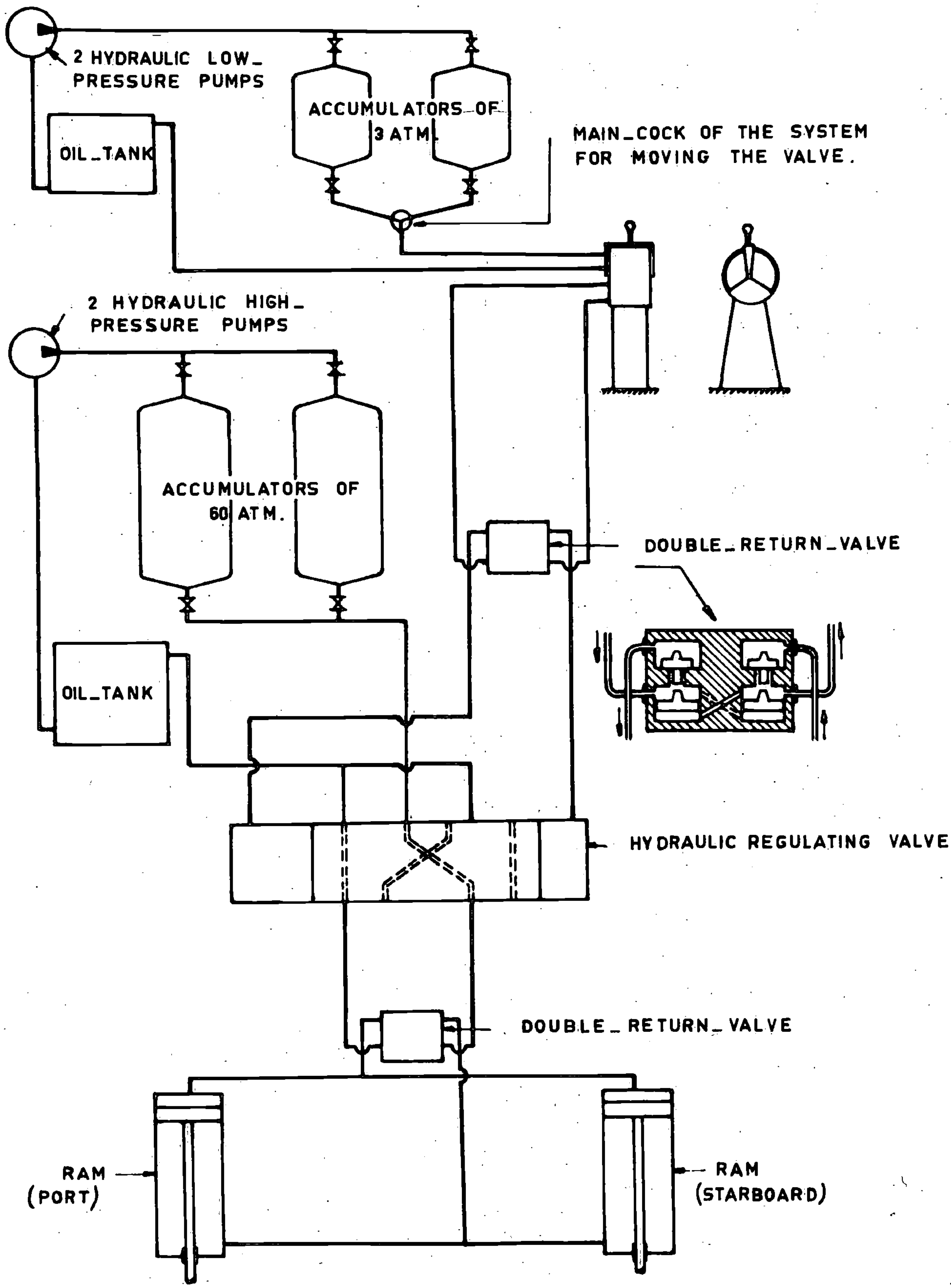


Figure 11. Scheme of the two hydraulic systems.

pressure of 60 atm. and two hydraulic accumulators of 60 atm., one of which is a spare accumulator. This small hydraulic engineroom can be entered from the strengthdeck. Only the ram rods pass through the watertight bulkheads.

The rams are controlled by means of a hydraulic remote-control system, working at low pressure (3 atm.) from the wheel house. This low pressure system consists of an oiltank, two small pumps and two low-pressure accumulators of 3 atm. (see figure 11). Here also, one accumulator serves as a spare. A main-cock, is provided which can shut off the whole system, so that the ship may navigate as any other normal ship. During this period the flaps are maintained in the retracted position by the 60 atm. pressure-oil system. The manoeuvring handles of the valves may be combined with the ship's telegraph so as the ensure a fool-proof working of the flaps.

All these ideas are only approximative and have no precise values. They are uttered here only as possible constructive realisations, showing that the constructional part of the problem of installing braking flaps is by no means an utopy.

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M = engineering department C = corrosion-antifouling department S = shipbuilding department

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NEDERLANDS SCHEEPSSTUDIECENTRUM TNO

NETHERLANDS SHIP RESEARCH CENTRE TNO
SHIPBUILDING DEPARTMENT LEEGHWATERSTRAAT 5, DELFT

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THE BRAKING OF LARGE VESSELS II

(HET REMMEN VAN GROTE SCHEPEN II)

by

PROF. IR. H. E. JAEGER
(Delft University of Technology)

in collaboration with

M. JOURDAIN
(French Shipbuilding Research Institute)

Issued by the Council

VOORWOORD

Dit rapport is een vervolg van rapport no. 53 S van de hand van dezelfde schrijver, dat in augustus 1963 werd gepubliceerd.

Na die datum zijn nog vele proeven en studies uitgevoerd door het „Institut de Recherches de la Construction Navale" (IRCN) in Parijs en het Nederlands Scheepsstudiecentrum TNO te Delft, waarvan een aantal in nauwe samenwerking tussen de twee instituten.

Als resultaat van deze proeven en studies wordt meer informatie gepresenteerd met betrekking tot het probleem van het remmen van grote schepen.

Dit rapport verklaart hoofdzakelijk de resultaten die verkregen werden uit proeven in een sleeptank van het Nederlandsch Scheepsbouwkundig Proefstation te Wageningen in 1965 en 1966, maar enige gedeelten van dit rapport zijn tot stand gekomen in nauwe samenwerking met de heer M. Jourdain, Directeur van het IRCN, die de resultaten van de proeven op zee verstrekte en hielp bij het opstellen van het programma voor de modelproeven in 1965. Deze medewerking zij hier met dank vermeld.

HET NEDERLANDS SCHEEPSSTUDIECENTRUM TNO

PREFACE

This report is a sequel of report no. 53 S, published in August 1963 by the same author.

Since then many trials and studies have been executed by the French Shipbuilding Research Institute („Institut de Recherches de la Construction Navale" or IRCN) at Paris and the Netherlands Ship Research Centre TNO at Delft, several of which in close collaboration between the two Institutes.

As a result of these tests and studies more information is given concerning the problem of the braking of large vessels.

This report explains principally the results obtained during towing-tank tests executed at the Netherlands Ship Model Basin at Wageningen in 1965 and 1966, but certain parts of this report are established in close collaboration with Mr. M. Jourdain, Director of the IRCN, who has provided the sea trial results and helped to draw up the first trial program for the model tests of 1965. This kind cooperation is gratefully acknowledged here.

THE NETHERLANDS SHIP RESEARCH CENTRE TNO

CONTENTS

LIST OF SYMBOLS

A	Disc area of propeller
A_0	Developed blade area of propeller
C_B	Block coefficient
C_M	Midship section coefficient
C_P	Prismatic coefficient
D	Stopping distance also: propeller diameter
D_{16}	Stopping distance for a basic speed of 16 knots
H	Pitch of propeller
K	Braking force
L_{pp}	Length between perpendiculars
M	Mass of the ship
Q	Torque
R_a	Friction correction for hydrodynamic resistance
R_n	Nominal resistance of underwater hull
S	Distance covered
T	Resultant braking forces taken up by propeller
V	Speed of ship
W	Complementary resistance of underwater hull
a	Acceleration
d	Diameter of boss of propeller
m	Added mass of entrained water
n	Number of revolutions per minute
t	Time
Δ	Displacement (in metric tons)
ΔC_f	Roughness allowance
ΔR	Correction(s) for hydrodynamic resistance

THE BRAKING OF LARGE VESSELS II

by

PROF. IR. H. E. JAEGER

in collaboration with

M. JOURDAIN

Summary

In the first part of this report a review is presented of full scale stopping trials with a number of single screw turbine tankers, the results of these trials are analysed and some preliminary conclusions are drawn.
The second part treats the extensive model tests that were performed with a model of one of the tankers to study the various factors that influence the head reach and to obtain a better correlation between full scale and model experiments. The quasi-stationary approach that has been applied is explained.
Finally the general conclusions and some questions that remain to be solved are discussed.

1 Introduction

This report is a sequel to report No. 53 S [1] of August 1963, issued by the Council of the "Netherlands Research Centre TNO for Shipbuilding and Navigation" (Now the "Netherlands Ship Research Centre TNO") and again treats the difficulties arising out of the ever increasing size of modern bulk-carriers and supertankers during the braking and stopping of these large vessels.

The "crash-stop" test has become more and more important as ships grow in size and it has been generally recognized now, that the head-reach or stopways of such large ships have become quite impractically long.

One of the goals of this report is to envisage the braking qualities, which future tankers should possess.

A first point to mention is, that for all big tankers the service speed is about the same viz. 16 knots and therefore the speed gradient, $V/\sqrt{L}$, differs only slightly and descends with growing size. However when entering port and during manoeuvring the advance speed will be lower and this, in practice, is often the initial speed of the "crash-stop" manoeuvres.

Elimination of the kinetic energy due to the mass and speed of the ship is accomplished by means of the hydrodynamic hull resistance and propeller thrust under backing conditions ("crash-stopping"), insofar as wind and sea effect and exterior forces acting on the ship are ignored. The hydrodynamic phenomena also depend on the depth of water, where this has its ratio to the ship's draft close to unity. If the efficiency scattering while going astern is neglected, it may be admitted that the propeller thrust under backing conditions is proportional to the available backing power delivered by the screw propeller. This backing power depends of course on the engine power installed and on the type of machinery.

But in going astern another element must be taken into account: the time in which the available backing power becomes effective. This time depends both on the type of machinery and on the way in which it is manoeuvred.

If this simple scheme is accepted as an approximation for "crash-stopping", an elementary calculation shows that, on the basis of a given initial speed, the stopping time and the head-reach are proportional to the length, for ships of the same type.

"Crash-stopping" means turning astern with the propeller, and the "Institut de Recherches de la Construction Navale" (IRCN) in Paris therefore, took in hand, since 1960, a series of stopping and manoeuvring tests during the sea-trials of large tankers (from 50,000 tons d.w. to 110,000 tons d.w.) with the intention of studying stopping conditions in detail and analysing the consequences.

As during 1961 the "Netherlands Ship Research Centre TNO" was studying the same problem, both the French and the Dutch Research Institutes have worked out the problem together for many years and have exchanged their results in close collaboration. The most important of these were published in the Transactions of the "Association Technique, Maritime et Aéronautique" (ATMA) in Paris [2, 3, 4, 5, 6], and a general review of this combined work was presented to the "Society of Naval Architects and Marine Engineers" (SNAME) as a "Diamond Jubilee" Paper in June 1968 at New York [7].

As the publications [5] and [6] were especially the object of studies carried out by the "Netherlands Ship Research Centre TNO" (in collaboration with the

IRCN) in 1965 and 1966 at Wageningen, they explain this second report on the matter, as follow-up of report No. 53 S of 1963.

The report also provides evidence of the influence of the process of manoeuvring the engine. On the other hand, the factors whose influence is difficult to estimate theoretically – that is to say, the displacement in ballast, the initial speed, and the depth of water – still have to be investigated.

The first part (Section 2) gives a general review of the French sea-trials, executed by the IRCN.

The second part (Section 3) of this report contains the part financed by the Netherlands Ship Research Centre as far as the determination of the correlation between the results of model-tests and real sea-trials is concerned.

Model-tests were carried out in the Netherlands Ship Model Basin in Wageningen in 1965 and 1966 on one of the French tankers, which had been subject of extensive braking and stopping trials at sea.

2 Sea trials

2.1 *General*

As already mentioned, the IRCN in Paris took in hand the trials to be held at sea. Several tankers were tried and by courtesy of the IRCN a full report on these trials can be given in this report. They were carried out under favorable conditions so that different aspects of stopping characteristics could be studied. All the ships tried were single-screw turbine tankers, varying in deadweight from 50,000 to 100,000 tons (tons of 1000 kgf). Most trials were made in the loaded condition. Three of the ships were submitted to more extensive trials:

Tanker A (length 217 m) was tried at three displacements (66,000, 49,000 and 38,000 tons of 1000 kgf). The initial speed and the type of manoeuvres were varied. One trial, the so-called "natural" trial, was executed by stopping the screw propeller in order to isolate the influence of the hydrodynamic resistance.

Tanker D (length 220 m) was tried at a constant displacement of 67,000 tons of 1000 kgf in a series of special trials in order to study the influence of rudder manoeuvring.

Tanker E (length 238 m) was tried at a constant displacement of 89,000 tons of 1000 kgf in a series of trials designed to study the influence of the initial speed.

A few other tankers were tried, only normal crash-stop tests being executed.

With ships A, D and E, the headreach was obtained by simultaneously measuring the headings indicated by the gyrocompass and the distance covered at sea, the latter by means of a Pitot log during that time and as

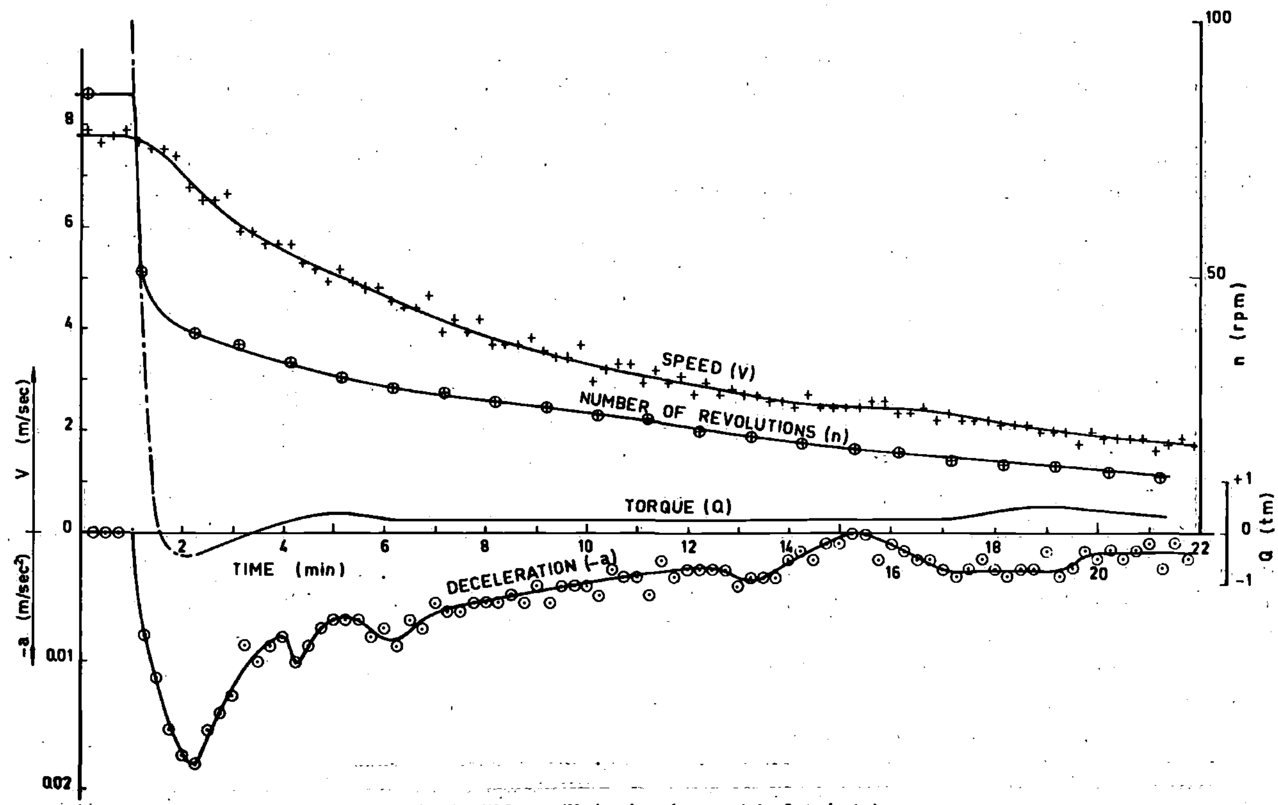

Fig. 1. "Natural" slowing-down trial of tanker A

a function of it. Certain ships had their stopping way measured by radar or by Decca Navigator.

In ships A, D and C, the latter being a tanker of 246.5 m length and 95,000 tons of 1000 kgf displacement, strain-gages were installed to record the torque and the thrust. The number of revolutions of the screw-propeller and the characteristic moments of changing the manoeuvring of the machinery and the steam pressure were recorded also. All the detailed results of these trials and experiments have been published in ATMA publications [2–6].

2.2 *"Natural" trial of tanker A*

As the command "going astern" cannot be followed up immediately, there is always an initial phase, more or less long, during which the slackening of the speed is caused by the hull resistance only. This way of slackening will be called "natural". Therefore, as a basis of comparison, the natural slowing down of the ship, with cut-off steam and propeller turning slack, was measured in so called "natural" trials.

Figure 1 gives the results of these trials carried out with tanker A in ballast. It is clear from this figure that during the first half-minute the number of revolutions and the torque decrease very rapidly; but, while the number of revolutions decreases more slowly after it has reached about half its initial value, the torque remains thereafter practically constant and practically zero. Furthermore, other trials were carried out from which it appeared that the evolution of the thrust is analogous to that of the torque, except that the final values are slightly negative.

Gradually, the number of revolutions decreases very slowly and uniformly and the torque and the thrust remain constant. During these "natural" trials, the speed decreases sharply during the first minute, the

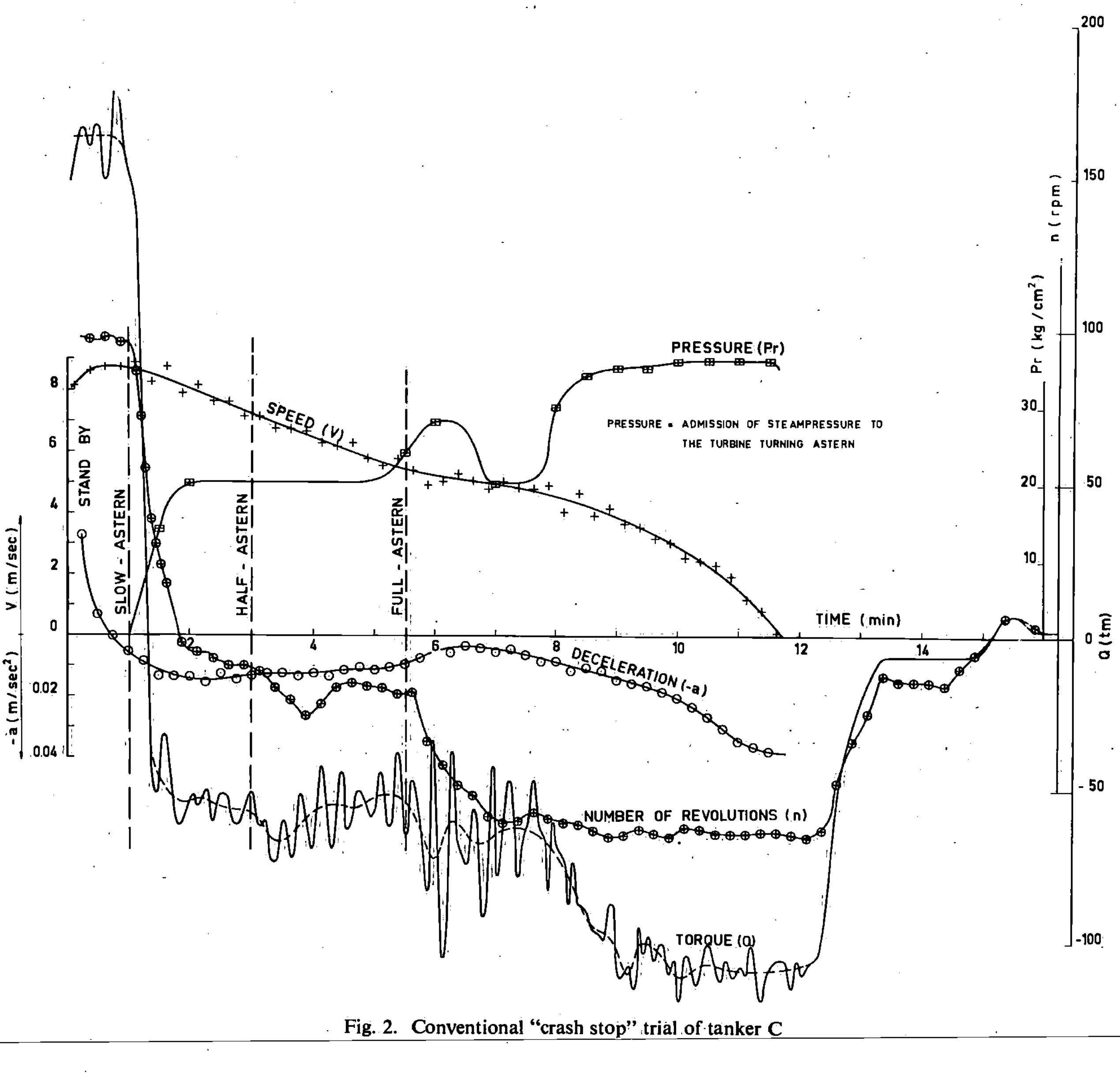

Fig. 2. Conventional "crash stop" trial of tanker C

deceleration reaching a maximum of 0.018 m/sec^2 at the end of that period. At the end of the trial the deceleration was low and the slow evolution of the speed was analogous to that of the number of revolutions. On the other hand, the transitional period for the speed was much longer and more progressive than for the number of revolutions.

There are all kinds of fluctuations in the deceleration of the vessels, caused probably by the wave system generated by the ship itself [8]. It will be seen from figure 1 that the speed decreases during the first minute by 10 percent. In two minutes it is reduced by 20 percent, in three by 30 percent, in five by 40 percent, and in ten by about 60 percent. A further deduction from the test results is that merely through hydrodynamic resistance, the vessels lose about one fifth of their kinetic energy in one minute, about one third in two minutes, and about half in three minutes.

2.3 *Conventional trials*

In figure 2 the results of a measured conventional trial with tanker C are given. This trial was carried out by manoeuvring without excessive hurry. The thrust indications, not measured at these trials, are deduced from analogous trials conducted with other ships.

Compared with the "natural" trial, the essential difference is that the direction of rotation of the propeller shaft is changed in about one minute, the torque becoming rapidly negative at about one third of the initial positive value. Notwithstanding this, the maximum initial deceleration remains of the same order as with the "natural" trial, figure 2 representing a good mean value.

After its first maximum, the deceleration remains nearly constant after the first two minutes, though the number of revolutions and the applied torque are very important and change materially.

As during this period the evoutlion of the speed is quite in conformity with that of the "natural" trial, the conclusion must be that the developed torque to the propeller has hardly any effect as regards the braking of the vessel.

There are great fluctuations in this torque, and the same applies to the thrust. These fluctuations are less marked if the number of revolutions remains constant and they increase in importance when an attempt is made to modify the number of revolutions. The irregular functioning of the propeller in this respect has been studied in [9]. Though this irregular functioning

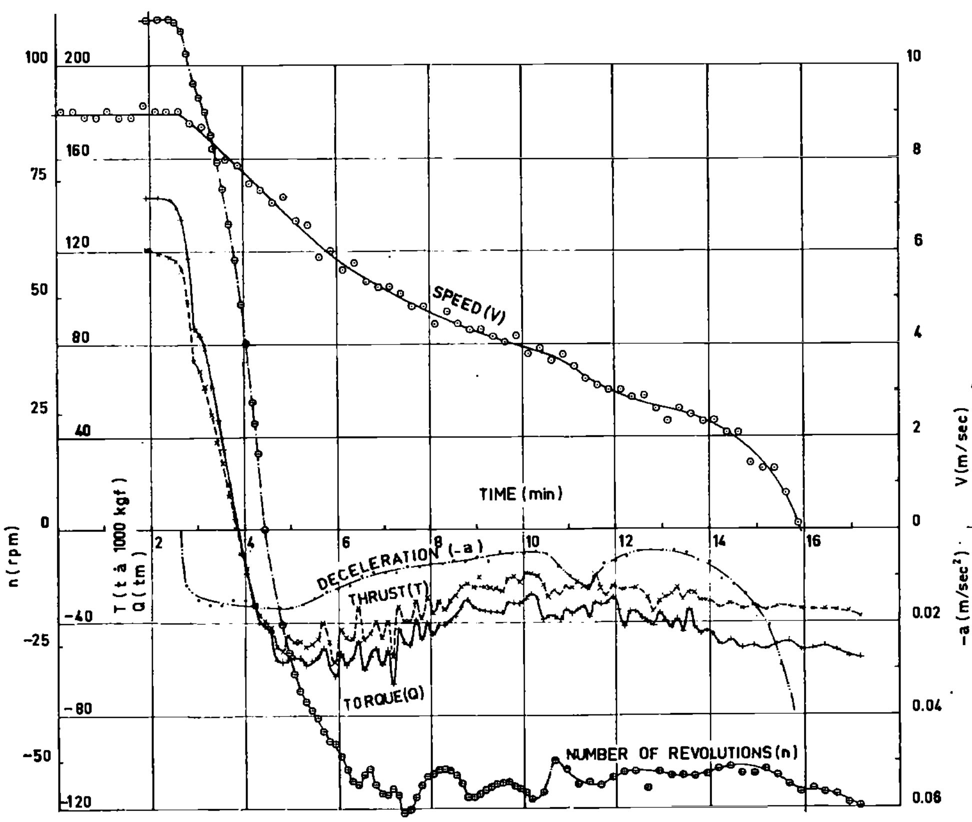

Fig. 3. Conventional "crash stop" trial of tanker D

induced intolerable vibrations, it may be asked whether, if they are kept within bearable limits, there is not an advantage in braking more quickly. The answer to this suggestion is given in figure 3.

This figure relates to a trial with tanker D in load; the number of revolutions decreases constantly, without variations, the evolution of the speed being of the same type as in figure 2, and the initial deceleration decreases for 8 min, while the revolutions then increase to a constant rate. The moment the number of reverse revolutions has reached a maximum, the torque, the thrust, and the deceleration decrease. After seven minutes, figure 2, or about eight minutes, figure 3, the number of revolutions remaining constant, the disturbance dies off – the fluctuations disappear, the torque and the thrust increase till they reach the normal level, the slowing-down increases progressively, and the speed is reduced towards zero.

These latter phenomena well illustrate the final period of stopping. During this final period only, the braking of the vessel depends directly on the applied astern power of the propeller.

It was tried to arrive at a simple theoretical approach to describe these trials. The vessel only was considered, not taking into account the entrained mass of water. Therefore the mass M is constant and sharply defined. The measurement of the speed V as a function of the time makes it possible to express the slowing down as $-\mathrm{d}V/\mathrm{d}t$, and the braking force K is given by

$$K = -\frac{M\mathrm{d}V}{\mathrm{d}t}$$

(Note: This definition is not the same as the one given in section 3.5, concerning the analysis of the model tests for the study of the correlation between model and ship by means of the quasi-stationary method.)

This braking force K is the resultant force parallel to the axis of the ship and consists of the total of all forces of whatever kind exercised by the water on the propeller and the underwater hull. This force K is arbitrarily broken down as follows

$$K = T + R_n + W$$

where

T = resultant forces taken up by propeller,
R_n = nominal resistance of underwater hull,
W = complementary resistance of underwater hull.

These symbols are defined as follows: T is at every moment equal to the thrust transmitted through the thrust block to the ship; this thrust can be measured and therefore is well defined. R_n is defined at every determined speed as the force necessary to maintain the vessel at that speed. It is the normal hydrodynamic resistance augmented by the thrust deduction forces. By measuring the thrust at different stationary speeds, it would be possible to determine this nominal resistance. As this was not done in these trials, it is assumed that the thrust is proportional to V^2. The value measured at the initial speed is chosen as a basis.

$W = K - T - R_n$ and represents globally the forces of every kind working on the underwater hull subject to the fluctuating character of the flow.

Figure 4 is deduced from figure 3. It represents the break-down of this normal trial into braking force $K = -M(\mathrm{d}V/\mathrm{d}t)$, thrust T, nominal resistance R_n, and the sum of $T + R_n$. The difference between K and $T + R_n$ gives an estimation of the complementary resistance W.

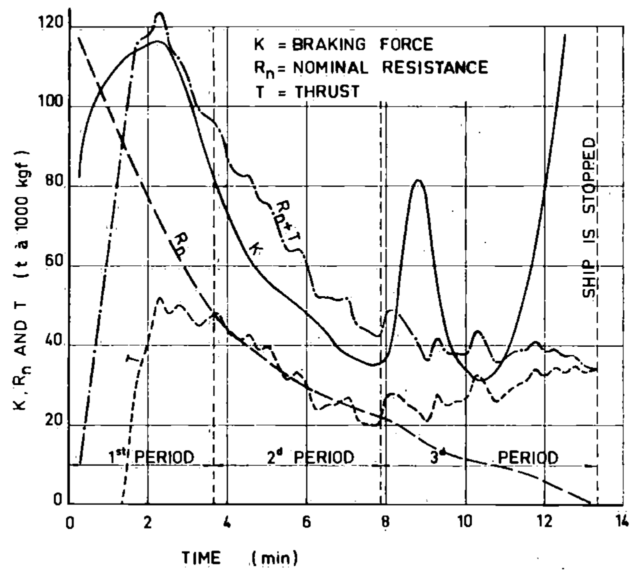

Fig. 4. Analysis of conventional trial from figure 3

Figure 4 makes it possible to subdivide schematically the stopping manoeuvre into three parts, indicated in that figure, giving three successive comparable periods:

During the first period (figure 4) the thrust T is reversed and then attains progressively a value of the order of importance of the decreasing nominal resistance R_n. The complementary resistance $(K - T - R_n) = W$ is large.

During the second period, when T has become as large as R_n, both forces decrease more or less together, as if some interaction between them makes them follow the same law. The complementary resistance W is small and negative.

During the third period, the nominal resistance having become smaller and smaller, the thrust becomes larger than this resistance and grows till it reaches the normal thrust for the number of revolutions astern. The period ends when $R_n + T = T$ or $R_n = 0$. During this end period the complementary resistance is uncertain and feeble. The thrust astern determines the braking.

During the first period, the speed decreases by one third over a distance of about 40 percent of the head reach. At the end of the second period, about 80% of the stopway has been covered and the speed is reduced to about one half.

At all normal trials it was found that these values differed little, while the parallelism between thrust and resistance during the second period was always present. This remarkable particularity is probably related to the effect of the suction of atmospheric air in the propeller (ventilation) as explained by Bindel and Garguet [9].

Thus the slight influence of the astern power installed in a ship may be explained. In fact, this power cannot be important during the first period, when it grows slowly and progressively, nor during the second period, when the growing of the thrust is braked off. It can only become fully effective in the third period, but this period only consists of 20 percent of the stopway and every gain will be only a small part of the whole way. Therefore, one can say that the hydrodynamic stopway of a ship depends on the design of the submerged part of the hull and *not* on the power installed in the machinery.

Two small observations must be made: First of all, it is essential, when stopping a ship with its machinery, not to lose time between the order "full astern" and the execution of that order. Otherwise one prolongs the first period unnecessarily. This supplementary "dead time" is an explanation of the great dispersion observed in "crash-stop" trials at sea.

Secondly, as mentioned in the foregoing, at the beginning of the third period the speed is only about half full speed (vessel in harbour, in fog, and so on), the first and second periods are appreciably reduced and the thrust available may be used more efficiently and more quickly. As seen in figure 5 and as discussed in [5], the stopway curves, when starting from different initial speeds, cannot be superimposed. On the other hand, the influence of the initial speed on the stopping time and the stopway is considerable, figure 6.

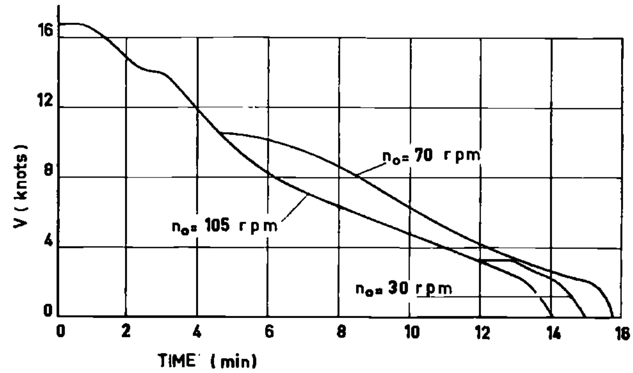

Fig. 5. Tanker E, comparison from three trials

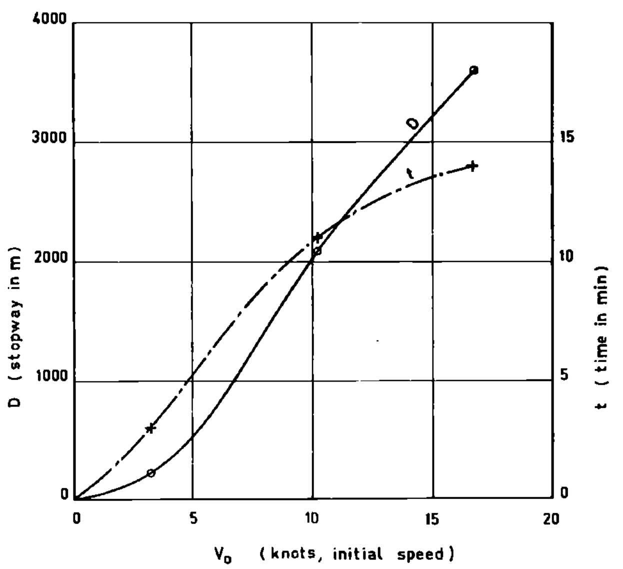

Fig. 6. Tanker E, influence of initial speed on stopping-time and stopway

Table I. Summary of conventional trials

Tanker	Δ tons	L_{pp} m	V_0 kn	t sec	D m	D_{16} m	$-a$ m/sec^2	D_{16}/L	Track	Observations
A	38,050	217	15.3	585	2,155	2,250	0.013	10	quarter of a circle	
A	66,370	217	17.3	735	2,930	2,710	0.012	12.5	half-circle	
A′	62,500	217	16.8	660	2,575	2,400	0.013	11	half-circle	
A″	35,000	217	18	490	2,310	2,050	0.019	9.5	quarter of a circle	windforce 7, rough sea
C	95,400	246.5	17	645	3,200	3,000	0.013	12	quarter of a circle	
C′	95,400	246.5	15.5	770	3,070	3,170	0.010	13	slight curvature	windforce 5, rough sea
D	67,250	222	17.25	810	3,590	3,330	0.011	15	slight S-curve	windforce 4, rough sea
E	88,835	238	16.5	815	3,300	3,200	0.010	13.5	half-circle	
F	62,560	215	16.5	465	2,150	2,080	0.018	9.5	half-circle	
G	103,780	258.5	16.6	960	3,300	3,180	0.009	12.5	half-circle	windforce 4, rough sea

Δ = Displacement (tons of 1000 kgf)
V_0 = initial speed
t = stopping time

D = stopway
D_{16} = stopway corrected for a basic speed of 16 knots

A′ and A″ are sister-ships of A
C′ is a sister-ship of C

It may be deduced from this last figure, that, with initial speeds of the order of magnitude of the full service speed, the stopway is approximately proportional to that initial speed. If this is so, then the normal trials executed at different speeds may be brought back to a basic speed, as is done in Table I, where this basic speed is taken as 16 knots. In this table the stopping times and stopways are indicated, not taking into account the aforementioned "dead time".

The stopway D is measured along the track covered by the vessel. The stopway D_{16} is the length of the path as corrected for an initial speed of 16 knots.

The modalities of the manoeuvring of the engine are not taken into account, as it is already proved that the astern-power used has not much influence. The mean deceleration of the ship $(-\mathrm{d}V/\mathrm{d}t)$ and the ratio D_{16}/L_{pp} are indicated.

With regard to the track of the vessel while stopping, it is well known that, while working with an astern-turning propeller, the ship cannot be steered by the rudder. Therefore the rudder was always blocked in the middle position at all the trials. The tracks observed, which are completely erratic, are indicated in figure 7. This figure gives observed tracks for tankers analogous to tanker A. The turning circles for this latter ship are also indicated.

Furthermore, the geometrical endpoints of the path covered by the vessel after 12 minutes of manoeuvring are indicated, and the limit line of these endpoints gives an idea of the area into which the ship may run while stopmanoeuvring.

It can be seen from Table I that, for the loaded vessels (F and A′), the shortest stopways have semicircular

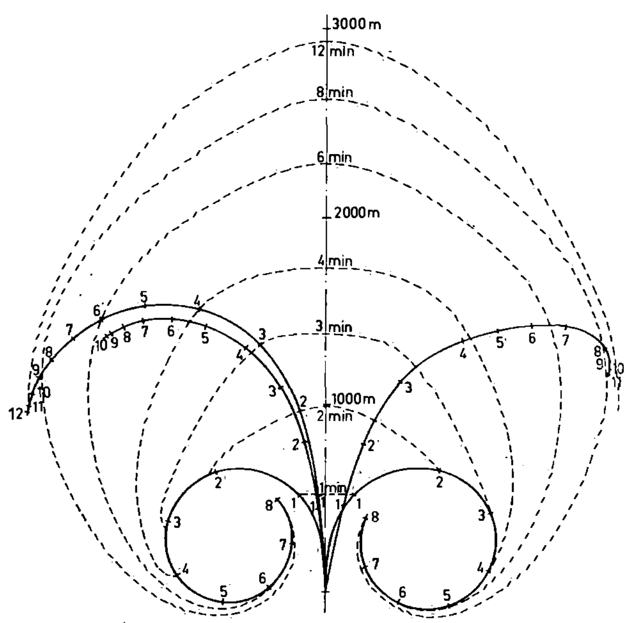

Fig. 7. Stopping tracks

paths, the longest stopway (D_{16} = 15 ships' lengths) following nearly a straight course. This seems to be so in most of the cases. Many stopways were measured and found to be between 12 and 13 ships' lengths for loaded tankers and about 10 ships' lengths for tankers in ballast. Another remarkable point is the constancy of the mean slowing down, which remains of the order of 0.01 m/sec^2. The published figures for the Idemitsu Maru [10] are of the same order: stopway 15 ships' lengths, deceleration 0.007 m/sec^2 for a vessel twice as large as the biggest ship tried.

The conclusion to be drawn from these trials is that a captain of a large tanker who gives the order "full-astern" to brake the ship takes a considerable risk. For nearly a quarter of an hour he will be incapable of either steering his ship or regulating his speed. He is at the mercy of any fixed or floating obstacle within the area indicated in figure 7. The tankers tried were only of 60,000 to 110,000 tons d.w. But the risks will grow with the dimensions of the vessel. It has already been stated and proved that the advantages to be expected from more elaborate manoeuvring or greater astern power from the engine are insignificant.

Therefore, two series of special trials were carried out in the hope of solving this problem.

2.4 *Use of rudder for stopping purposes*

Figure 7 suggests that if, before reversing the propeller, the rudder is turned, the ship in its gyration would have a more definite course.

Furthermore, such a manoeuvre with a large rudder angle would be of considerable use in braking the vessel. The special trials executed with tanker D and described in [4] confirm this. The half circles of the course pass into the tangent at this circle, opposite to the initial heading. The stopway in the initial direction becomes about three ships' lengths; the lateral transfer about five. The total length of the track is about 10 ships' lengths.

The aforementioned extra brake effect increases the deceleration to 0.018 m/sec^2 and decreases the stopping time to about 7 minutes for initial speeds superior to 17 knots.

2.5 *Use of reduced initial speed*

Another obvious solution consists in reducing the speed and thus beginning the manoeuvre with a lower initial speed. The question is how to know the quantitative influence of reduced speeds, as these are used in practice during the fog.

Special trials of this sort have been carried out with tanker E and are described in [5]. The results are also given in figure 6. With an initial speed of between 10 and 16 knots, the stopping remains practically proportional to that speed. With very low speeds, the variation is parabolic. At 10 knots initial speed, the stopway is still eight ships' lengths, at 5 knots initial speed it is hardly more than two ships' lengths.

2.6 *Influence of depth of water*

Another special trial, which must still be carried out, will try to establish the influence of the depth of water under the keel. These large vessels will always be on restricted depths when coming into harbour, alongside quays, and so on. It should be noted that this sort of trial is only of interest at very low speeds. As it is more a question of manoeuvring than of braking of large vessels, it must be looked into more carefully and in more detail elsewhere.

2.7 *The braking of large vessels by the propeller: First preliminary conclusions*

In view of what has been said in the foregoing, a general solution of the braking problem seems at first sight somewhat uncertain. Still, some preliminary conclusions can be drawn.

The course of a large ship during stopping by means of the propeller put into reverse is absolutely arbitrary and cannot be changed by the rudder from the moment the engine turns astern. Only when a definite turning movement is given to the whole ship before the propeller is put into reverse will the vessel follow the direction of the rudder. The ship will then follow a predetermined track, which looks like a change for the better but which track can only be used when the turning side is "free". As stated earlier, the stopway is very little influenced by the engine's manoeuvring and developed power while going astern. This is less true when the speed is reduced.

In all cases, the main effect of an increase of backing power is to decrease the duration of the final phase or third period. But, owing to the small values of the speed during this period, the influence on the stopway remains very moderate.

If the tactic of beginning to turn with the ship just before braking is considered impractical, then the fact remains that for large vessels with an initial speed of 16 knots a stopway of about 13 ships' lengths must be reckoned with. Moreover one has to count on the erratic course of the ship.

The foregoing figures are reduced to about ten ships' lengths in ballast condition with full initial speed, and eight ships' lengths on full load at 10 knots initial speed. But only with a very slow initial speed may a real shortening of the stopping way be expected. The influence of shallow water is probably important, but no quantitative nor qualitative data are available for the moment.

3 Model trials

3.1 *Purpose of model trials*

The aforementioned rather definite first conclusions have induced the author to try to obtain more information by executing trials on models. Even before the results of the systematic sea trials were analyzed, trials with models were made. Some of these completed the study of the influences detected at sea trials; others made it possible to study solutions which do not occur on any real ship; finally, several such tests were carried out to try to find a correlation between the model and the real ship, so as to arrive at comparative predictions on the behaviour of future vessels.

During the six years that model tests were carried out at the towing tanks at Wageningen and Delft, three types of programs were investigated:

Program I is a replica of the sea trials of tanker E, constituting a repetition and a systematic development of these sea trials with the object of analyzing the different influences for this particular ship.

Program II is a study to investigate the performances of an original system of hydrodynamic braking flaps. In practice this program was partially executed along the lines of program I (see [1]).

Program III is a kind of appendix to program I and consists of special model trials with the object of being able to apply certain methods of correlation. A special chapter is devoted to this correlation at the end of this report.

3.2 *Program I – Braking by propeller*

The model trials for this program were carried out in the "shallow water" laboratory of the Netherlands Ship Model Basin at Wageningen, described in [11]. A perspective drawing of the basin is given in figure 8 and a horizontal section of it in figure 9.

The great breadth of this basin permits the free manoeuvring of a model of 3 m length, telecommanded by electronic devices without interference of the basin walls.

Table 2 gives the principal dimensions of tanker E and its model as well as those of the propellers and the braking flaps used during the execution of program II. The model on a scale of 1:80 navigated absolutely free. Later it was also equipped with unfolding braking flaps for program II, propulsive machinery, and electrical steering gear. All these devices were telecommanded from the carriage following a predetermined program.

The scheme of each trial was as follows:

a. The propeller was maintained at a certain predetermined constant number of revolutions depending on the initial speed for the stopping trial.
b. The model was maintained on a straight course.
c. When the model reached a constant service speed, the braking of the vessel by means of the propeller began, as predetermined by the program for the number of revolutions (in accordance with the program for the sea trials), from the initial number of revolutions ahead to the final number astern.
d. This final number of revolutions was maintained until the model stopped completely.
e. During phases c. and d., the model was free and the rudder was manoeuvred as predetermined by the program.

As stated under c., the stopping trials were set up in accordance with the "crash-stop" sea trials for tanker E. To repeat these sea trials as exactly as possible, a model program no. 1 was drawn up as indicated in figure 10. The rudder angle was maintained at zero degrees during the whole of these trials. As similarity in the number of revolutions for sea and model trials was sought, similarity of speeds could not be pursued (see B, second series of tests). This question is debated when the correlation tests are discussed at the end of the report (see also section 3.4).

Four series of programs were executed, which include the following trials:

A. First series

To study the influence of the manoeuvring of the engine, following the number of revolutions (program 1). This series is illustrated in figure 10. First there was the class 2 program (No. 2A to 2E), identical to program 1 up to the speed of 10 rpm astern, followed thereafter by different types of increasing rates of revolutions until the full stop of the vessel.

Next, class 3 program (No. 3A to 3G) was carried out. It was based on program 3, simulating the manoeuvring of a diesel engine between the same limits of numbers of revolutions (105 forward up to 50 backwards) as program 1, figure 10.

B. Second series

Repetition of the sea trials for study of the influence

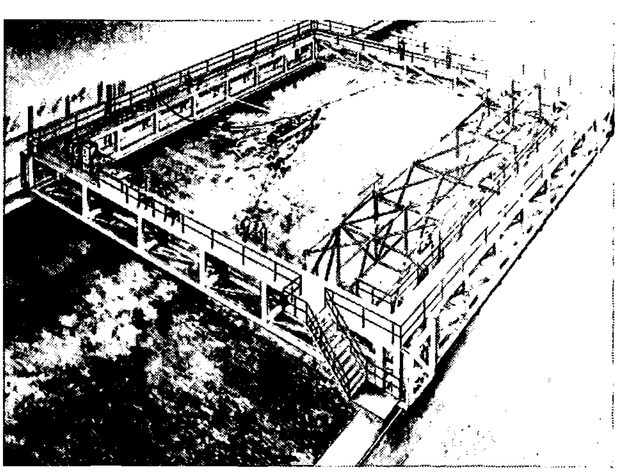

Fig. 8. Perspective drawing of model ship in the shallow-water laboratory of the Netherlands Ship Model Basin

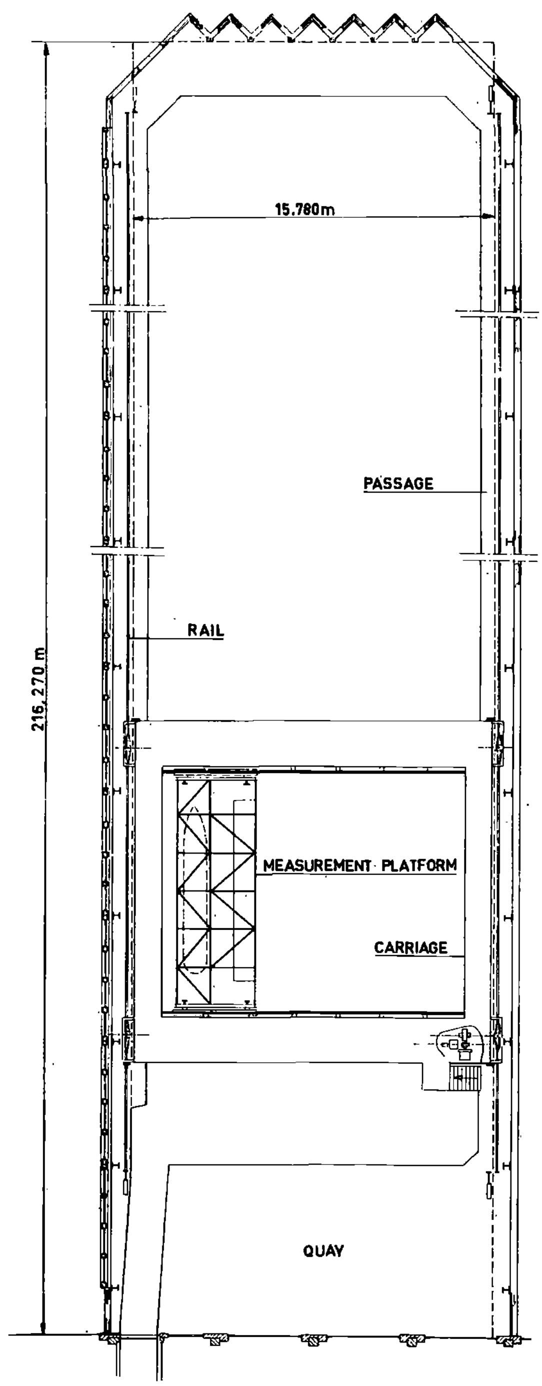

Fig. 9. Horizontal section of the shallow-water laboratory of the Netherlands Ship Model Basin

Table II. Principal dimensions for model, ship and their propellers. Tanker E of 65,000 tons (of 1,000 kgf) d.w. (scale 1 : 80)

		Loaded condition		Ballast condition	
		ship	model	ship	model
Model No. 3093 (Wageningen)					
Length between perpendiculars		237.74 m	2.9718 m	237.74 m	2.9718 m
Length on waterline		244.64 m	3.0580 m	232.60 m	2.9075 m
Moulded breadth		34.80 m	0.4350 m	34.80 m	0.4350 m
Draught forward		12.99 m	0.1624 m	7.35 m	0.0919 m
Draught aft		12.99 m	0.1624 m	9.30 m	0.1163 m
Mean draught		12.99 m	0.1624 m	8.325 m	0.1041 m
Displacement		86,743 m³	169.42 dm³		
Wetted surface without appendices		12,134 m²	1.8960 m²	9,771 m²	1.5267 m²
Wetted surface rudder included		12,366 m²	1.9322 m²	9,873 m²	1.5427 m²
Block coefficient (between p.p.)	C_B	0.808	0.808		
Coefficient of midship section	C_M	0.991	0.991		
Prismatic coefficient (between p.p.)	C_P	0.815	0.815		
Centre of buoyancy after forward perpendicular		115.19 m	1.440 m		
Propeller No. 1492 Type: series B (Wageningen)					
Number of blades		4	4	4	4
Diameter	D	7,200 mm	90.00 mm	7,200 mm	90.00 mm
Pitch (right handed)	H	5,000 mm	62.50 mm	5,000 mm	62.50 mm
Pitch ratio	H/D	0.694	0.694	0.694	0.694
Boss diameter ratio	d/D	0.187	0.187	0.187	0.187
Blade area ratio	A_0/A	0.539	0.539	0.539	0.539

of the initial speed [5]. This series also was based on program 1. A special program 1D was executed, extrapolating the forward number of revolutions at sea up to 140 per minute to give the model an initial speed comparable with the initial speed of the vessel at sea. Furthermore, programs 4 (figure 11) and 11 (figure 12) were carried out, thereby simulating special trials of tanker E. Programs 4A (figure 11), 7 and 10 (figure 12) closed this series.

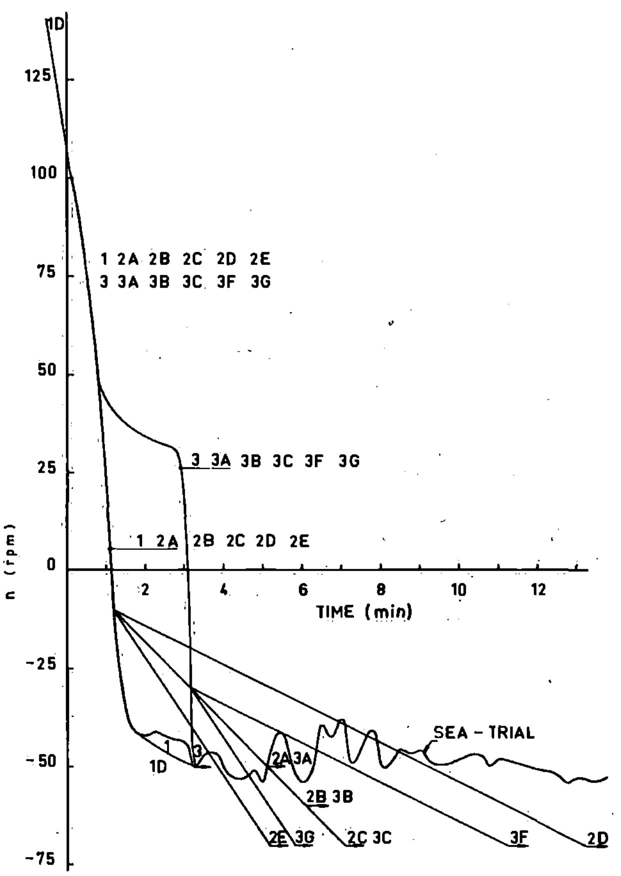

Fig. 10. Number-of-revolutions-programs (1, 2 and 3) for full-speed ahead (140 and 105 rpm)

3.2.1 *Influence of displacement*

C. Third series

Executed to investigate the influence of displacement. The model was ballasted as indicated in Table II and

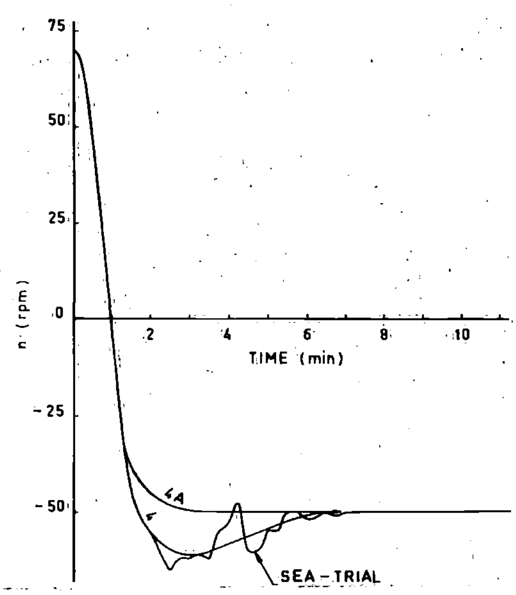

Fig. 11. Number-of-revolutions programs (4) for half speed ahead (70 rpm)

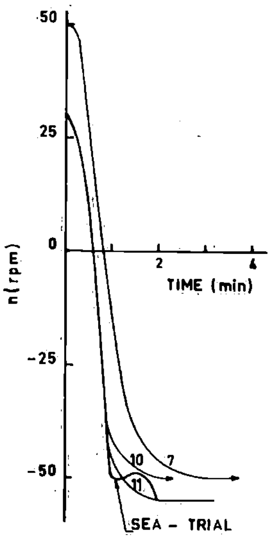

Fig. 12. Number-of-revolutions programs (7, 10 and 11) for slow-speed ahead (50 and 30 rpm)

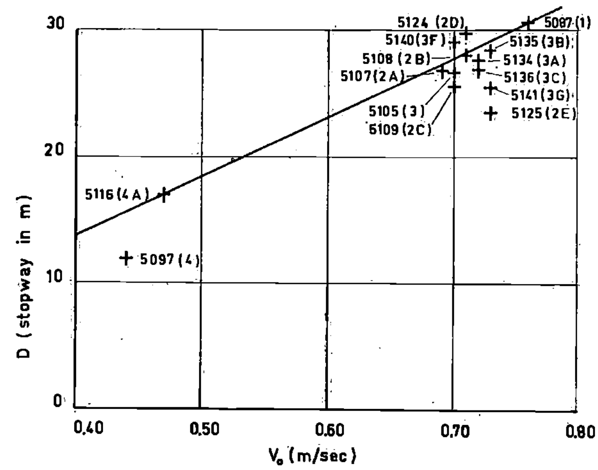

Fig. 13. Influence of manoeuvring of the engine for tanker E in loaded condition (the first number indicates the trial, the second (. . .) is the number of the program of number-of-revolutions)

trials were made in the conditions described as programs 1, 2E, 3, 3G, 4A, 7 and 10.

3.2.2 *Shallow water effect*

D. Fourth series

To investigate the shallow water effect, all the trials were executed in accordance with program 1 for the number of revolutions, the ship being tried fully loaded and in ballast. The "real" depth of water tried with the model corresponded to 73, 30, 25, 20 and 15 meters.

3.2.3 *Results of model tests*

The results of all these model tests represent a considerable volume of data. As these concern only tanker E, the results given in a number of tables in [5] are not repeated here. It is only of importance to consider what these trials confirmed or added to the knowledge about the actual ship. The following figures indicate therefore only the most important results.

Figure 13 shows the influence of the manoeuvring of the engine (series of trials A). Though the model probably overestimates that influence, its effect seems small at normal speed. Only in program 2E some gain is shown, but this fast manoeuvring does not seem practicable. At low speed the difference between manoeuvres 4 and 4A shows that the manoeuvring of the engine has more effect at such a low speed. The model test confirms here the theoretical analysis.

The results of the repetition of the sea trials (series of trials B) are shown in figure 14. The curve for the model tests and that for the sea trials are similar, and the conclusion is that, within a limited zone, the model results are transposable to the ship, for comparative purposes only.

This observation lends credit to figure 15, which indicates the influence of the displacement (series of trials

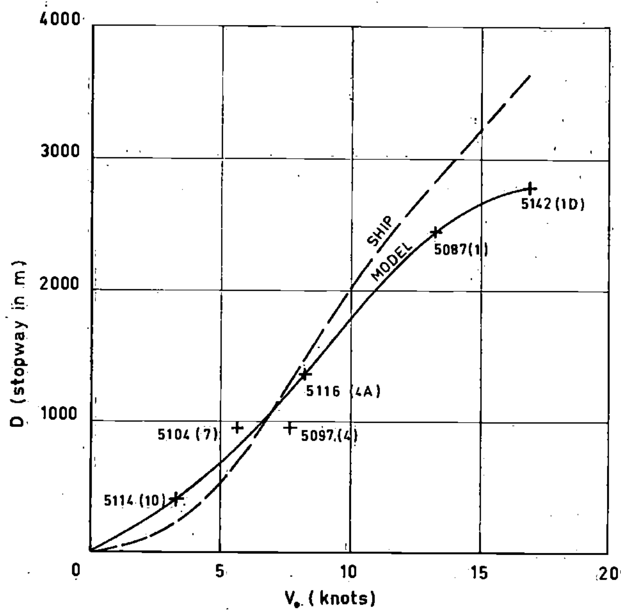

Fig. 14. Correlation model–ship (the first number indicates the trial, the second (. . .) is the number of the program of number-of-revolutions)

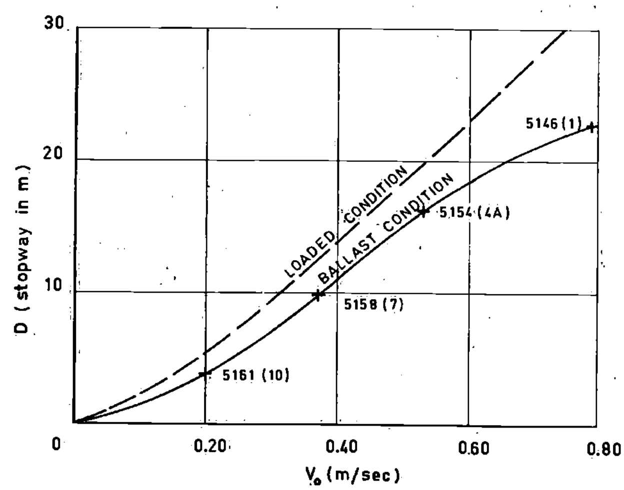

Fig. 15. Influence of displacement (the first number indicates the trial, the second (. . .) is the number of the program of number-of-revolutions)

C). It seems that this influence is relatively small: the stopway is reduced by one fourth for half the displacement.

The results of the tests on the influence of the water depth (series of trials D) did not lead to clear indications. The models were following very irregular courses and no deductions could be made from these tests. It seems that other model techniques must be invented with a view to achieving correlation between model and ship in this respect.

The trials concerning the special manoeuvring of the rudder have qualitatively confirmed the observations made during the sea trials. In this case also, correlation between model and ship is next to impossible to obtain.

3.3 *Program II – Braking by special devices*

This program was the first of the model tests executed already in 1962 at the Model Basin of the Delft University of Technology and was an attempt proposed by the author to come to a better and more effective solution of the braking of large vessels. This method of installing hydrodynamic braking flaps is described in [1] as Report No. 53 S of August 1963: "The braking of large vessels".

3.4 *Program III – Correlation model – ship*

When program I of the model tests was established, the initial object was to obtain from the results a comparison between ship and model. This comparison was to be a global one, without entering into details of how the parallel between these tests would be constituted. If this global comparison were satisfactory, an extension of program I to cover a critical evaluation of the different influences would be necessary. The global comparison has been covered in the description of program I but the transposition from the model to the ship which was then used was not accurate. If one is to use a free model in order to reproduce sea conditions, the initial condition has to be one of autopropulsion for the model; that is to say, one has to choose between the similarity of the number of revolutions or the similarity of the speeds, the two being impossible of realization at the same time (see 3.2).

In program I the choice of the similarity of the number of revolutions was adopted for practical reasons. Nevertheless, the extrapolation of the results from model to ship was actually made on a speed basis, because the speed was the essential parameter for the ship. But similarity as regards the ship's resistance, also essential, could not then be achieved because of the scale effect of the friction coefficient. To escape from this dilemma, the Model Basin at Wageningen proposed to forego a direct and global comparison and to use a factitious model trial in complete similarity with the ship, in the same way as is done in determining a ship's resistance by applying a friction correction to the friction of the model. On the hypothesis that, at any moment, the variables used were interdependent of each other, as if the whole procedure were stationary, the relations of stationary trials could be determined and thereafter one could reconstruct the whole trial as a continuous sequence of these stationary states. This so called "quasi-stationary" method was proposed by Van Manen and is explained by him [12] on the basis of the work done in the U.S. by Thau [13].

Program III consisted of three series of model tests:

No. 1. A first series of trials analogous to the autopropulsive trials at constant speed and variable decreasing number of revolutions. At each trial, the force, which exists between the model and the carriage, here called braking force K, was measured and thus $K = f(V \cdot n)$, the force of the model in a stationary state, was obtained.

No. 2. A second series of trials consisted of model stopping tests on a straight course. The program for the number of revolutions either belonged to program 1 or this number of astern revolutions was held constant and was fixed from the beginning of the trial. The second series was executed in the shallow-water towing tank at Wageningen.

The following experimental set up was used (see figure 16): The model was attached in its midship section to a cord, which was rolled around a wheel 1. On this wheel were mounted an electric light 1 and a photo-electric cell 2. In this way the relative movement of the model to that of the carriage was transformed into an oscillating rotation of the wheel, 1. Wheel 5, with a thousand circumferential teeth, was fixedly connected to the slotted wheel 4, which rolled along the rails of the carriage. This wheel 4 had one-meter circumference. The circumferential speed of the slotted wheel equals

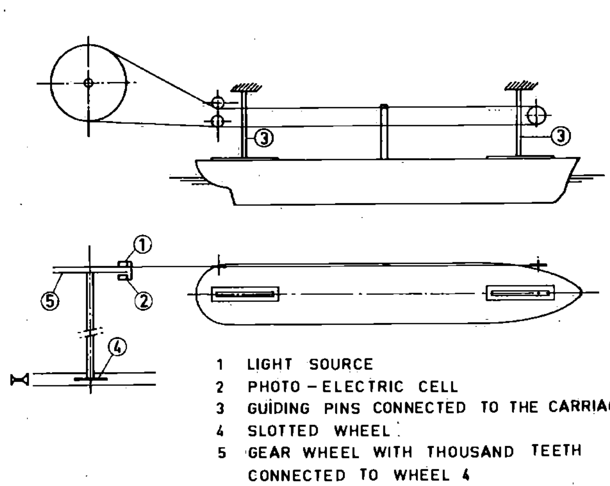

Fig. 16. Trials of model on a straight course (trial no. 2)

the linear speed of the carriage and the speed of the model was given by superimposing the speed of wheel 1 in relation with that of wheel 5.

No. 3. The third series of tests consisted of the normal stopping tests for the free model in accordance with the programs for the number of revolutions of program 1 (number 1, 4A, 7 and 10; see figures 11 and 12). For all these three series, the model was the same as the one for program I. The results of the first series No. 1 are indicated in figure 17; they include five different speeds. The results of the second series No. 2 are indicated in figure 18; they include only the trials with a constant number of revolutions. The results of the third series No. 3 and those of the second series No. 2 concerning the trials with a program of number of revolutions in accordance with the sea trials of tanker E are given in figure 19.

3.5 *Principle of quasi-stationary method*

As explained by Van Manen [12], it is possible to study the stopping of a vessel by means of the fundamental dynamic law. Force equals mass by acceleration, or, in this case

$$K = -\frac{\mathrm{d}[(M+m)V]}{\mathrm{d}t} \tag{1}$$

(Note: See note after first equation in section 2.3 "Conventional Trials" on page 11)

where K equals braking force, not only deriving from the mass M of the vessel at speed V but also from the mass m of the water carried along with and surrounding the ship, which is supposed to advance also with the mean speed V.

The global effect of this surrounding water is included in equation (1). Hence the fictive mass $(M+m)$ of the vessel is only considered when solving this equation.

In a first approximation, Van Manen [12] has admitted that m = constant and a certain percentage of M. He takes therefore $M+m = 1.05M$. If V_0 represents the initial speed at the moment t_0 and V represents the speed at an arbitrarily chosen moment, the integration of (1) immediately gives the time difference $(t-t_0)$ necessary to reduce the speed V_0 to V.

$$t-t_0 = -(M+m)\int_{V_0}^{V}\frac{1}{K}\cdot\mathrm{d}V \tag{2}$$

In the same way the distance covered $(S-S_0)$ is found, while keeping in mind that $\mathrm{d}V/\mathrm{d}t = (\mathrm{d}V/\mathrm{d}S)\cdot(\mathrm{d}S/\mathrm{d}t) = V(\mathrm{d}V/\mathrm{d}S)$, by the integration

$$S-S_0 = -(M+m)\int_{V_0}^{V}\frac{V}{K}\cdot\mathrm{d}V \tag{3}$$

If the integrals are continued until $V = 0$, formulas (2) and (3) give respectively the stopping time and the stopway until the "full stop" of the ship from an initial speed V_0. Both formulas can be used both for the vessel and for its model. As $M+m$ is supposed to be a known constant in this approximation, it is only necessary to apply (2) and (3) to know the relationship between K and V. Here the quasi-stationary hypothesis makes its appearance.

The trials no. 1 have made it possible to determine for a model in a stationary state the relation $K = f(V,n)$, represented graphically by figure 17.

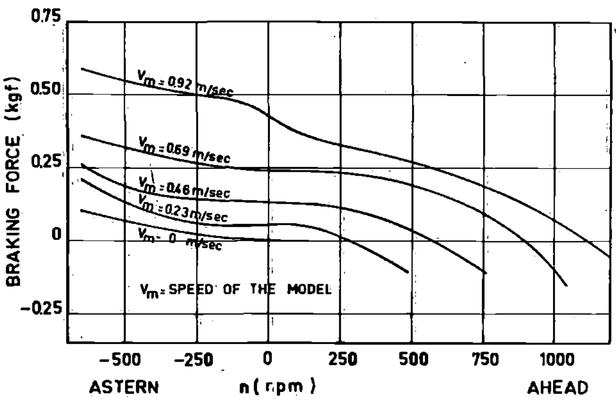

Fig. 17. Braking force of model on a straight course, measured as a function of the number of revolutions of the propeller (trial no. 1)

In this case it is admitted that the variation of V and n (number of revolutions) is slow enough during a stopping test to remain valid within that interval for this relationship. Especially when n = constant during the stopping manoeuvre, the relation $K = g(V)$ may be immediately obtained by reading the values of the ordinates from figure 17 in relation to the abscisses n.

If n is variable as a function of the time, one has to proceed by successive approximations by choosing arbitrarily a function $V = h(t)$ as a first approximation, which will give a second approximation of the same function, so that at the end the convergence by an iteration process is given. In principle, this process makes it possible to calculate, starting from figure 17, representing the trials no. 1, the results of a stopping test from the same model executed at a given program of number of revolutions.

Now, what one wants are not the results from the model, but from the ship. Therefore, it is necessary to deduce the braking force K_1 of the ship from the braking force K of the model. To achieve this, the following reasoning is followed: The braking force K_1 may be considered as the sum of the propeller thrust and the hydrodynamic resistance of the underwater body of the ship, the interaction between these two terms being neglected because the relationship between them is unknown.

By the same hypothesis it is admitted that the propeller thrust only depends on V and n. If K is the braking force of the model deduced from trial no. 1 for a torque given by V and n, the part of K due to the propeller thrust is the same for the vessel, while the hydrodynamic resistance of the underwater body is diminished by the friction correction R_a, which is a function of the speed only, and which, if convenient, may be augmented by different corrections, ΔR, due to rugosity, meteorological influence, and so forth.

In the case that a ship trial should actually have been performed, it should be possible to evaluate these corrections with a certain amount of precision through this ship's propulsion trial. On the other hand, if one wants to predict results from sea trials, one must take care not to be too optimistic. However, it seems sufficient to appraise the rugosity by means of previous examples and to neglect further corrections.

The proposed method is simple and complete, but it includes the hypothesis of the constancy of $(M+m)$, a value arbitrarily chosen, which leads to a fairly rough approximation. Consequently, the Wageningen Model Basin considered the possibility of determining experimentally the value of $(M+m)$. This was the goal of trials no. 2. Indeed each of these trials gives the relationship $V = h(t)$ achieved with a program $n = j(t)$, chosen expressly, figure 18, and consequently determines the relationship between V and n, which makes it possible by means of the hypothesis of quasi-stationarity to define the braking force K at each moment as a function of V.

This is done with the help of the results of the trials no. 1, figure 17.

Returning to the integration of formula (1), considering that $(M+m)$ is variable, one finds

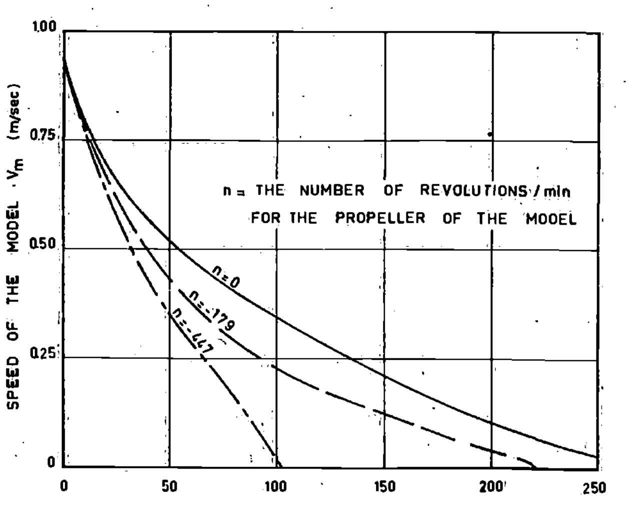

Fig. 18. Stopping trials for model on a straight course (trial no. 2)

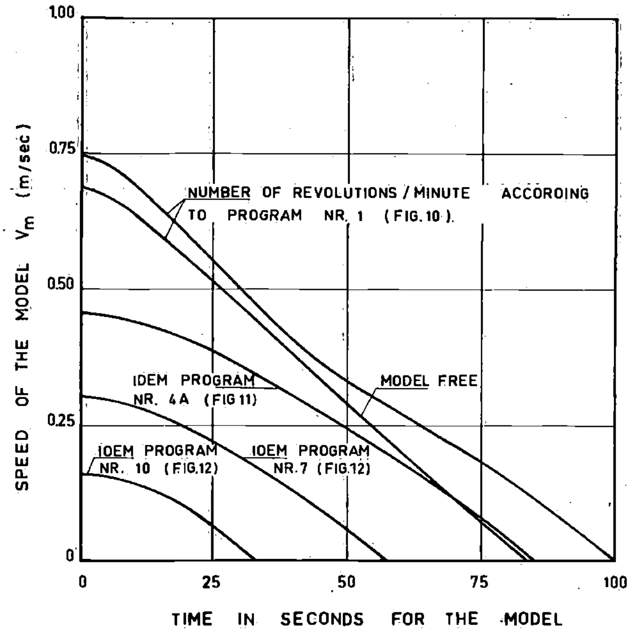

Fig. 19. Stopping trials for free model (trial no. 3) and for model on a straight course (trial no. 2)

$$\int_{t_0}^{t} K\cdot dt = -[(M+m)V]_t + [(M+m)V]_{t_0} \qquad (4)$$

If the limit of the integration is t_e where $V = 0$, one arrives at

$$\int_{t_0}^{t_e} K\cdot dt = [(M+m)V]_{t_0} \qquad (5)$$

because at that moment

$$[(M+m)V]_{t_e} = 0$$

Combining equations (4) and (5) it is clear that

$$\int_{t_0}^{t} K\cdot dt = -[(M+m)V]_t + \int_{t_0}^{t_e} K\cdot dt$$

or

$$[(M+m)V]_t = \int_{t}^{t_e} K\cdot dt$$

and

$$[(M+m)V]_t = \int_{t_e}^{t} K|dt| \qquad (6)$$

The result of this calculation is given in figure 20 for all no. 2 trials worked out for a final number of revolutions of 50 rpm astern (continuous line).

This figure shows that the accuracy and the reproducibility of the trials are excellent since all curves coincide at the speeds where the rate of -50 rpm is stabilized.

But in the beginning of the trial, where the number of revolutions is variable, and rapidly varying, the curves are frankly diverging. Also the dotted line of figure 20, representing the trial at a constant number of revolutions of -20 per minute, is clearly different

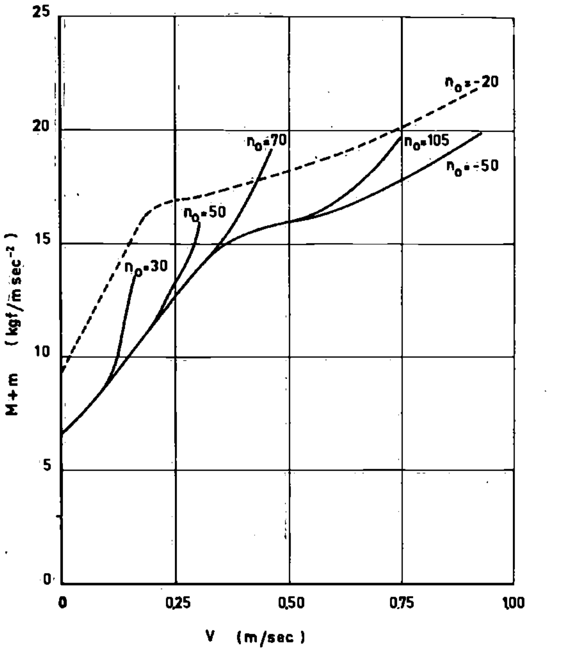

Fig. 20. Model of tanker of 65,000 dwt: $(M+m)$ as function of the speed

from the line of -50 rpm although with a similar outline.

Furthermore, M for the model is 17.28 kgf/msec^{-2} and thus $1.05M = 18.14$ kgf/msec^{-2}. Figure 20 shows that $M+m$ deduced from trials 1 and 2 varies approximately between 6.5 and 22 and depends clearly both on the speed and on the number of revolutions. This proves that, if the validity of this procedure to determine $(M+m)$ is accepted, the approximation of $(M+m) = 1.05M$ is too rough and hardly susceptible of amelioration by taking into account any other factor than 1.05. The Model Basin in Wageningen tried a better approximation by adopting, as the relationship between $(M+m)$ and V, a mean curve deduced from figure 20 and neglecting the influence of n on $(M+m)$. Alas, if one adopts this second approximation, a new difficulty turns up when evaluating the extrapolation to the ship. A supplementary hypothesis is required for the extrapolation of $M+m$; viz. that m/M depends only on the Froude number.

If this second hypothesis is accepted, the calculation can be continued as for the first approximation. The integration of formula (1) gives instead of formulas (2) and (3) the following

$$t-t_0 = -\int_{V_0}^{V} \frac{M+m}{K+V\dfrac{dm}{dt}}\,dV \tag{7}$$

$$S-S_0 = -\int_{V_0}^{V} \frac{M+m}{K+V\dfrac{dm}{dt}}\,V\cdot dV \tag{8}$$

The K_1 values necessary for the extrapolation from K to the ship's braking force are estimated as for the first approximation, but it is difficult to allow a value for dm/dt, which is not absolutely arbitrary. Theoretically this difficulty can be overcome if one starts by transforming formula (1) to

$$K = \frac{-d\,[(M+m)V]}{dV}\cdot\frac{dV}{dt} \tag{9}$$

and integrating this formula to

$$t-t_0 = -\int_{V_0}^{V} \frac{(M+m)+\dfrac{Vdm}{dV}}{K}\,dV \tag{10}$$

and

$$S-S_0 = -\int_{V_0}^{V} \frac{(M+m)+\dfrac{Vdm}{dV}}{K}\,V\cdot dV \tag{11}$$

so that the differential quotient dm/dt is replaced by dm/dV. The last is known because the hypothesis was that $(M+m)$ is a function of V.

Now, figure 20 shows that dm/dV depends to a large extent on the program of the number of revolutions, so that a value deduced from the mean curve of figure 20 does not represent an admissible approximation. It seems extremely inconvenient to find a suitable experimental procedure to overcome this difficulty; therefore, it is to be feared that notwithstanding its greater complications the second approximation does not serve our purpose any better than the first one.

So a more direct way of calculation is necessary. One must try to avoid the calculation of $(M+m)$. Coming back to the hypothesis that $(M+m)$ only depends on the Froude number, one admits that this may correspond to a fixed program of the number of revolutions. Both formulas (10) and (11) can be written as

$$y = \int_{V_0}^{V} \frac{f(V)}{K}\,dV \tag{12}$$

y representing here one or the other of the terms $t-t_0$ or $S-S_0$, and $f(V)$ representing here one or the other of the terms $[(M+m)+Vdm/dV]$ or $[(M+m)+(Vdm/dV)]V$.

This third approximation to determine y is best handled in the following way: One begins with a model test of a certain program of number of revolutions corresponding to the program of a real ship on which stop tests have been carried out. Starting with the same initial speed, taking into account Froude's law, the functions y are experimentally determined, starting from a speed V. This test is executed by applying to the model, until the moment that it becomes free of the carriage, a tensile force equal to the friction correction as set by the ITTC. Thus the braking force K of the model may be determined as given in figure 17.

If a fictitious trial is now imagined, where this tensile force is not only applied before the stopping manoeuvre but also during that whole manoeuvre itself, it must be possible to achieve the corresponding braking force K_1 of the vessel.

This K_1 may be calculated by formulas (2) and (3). Then the functions y_1 for the vessels are given by the same formula (12), replacing K by K_1. Hence

$$\frac{dy_1}{dV} = \frac{dy}{dV} \cdot \frac{K}{K_1} \tag{13}$$

In the right-hand part of the equation, all the values are known, dy and K by the trial, K_1 by the calculation.

It is now sufficient to calculate the integrals $\int dy_1/dV$ to obtain y_1 as regards the fictitious model trial in total similarity to the sea trial. This procedure gives a possible extension to the solution of the problem. Submitting the model permanently to a variable traction, which equals the friction correction for the given speed by means of a convenient dependable contraption, it will be possible to realize a fictitious trial; and an extrapolation of the results to the ship is true if it is admitted that there is no scale effect other than that of the friction. This procedure requires another set of experiments, but on the other hand the trials no. 1 may be omitted. The extrapolation does not use any stationary trial nor the basic hypothesis of the quasi-stationary method.

3.6 *Results obtained*

To the tanker E of 65,000 tons d.w. of 1000 kgf, the different methods have been applied. The friction correction has been calculated following the ITTC formula of 1957 with a roughness allowance $\Delta C_f = 0.0002$. This coefficient was deduced from the propulsive sea trials of the same day. As the weather was fine (Beaufort scale No. 1), no other correction was applied. With these figures, the relationship between the braking forces of the ship and its model at corresponding speeds remained in the neighbourhood of $\frac{2}{3}$ (taking account of the scale), from full speed to half speed. Consequently, whatever the extrapolation procedure may be, the predictions for the ship are expected to be about 50 percent higher than the uncorrected results obtained with the model.

No model trial was directly comparable with the sea trials, but it is evident, as a comparison with figures 18 and 19 confirms, that the trial starting with 16 knots initial speed and propeller in reverse with a constant number of revolutions of -50 is optimistic in relation to the trial where the number of revolutions progressively varies from $+105$ to -50 rpm. Now the stopping time of the model was 100 sec, which means 15 min for the vessel. In reality this time was measured as $13\frac{1}{2}$ min. The complete extrapolation calculation by the third approximation method at the rate of -50 rpm gives in fact a stopping time and a headreach which are about 50 percent higher than those achieved with the ships in reality.

It seems therefore that none of the three approximations for trying to achieve a correlation between the behaviour of the model and of the ship during stopping manoeuvres is sufficiently successful to be definitely adopted. But they are represented here because this paper gives an explanation of why it has become necessary to search in other directions to find a solution to this problem.

This lack of success in predicting from model tests the behaviour of a large vessel during "crash-stop" manoeuvres is probably due to the large-scale effect and to the fact that the phenomena during backing are not even approximately stationary, especially during the period where the propeller is sucking air from the atmosphere (ventilating).

Therefore the question of correlation of "braking by the propeller" tests with "crash-stop" sea trials worked out by simple application of a correlation coefficient is not yet solved, though there are indications that by analyzing further tests the method of quasi-stationary measurements still may give useful results. However it must be kept in mind that it is premature to suppose that such a coefficient is not influenced by scale effect and it will be necessary to accept finally a more or less close approximation.

Nevertheless, it is feasible perhaps, considering the relatively small variation in scale between tankers of 100,000 and 500,000 tons d.w., to view the first as a model of the second and to extrapolate the results of "crash-stop" sea trials from one ship to the other, taking into account the corresponding speed coefficients and respecting also the similarity of the number of reverse revolutions during the manoeuvre.

4 General conclusions

The stopping conditions of a large vessel in general and of large tankers in particular have a precarious character. For an initial speed of 16 knots at full load the stopping time is about 15 minutes and the stopway about 13 ship's lengths. When the initial speed is lowered, the stopping characteristics are reduced accordingly. The effect of shallow water remains to be made clear.

The "dead time" following the order "full-speed astern" is important, but after that time the manoeuvring of the engine has little effect on the stopping and the backward power developed has not much influence either.

If circumstances permit, an intelligent use of the rudder may help to put the ship on a predetermined track and thus shorten the headreach, but this implies the necessity of beginning a turning circle.

The use of folding-out braking flaps as described in [1] constitutes an important improvement in the conditions for stopping.

These flaps may be used in all circumstances. From the model tests it seems that the stopway may be reduced by about half its length and the course stability of the ship is favourably influenced, though the same uncertainties in manoeuvring when backing with the propeller persist in both cases either with or without flaps.

Since 1963 an evaluation of the weight and the cost of installation of braking flaps has been available. One of the big Dutch shipyards was willing to give a rough estimation of the type of construction suggested in [1]. The weight estimation is:

Weight of one pair of flaps	29 tons
Weight of watertight cases in the ship	20 tons
Weight of hinges	5 tons
Weight of foundations for hydraulic system	2 tons
Weight of compensation in the structure due to the 2 notches in the ship's hull	4 tons
Total weight of hydraulic installation	5 tons
Total extra weight for installation of two flaps in W.T. cases + hydraulic installation	65 tons

Price for the whole installation, about ƒ230,000, or about $ 70,000 when installed in the Netherlands.

In general model tests reproduce well the particularities observed on board and certainly give approximate information about the different influences.

A quasi-stationary approach as described in [12] is an attractive model test technique for qualitative considerations of the stopping abilities of large ships. However, in the present state of knowledge, a fair correlation between model trials and sea trials has not yet been reached. The basis of the correlation is not yet complete and therefore a gap exists between the prediction from the model and the real stopping time and stopway. As long as this gap in the model tests and in our knowledge is there, it may be fruitful to use "crash-stop" sea trials for ships of different dimensions in order to reach comparative solutions to the problem of the braking of large vessels. It certainly will be useful to continue sea trials and the corresponding model tests to gain more ample information about the parallel between models and ships.

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